DESIGN AND ANALYSIS OF DIFFERENTIAL GEAR IN AUTOMOBILE

A project report submitted in partial fulfilment of the requirement for the award of the degree of

BACHELOR OF TECHNOLOGY

IN

MECHANICAL ENGINEERING

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CERTIFICATE

This is to certify that the Project Report entitled "**DESIGN AND ANALYSIS OF A DIFFERENTIAL GEAR IN AUTOMIBLE**" being submitted by Patoju Sai Sankar (318126520105), Pudi Teja Kumar (319126520119), Gude Raghu (318126520081), Kallapalli Ranjith Kumar (318126520107), Sunkara G S Meher Teja (319126520122). In partially fulfilment for the award of Degree of Bachelor of Technology in Mechanical Engineering. It is the work of bonafide, carried out under the guidance and supervision of Mr. D.S.S. Ravi Kiran, Dr. K Naresh Kumar Assistant Professors Department of Mechanical Engineering during the academic year 2018 to 2022.

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ABSTRACT

The main function of differential is to allow the back wheel to rotate at different rpm while receiving the power from the engine. During this process, severe stresses will be developed in the differential. Therefore, the materials selected for the differential has to withstand the applied torque acting on it and effectively transfer the torque to the wheels without any failure.

Within this work, an attempt was made to alter the material properties of differential and study its behaviour against the different toques applied to the input shaft. The geometrical model of the differential was developed by using Solid Works 2015 for the four-wheeler vehicle. Further, the model was imported to Ansys 2020 version to perform static behaviour at different moments applied on the input shaft of differential system.

The materials for the differential are considered as grey cast iron, Al Alloy (Al 6160), Ti alloy (Ti64), Structural Steel. The analysis was performed by altering the material propertied and input torques of 100Nmm, 500 NM, 1000Nmm & 1500 Nmm applied to the input shaft. The values of vonmises stresses and deformations by varying the input torques and materials are tabulated and noted.

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CHAPTER 1

INTRODUCTION

1 INTRODUCTION

1.1 DIFFERENTIAL

The differential allows each rear wheel to turn at different speeds. During cornering but at the same time, it gives equal torque to each wheel when both wheels have the same traction. A system of gears in the differential arrange in such a way that it connects the propeller shaft to the rear axle. The difference in a word intends to provide relative movement to rear wheels.

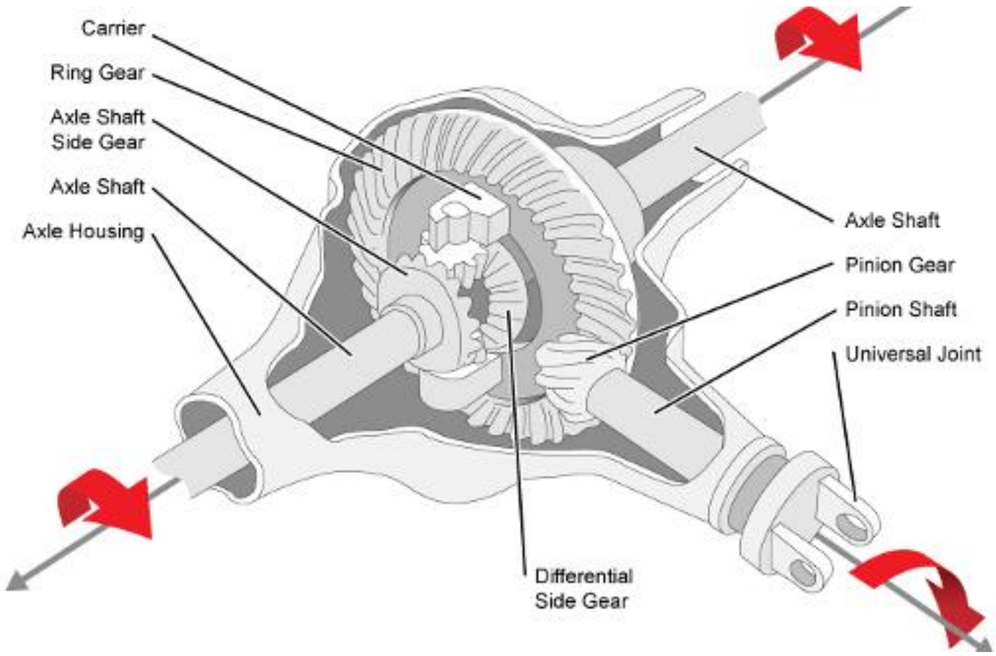


Fig 1.1 Differential

1.2 NEED OF DIFFERENTIAL

- The differential allows the non-steering wheels to rotate at different speeds so the car can corner without putting undue wear on the tires.
- The wheel on the inside of a turn moves a shorter distance compared to the outer wheel.
- If the axle does not allow the wheels to turn independently of each other, the tire of one wheel will be pulled across the ground.

1.3 PARTS OF DIFFERENTIAL

1. Differential side gear or sun gears

2. Pinion shaft or cross pin
3. Axle shafts or half shafts
4. Ring gear or crown wheel
5. Drive pinion or bevel pinion
6. Differential pinions or planet gears
7. Differential case or Housing

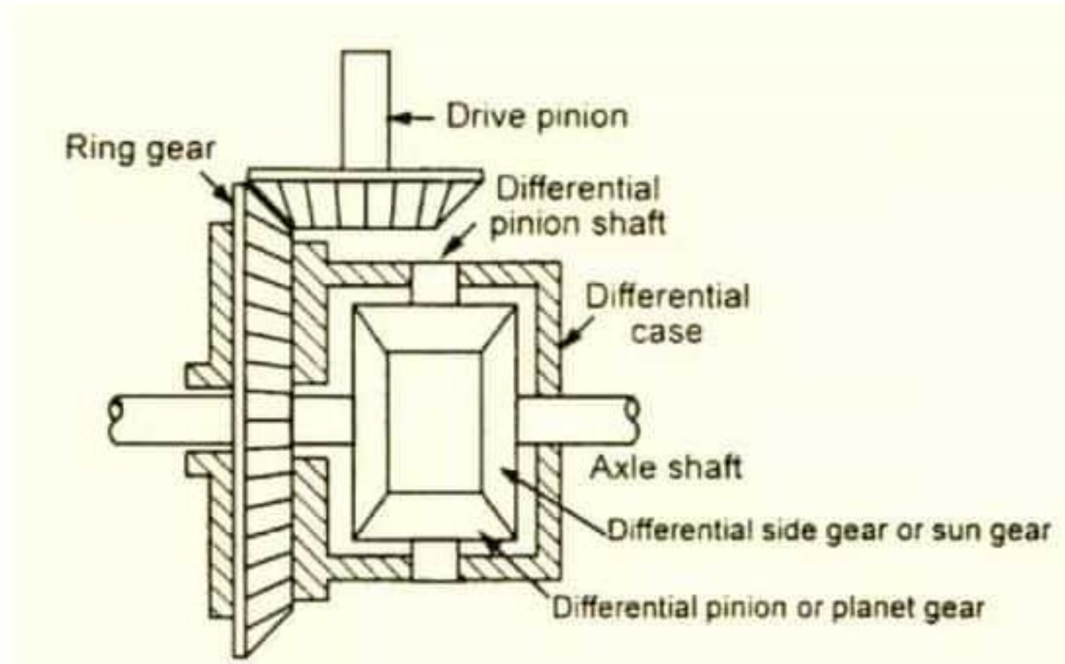


Fig 1.2(a) Parts of Differential

1.3.1 Differential Side Gear or Sun Gears

The differential consists of a small bevel gear called differential side gear or sun gear. It is mounted on the inner ends of each axle. In this, two bevel gears are fixed together to combine both the driving and driven shafts at 90° angles.

1.3.2 Pinion Shaft or Cross Pin

There are two pinion gears and their supporting shaft is called the pinion shaft. It is fitted in the differential case.

1.3.3 Axle Shafts or Half Shafts

An axle shaft is a solid shaft that is located between the differential and gear set of an axle housing. It transfers rotational force from the transmission system to the wheels attached to the axles.

1.3.4 Ring Gear or Crown Wheel

The ring gear is also known as a crown wheel. They act as an equalizer in dividing the torque between the two driving wheels while allowing one to turn faster than the other.

1.3.5 Drive Pinion or Bevel Pinion

The drive pinion is also known as bevel pinion. It is assembled with a differential housing called a differential case or carrier.

The driver shaft is connected to the drive pinion by a universal joint and it engages with a ring gear. Therefore, when the driver rotates the shaft the drive pinion rotates, and thus, the ring gear rotates.

1.3.6 Differential Pinions or Planet gears

Planetary gears are used in the differential. Since the axles of the planetary gears rotate around the common axis of the sun and ring gear that coincides and rolls in the middle of the differential system.

1.3.7 Differential Case or Housing

A differential case is attached with two-wheel axles and differential side gears. It is consists of bearings that rotate two axle shafts.

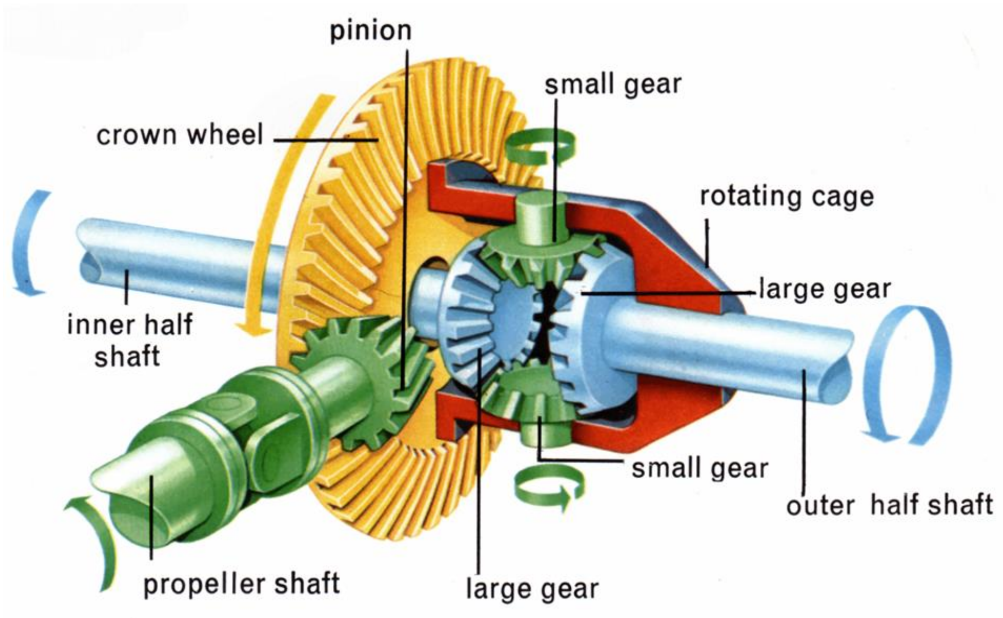


Fig 1.2 (b) Parts of Differential

- A small bevel gear called differential side gear is mounted on the inner ends of each axle.
- Two bevel gears are placed together to connect both driving and driven shafts at an angle of 90°.
- The differential case is connected with two-wheel axles and differential side gears.
- The differential case has bearings that rotate two axle shafts. Then, two pinion gears and their supporting shaft called pinion shaft, fit to the differential case.
- Then, the pinion shaft meshes with two differential side gears connected to the inner ends of the axle shafts.

1.4 OPEN DIFFERENTIAL

A differential in its most basic form comprises two halves of an axle with a gear on each end, connected together by a third gear making up three sides of a square. This is usually supplemented by a fourth gear for added strength, completing the square.

This basic unit is then further augmented by a ring gear being added to the differential case that holds the basic core gears – and this ring gear allows the wheels to be powered by connecting to the drive shaft via a pinion.

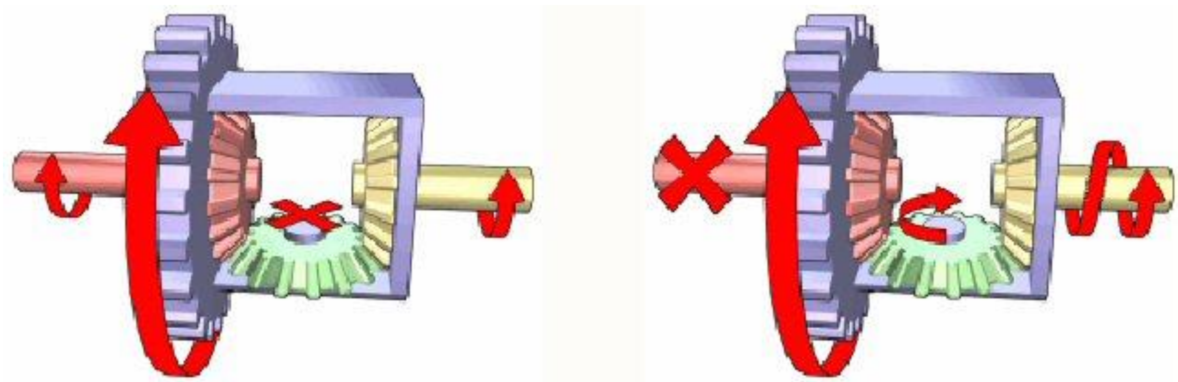


Fig. 1.3 Open Differential

In this example you can see the three sides of the internal gearing that make up the core mechanism, with the larger blue gear representing the ring gear that would connect to the drive shaft. The left image shows the differential with both wheels turning at the same speed, while the right image illustrates how the inner gears engage when one wheel turns slower than the other.

This gearing arrangement makes up the open type differential, and is the most common type of The benefit of this type is mostly limited to the basic function of any differential as previously described, focusing primarily on enabling the axle to corner more effectively by allowing the wheel on the outside of the turn to move at a faster speed than the inside wheel as it covers more ground. It does also benefit from its basic design being relatively cheap to produce.

The disadvantage of this type is that because the torque is split evenly between both wheels, the amount of power able to be transmitted through the wheels is limited by the wheel with the lowest amount of grip.

once the traction limit of both wheels combined is reached, the wheel with the lowest amount of traction will begin to spin – reducing that limit even further as there is even less resistance from the already spinning wheel.

These types of differences are the most common that are easily found in passenger cars. It only allows individual wheel speed or slips to be changed. In good road conditions, this allows the outer wheel to spin at a faster speed than the inner wheel.

1.4.1 Advantages of Open Differential

- This allows for different wheel speeds on the same axle, which means there will be no wheel slip when going around a corner, as the outer tire will travel further.
- From an efficiency point of view, there will be less energy loss through differential than other types.

1.4.2 Disadvantages of Open Differential

- When traction is reduced in one wheel, it substantially limits the amount of power produced by the vehicle. If one wheel cannot dissipate as much power, the other will receive an equally small amount of torque.

1.5 FUNCTIONS OF DIFFERENTIAL

The main function of differential is to allow the back wheel to rotate at different rpm while receiving power from the engine

When a car turns a corner, one wheel is on the "inside" of a turning arc, and the other wheel is on the "outside." Consequently, the outside wheel has to turn faster than the inside one in order to cover the greater distance in the same amount of time. Thus, because the two wheels are not driven with the same speed, a differential is necessary. A car differential is placed halfway between the driving wheels, on either the front, rear, or both axes (depending on whether it's a front-, rear-, or 4-wheel-drive car). In rear-wheel drive cars, the differential converts rotational motion of the transmission shaft which lies parallel to the car's motion to rotational motion of the half-shafts (on the ends of which are the wheels), which lie perpendicular to the car's motion.

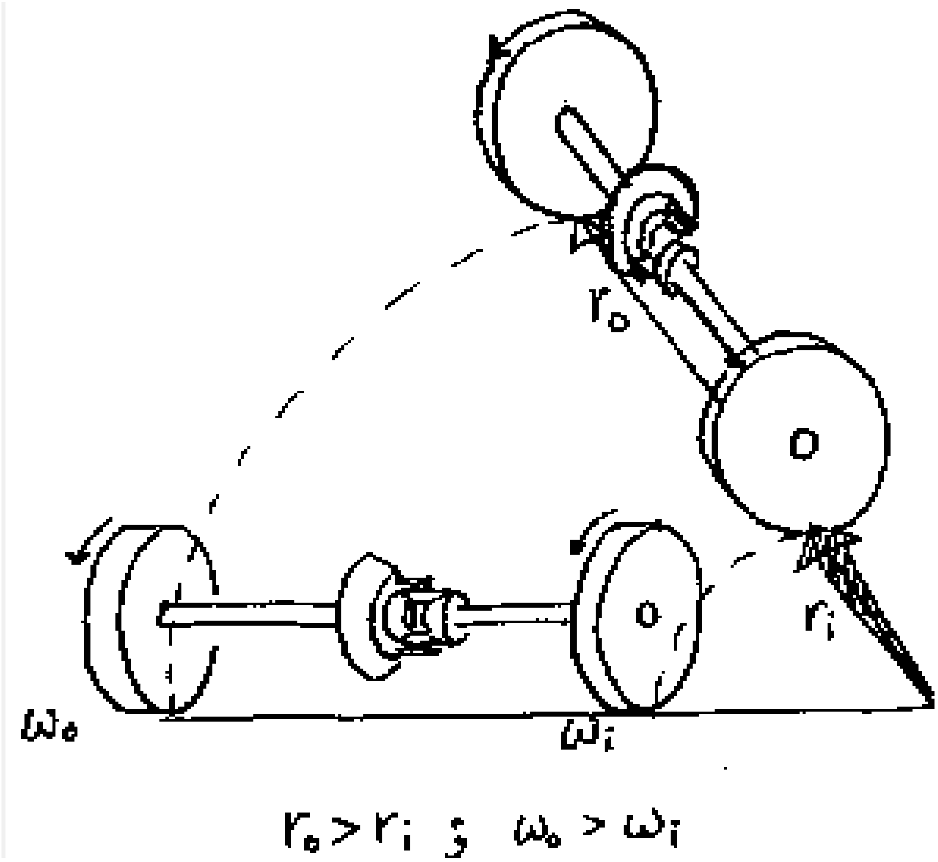


Fig.1.4 Wheels while making Right Turn

1.5.1 Case 1

When the car is traveling straight, both wheels travel at the same speed. Thus, the free-wheeling planet pinions do not spin at all. Instead, as the transmission shaft turns the crown wheel, the rotary motion is translated directly to the half-shafts, and both wheels spin with the angular velocity of the crown wheel (they have the same speed).

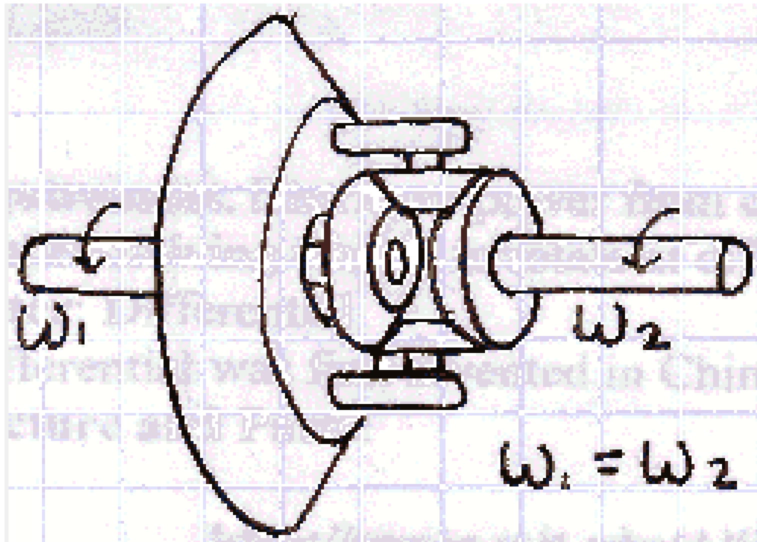


Fig.1.5 Wheels while Moving in Straight Direction

1.5.2 Case 2

When the car is turning, the wheels must move at different speeds. In this situation, the planet pinions spin with respect to the crown wheel as they turn around the sun gears. This allows the speed of the crown gear to be delivered unevenly to the two wheels.

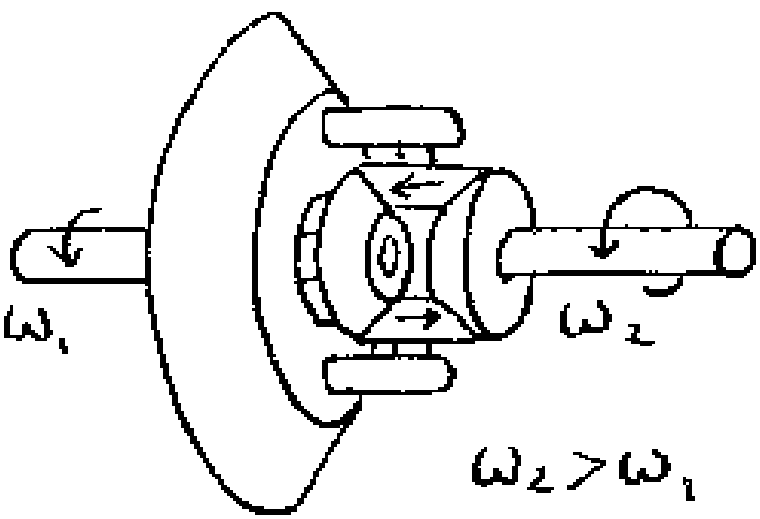


Fig.1.6 Wheels While making left Turn

1.5.3 Advantages

• Allows for completely different wheel speeds on the same axle, meaning no wheel slip will occur while going around a corner, as the outside tire will travel further.

• From an efficiency standpoint, less energy will be lost through the differential versus alternative options.

• Cost.

1.5.4 Disadvantages

The problem is an open diff always tries to balance the torque.

Almost no torque is needed to spin one wheel, and since the open differential always sends the same amount of torque to both output shafts

1.6 EFFECTS OF THE DIFFERENTIAL IN CHASSIS SET-UP

- The effect of a differential type on acceleration out of a corner

  With an open differential the lightly loaded inside wheel is free to spin up under power if enough torque is available. This limits the available acceleration. Limited slip differentials, for instance, have been used with varying success. It is unlikely that the simple addition such a differential will be successful. As with any major change, a period of development must be gone through to make the new set-up work. The type of limited slip differential chosen will be very important. Undesirable jerkiness occurs with many types and research (see [35]) has shown this to be undesirable for combined acceleration and turning. The locked rear end or spool is another common solution to the problem of wheel spin because the locked axle has high resistance to yaw, more front cornering power may be required to keep the car neutral. If all the turns are the same directions, stagger (differential tire circumference) may be used to "split the difference" between low drag when straight running and when turning.

- The effect of differential type on straight line acceleration

  If both tires are on similar coefficient pavement at similar vertical load, the differential type should not affect straight line acceleration capability (wheel spin limited). This is true of most independent suspensions and some torque tube solid axles. With solid rear axles this is not true since the wheel loads differ on acceleration. With the differential partially locked, small differences in tire size may cause the car to pull on acceleration (especially true on some front drives). This can be diagnosed by swapping the tires; if it pulls the other way, the wheels are being locked to the same speed by the differential and the tires differ in circumference.

- The effect of differential type on dropped throttle in a turn

  An open differential that distributes the torque evenly will probably have the least effect on dropped throttle behaviour. A differential that remains locked (possibly due to some preload) when throttle is dropped produces a stabilizing yawing moment or "yaw damping" moment. Some limited slip differentials may put shock loads into the drivetrain when they lock and unlock. This can have effects that are hard to predict.

- The effect of differentials on steering kickback

  This effect occurs stronger using limited slip differentials rather than open differential and just when it is in the steering wheels side, usually on the front.

  If any limited slip differential is fitted that locks-up or unlocks suddenly, it will be reflected to the steering. If there is any scrub radius, the change in drive torque will produce a torque about the kingpin which will be noticed at the steering wheel. Event with center point steering a change in engine torque will change the tire self-aligning torque and this will change the steering force in a turn. All these causes happen in an open differential but in a lower dimension. Due to the small friction inside the differential, the torque steer might be affected by the friction. Mostly in production cars whose customers should not have any steering feel. The car should become as perfect and soft as the supply can

1.7 INFLUENCING PARAMETERS

The theoretical study of the different parameters that could affect the friction inside the differential and, hence, the torque difference between left and right shaft, is listed below:

1.7.1 Gear meshing

This depends upon the tooth design, the lubricant, the speed and the resulting coefficient of friction. Using different teeth angle the losses in the meshing change. Thus, taking into account first the cylindrical gears, it can be said that the spur gears the efficiency of the gears may differ between 98 and 99% per each mesh (see [32]). This difference could be an input to maximize the efficiency. If the teeth are helical the efficiency can rise slightly, moreover the noise is reduced, but a new axial load appears. The coefficient of friction is influenced by the surface roughness, but primarily the relative radius of curvature and velocities of the surfaces, and by the thickness and mean viscosity of the oil-film. The temperature also affects to the efficiency, the overall efficiency has a maximum in an interval of temperatures. It decreases in cold or too hot conditions. In fact, it is due to the change of properties of the lubrication which is optimum in a certain interval. Regarding the bevel gears, the losses are higher than the spur gears, but it changes perpendicularly the direction of the input and output shafts.

1.7.2 Contact surfaces

It is affecting the bevel gear differential since there is a contact between the planetary and the housing itself by means of a bushing that tries to reduce the friction. This contact may be crucial since its high slippery all the time that the housing and planetary are not turning as one solid part. However, the planetary gear is not just having contact on the housing but also on the pin. When the vehicle turns, there is a relative speed between the pinion pin and the planetary so that some friction torque appears on this surface.

Moreover, a contact between the sun gears and the housing also exists. Different concepts of surfaces can be taken into account for this contact. Those could be either flat or spherical. Currently, the spherical is been used but the flat used to be used as well (actually, it might be the future differential generation design in VCC). Hence, a different concept solution for the shape is not taking into account in the model and simulation, since just the current differential M66 from VCC is modelled. Finally, another slippery surface, likely not as critical as the former contact surfaces, exists between the drive shaft and the housing. The drive shaft is holding on both the housing and the internal splines of the sun so that between a slight friction exists between the housing shoulder and the driveshaft. In fact, some modifications can be pointed out before beginning the model and the simulation, such as using a bearing between the shaft and the housing or machining the shaft to have less contact surface (although the pressure will increase). Further study will be done in the following chapters to quantify this effect.13 Bearing losses

Rolling bearings involve rolling friction (typically proportional to radial load on bearings) together with an added oil-drag loss in the lubricant within bearings, but the total is less than the tooth-friction loss. When plain bearings are used, the bearing friction greatly exceeds the tooth-friction losses. It cannot be changed since they are supplied but some different concepts of bearings may be studied to check how the friction changes, if this friction is relevant compared to the overall.

1.7.3 Lubrication

The efficiency of gears with a high sliding percentage, worm and hypoid gears, for instance, may increase up to 15 percent if synthetic oil is used instead of a mineral oil. Even in the case of spur, helical and bevel gears (which have a naturally high gear efficiency), it is possible to increase gear efficiency of up to one percent by using a synthetic gear oil. This may not seem like much at first, but it may result in considerable cost savings depending on the nominal output of the gear unit, especially in the case where several gears are deployed. The most important parameters to take into account in the used oil are the viscosity and the oil film thickness.Even though, there are different kinds of oil, this parameter cannot be modified since it is an input for the differential. The whole transmission (engine, gearbox and differential) has already been designed using a particularly kind of oil which has given the best results in optimum working conditions. Thus, the model will not include the viscosity effects in order to minimize the time consuming. Even though, the lubrication will affect the static test since the differential will not work in optimum conditions regarding oil film thickness and temperature, for instance.

1.8 Stiffness of the Housing

Finally, one of the other possible parameters to adjust in the differential itself could be the stiffness of the housing. It might contribute in the overall torque split slightly, since it also contributes in the stiffness chassis.

CHAPTER 2

LITERACTURE REVIEW

2 LITERACTURE REVIEW

1.DESIGN AND ANALYSIS OF DIFFERENTIAL GEAR BOX USED IN HEAVY VECHICLE

N. VIJAYA BABU, CH. SEKHAR

Differential is used when a vehicle takes a turn, the outer wheel on a longer radius than the inner wheel. The outer wheel turns faster than the inner wheel that is when there is a relative movement between the two rear wheels. If the two rear wheels are rigidly fixed to a rear axle the inner wheel will slip which cause rapid tire wear, steering difficulties and poor load holding. Differential is a part of inner axle housing assembly, which includes the differential rear axles, wheels and bearings. The differential consists of a system of gears arranged in such a way that connects the propeller shaft with the rear axles The analysis is conducted to verify the best material for the gears in the gear box at higher speeds by analysing stress, displacement and also by considering weight reduction. The analysis is done in Cosmos software.Modeling is done in the Pro/Engineer

2 .DESIGN AND ANALYSIS OF DIFFERENTIAL GEAR BOX

R. KARTHEIK, V. MOHAN KUMAR, S. MOHNPRABHU , V. MANOJ

Differential is used when a vehicle takes a turn, the outer wheel on a longer radius than the inner wheel. The outer wheel turns faster than the inner wheel that is when there is a relative movement between the two rear wheels. If the two rear wheels are rigidly fixed to a rear axle the inner wheel will slip which cause rapid tire wear, steering difficulties and poor load holding. Differential is a part of inner axle housing assembly, which includes the differential rear axles, wheels and bearings. The differential consists of a system of gears arranged in such a way that connects the propeller shaft with the rear axles. The main objective of this paper is to perform mechanical design of differential gear box and analysis of gears in gear box. We have taken Stainless steel, aluminium alloy, magnesium alloy, structural steel materials for conducting the analysis. Presently used materials for gears and gears shafts is Cast Iron, Cast Steel. So, in this paper we are checking as the other material for the differential gear box for light utility vehicles so, we can reduce the weight

3.DESIGN AND ANALYSIS OF OPEN DIFFERENTIAL

SUBHAJIT KONAR, VIJAY GAUTAM

. An automobile differential gear system is used to establish a differential motion between left and right driving axle which provides a smooth turning of the vehicle. When a vehicle takes a turn then the wheels at outermost position requires to cover a large distance than that of the innermost wheels. This speed variation can be achieved by using a differential gear system. It also transmits the power from the propeller shaft to each axle. A rear wheel drive vehicle requires a differential at the rear axle while all-wheel drive vehicle requires differential gear system for each and every axle. In this paper, an open differential is designed for a leading automobile and analysed the ability to work without failure. The analysis was done using the modified Lewis equation, Hertzian contact stress equation and AGMA equations. The analytical results were compared with results obtained by FEA. It is observed that results obtained from Lewis criterion are more conservative as compared to AGMA. The results obtained from modified Lewis, Hertzian contact stress and AGMA are in good agreement with the results obtained from FEA

4. DESIGN AND ANALYSIS OF DIFFERENTIAL GEAR BOX IN AUTOMOBILES

K. DINESH BABU, M. SIVA NAGENDRA,CH. PHANIDEEP , J.SAI TRINADH

The main objective of this paper is to perform mechanical design of differential gear box and analysis of gears in gear box. We have taken grey cast iron and aluminium alloy materials for conducting the analysis. Presently used materials for gears and gears shafts is Cast Iron, Cast Steel. So, in this paper we are checking as the aluminium can be the other material for the differential gear box for light utility vehicles so, we can reduce the weight.

Key words: Differential Gear Box, Structural Analysis, Design and Structural Analysis of

Differential Gear Box.

5. QUALITY ANALYSIS OF REAR AXLE ASSEMBLY BASED ON AHP

YUN – RUI WANG, JUAN LI AND HONG –YU GE

Product quality is an important factor for the survival and development of enterprise. Under the premise of ensuring the quality of design, the assembly process also plays a crucial role in the formation of product quality.

Through field investigation and analysis, combining with the current situation of product quality of an axle factory, using the pareto chart, researchers found quality fluctuation was mainly caused by abnormal sound at rear axle and rough surface. According to the principle and step of AHP, a three-level evaluation system was established. Combining with the actual assembly process, it was calculated that the influence of assemble rear wheel hub and brake hub were the greatest. This method changed the status of the enterprises' experience as quality judgment method, and provided a reliable theoretical basis for the implementation of modern quality management.

CHAPTER 3

GEOMETRIC MODELLING

3 GEOMETRIC MODELLING

3.1 SOLIDWORKS

SolidWorks is a modern computer aided design (CAD) program. It enables designers to create a mathematically correct solid model of an object that can be stored in a database. When the mathematical model of a part or assembly is associated with the properties of the materials used, we get a solid model that can be used to simulate and predict the behaviour of the part or model with finite element and other simulation software. The same solid model can be used to manufacture the object and also contains the information necessary to inspect and assemble the product. The marketing organization can produce sales brochures and videos that introduce the product to potential customers. SolidWorks and similar CAD programs have made possible concurrent engineering, where all the groups that contribute to the product development process can share information on real-time.

3.2 STAGES IN THE PROCESS

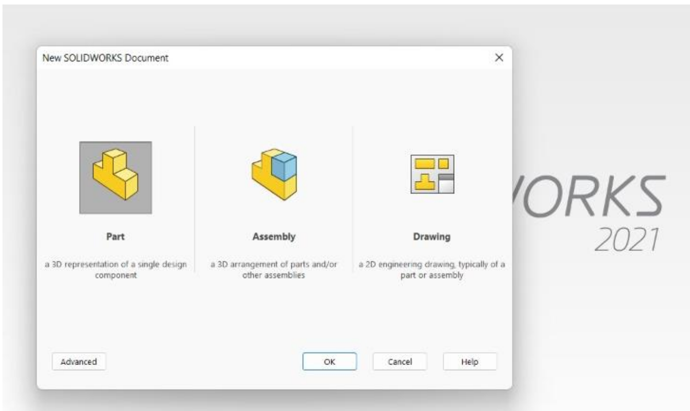


Fig. 3.1 Stages in the Process

- **Create a new part document**

  New parts can be created in inch, millimetre or other units. Parts are used to create and hold the solid model.

- **Sketch the profile**

  Sketches are collections of 2D geometry that are used to create solid features. These include lines, circles and rectangles.

- **Applying Sketch relations and dimensions**

  Geometric relationships such as horizontal and vertical are applied to the sketch geometry. Dimension size the geometry while the relations restrict the movement of the entities.

- **Extruding the sketch**

  Extruding uses the 2D sketch to create a 3D solid feature

3.2.1 Pinion Gear

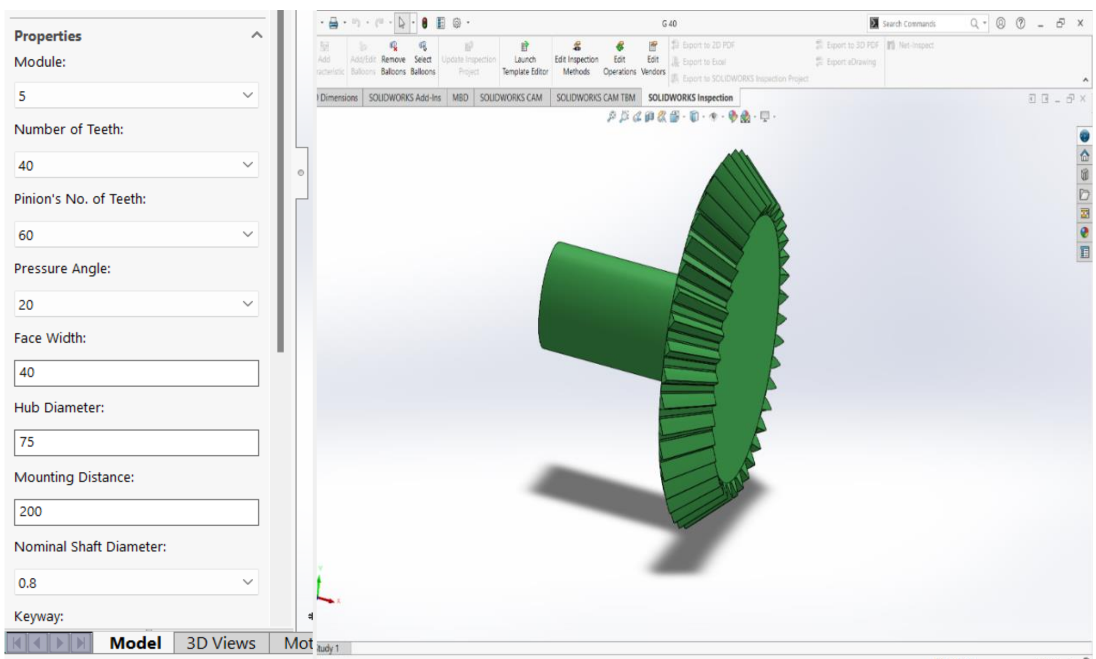


Fig3.2 Part Drawing of Pinion Gear

3.2.2 Crown Wheel

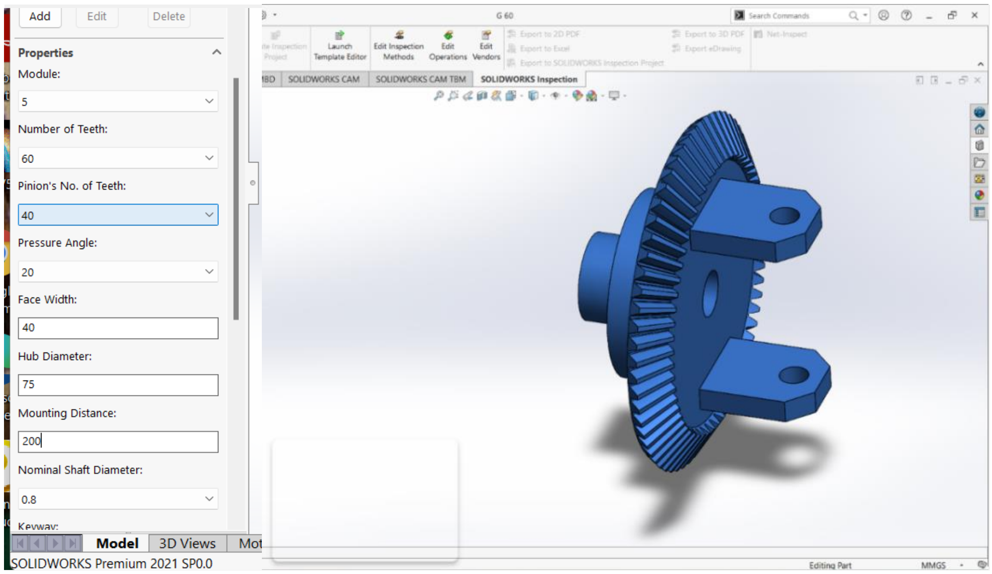


Fig. 3.3 Part Drawing of Crown Wheel

3.2.3 Side Gear

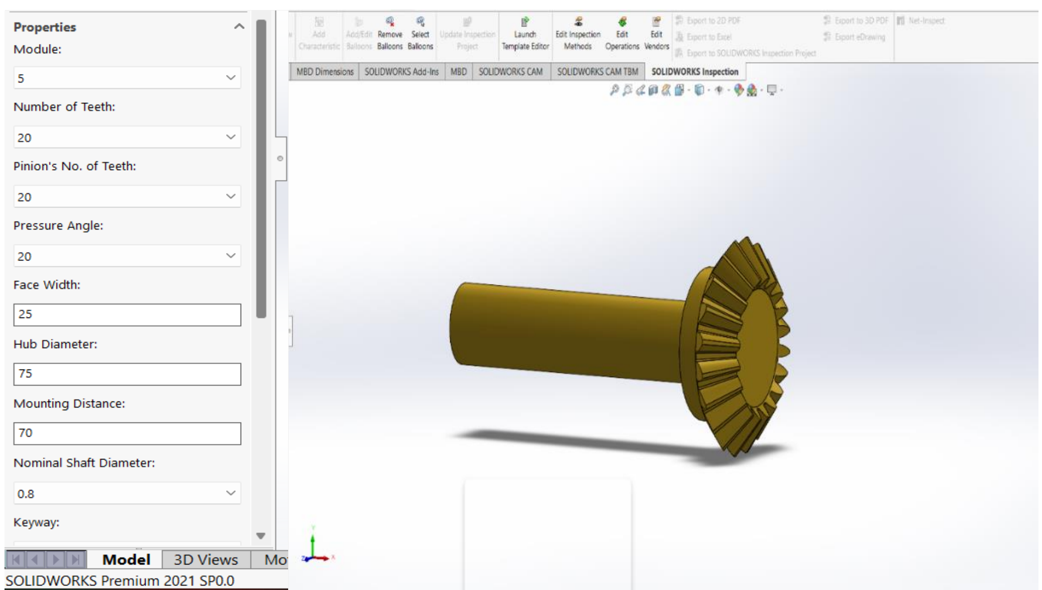


Fig 3.4 Part Drawing of Side Gear

3.2.4 Spider Gear

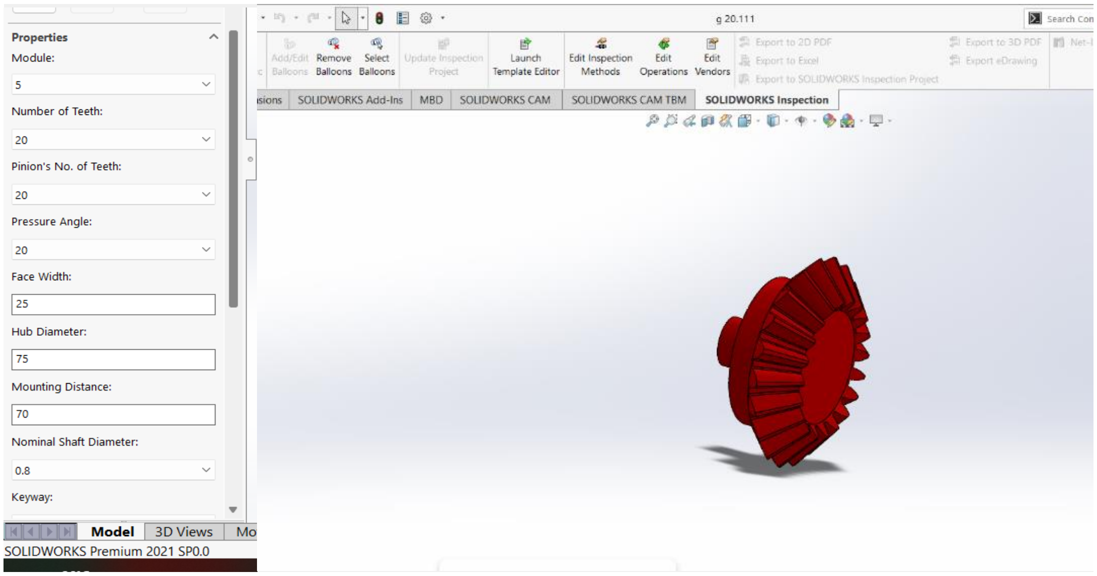


Fig 3.5 Part Drawing of Spider Gear

3.3 ASSEMBLY

3.3.1 Differential

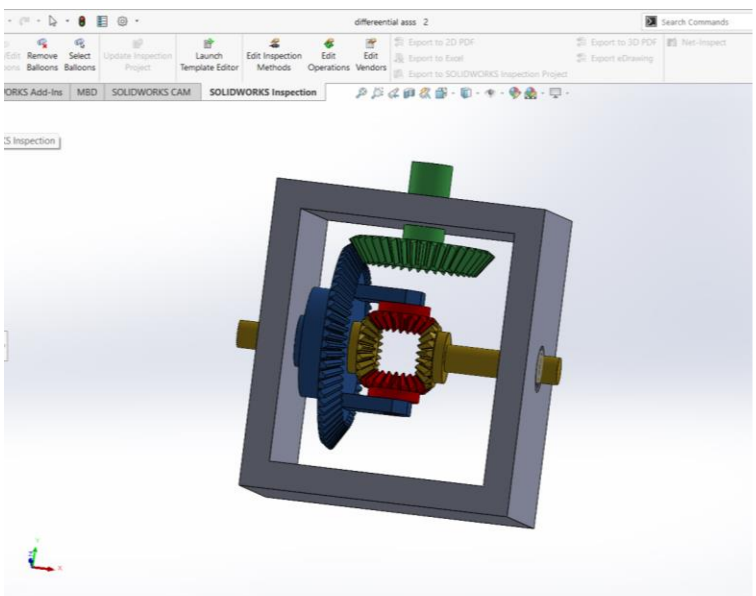


Fig 3.6 Assembly of Differential

3.4 CREATING DRAWINGS

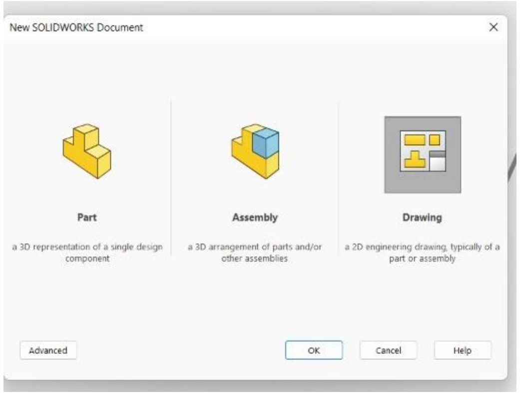


Fig 3.7 Selection of Drawing in SolidWorks

You can generate drawings in SolidWorks the same way you would generate them in 2D drafting and drawing systems. However, creating 3D models and generating drawings from the model have many advantages; for example:

- Designing models is faster than drawing lines.
- SolidWorks creates drawings from models, so the process is efficient.
- You can review models in 3D and check for correct geometry and design issues before generating drawings, so the drawings are more likely to be free of design errors.
- You can insert dimensions and annotations from model sketches and features into drawings automatically, so you do not have to create them manually in drawings.
- Parameters and relations of models are retained in drawings, so drawings reflect the design intent of the model.
- Changes in models or in drawings are reflected in their related documents, so making changes is easier and drawings are more accurate.

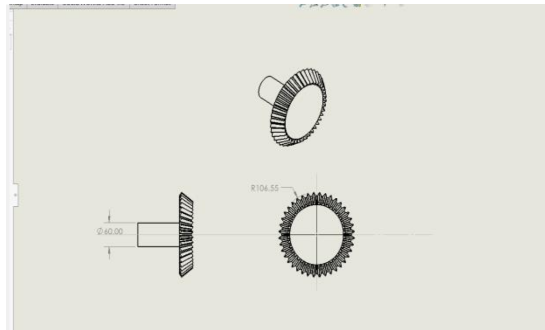


Fig 3.8 Pinion Gear Drawing

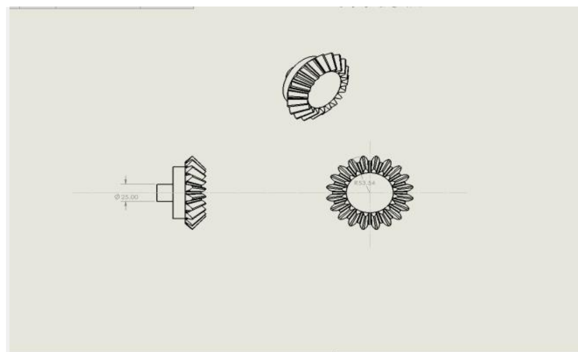
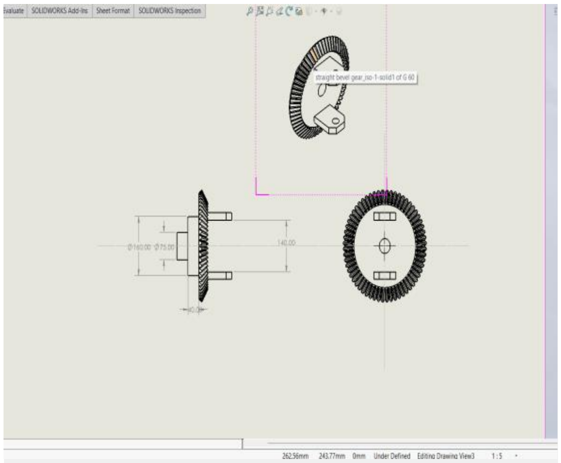


Fig 3.9 Spider Gear Drawing

3.4.1 Creating Drawings from Models

To generate drawings from part and assembly documents:

1. In a part or assembly document, click Make Drawing from Part/Assembly on the Standard toolbar and select a template in the Sheet Format/Size dialog box. The View Palette opens on the right side of the window.
2. Click to pin the View Palette
3. Drag a view from the View Palette onto the drawing sheet.
4. In the Drawing View Property Manager, set options such as orientation, display style, scale, etc. then click.
5. Repeat steps 3 and 4 to add views. Note: You can have any drawing views of any models in a given drawing document.



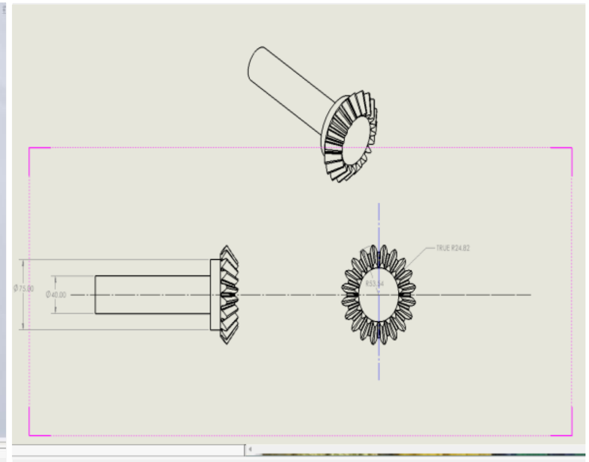


Fig 3.10 Crown Wheel Drawing **Fig 3.11 Side Gear Drawing**

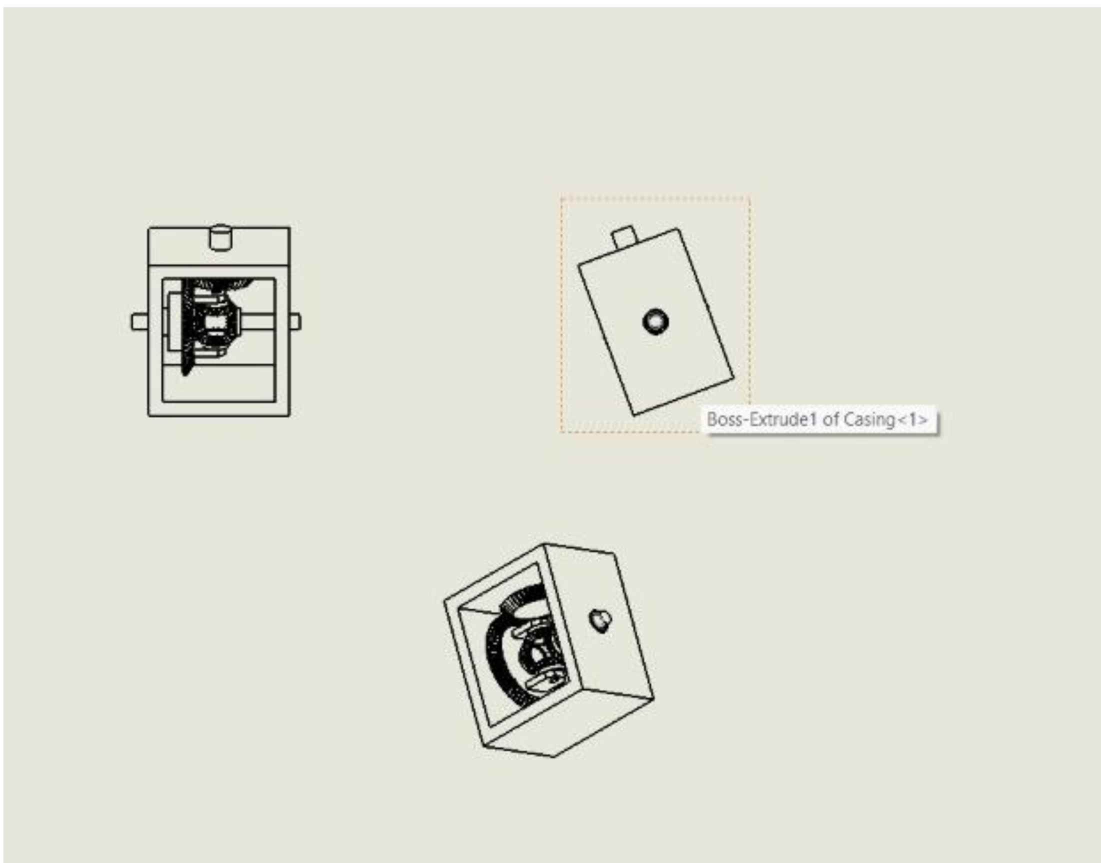


Fig 3.12 Differential Drawing

CHAPTER 4

DESIGN CALCULATIONS

4 DESIGN CALCULATIONS

4.1 SPECIFICATIONS

- Input power (kw) = 5
- Maximum velocity (V_{max}) = 114 kmph
- Input torque = 17 kg- m
- Number of teeth on gear (Z_g) = 60
- Number of teeth on pinion (Z_p) =40
- Pressure angle (α) =20
- Module (m) = 5.32 mm

Diameter of Pinion and Gear

$$D_P = m * Z_p$$

= 213 mm

$$D_g = m * Z_g$$

=319.5 mm

Velocity Ratio (Vr)

$$VR = \frac{Z_g}{Z_p}$$ =60/40 =1.5

Pitch Cone Distance(AO)

$$AO = \sqrt{\left(\frac{D_g}{2}\right)^2 + \left(\frac{D_p}{2}\right)^2}$$

$$= \sqrt{(319.5/2)^2 + (213/2)^2}$$

= 191.995 mm

Face Width(b)

b =1/3*(AO)

= 1/3*(191.277)

= 63.992 mm

Pitch Cone Angle(ϒ)

$\text{Tan } ϒ = \frac{Z_g}{Z_p}$

$ϒ = \tan^{-1}\frac{Z_g}{Z_p}$

$= \tan^{-1}\frac{40}{60} = 33.69^0$

Back Cone Angle (ζ)

ζ = π/2 – ϒ

= 90-33.69 = 56.31^0

Equivalent Teeth on Pinion(Z_{EP})

Z_{EP} = Z_P/cos(ϒ)

= 40/COS (33.69)

= 48.073

Equivalent Teeth on Gear(Z_{EG})

Z_{EG} = Z_G /cos(ζ)

= 60/cos(56.31)

= 108.138

Pitch Circle Diameter (D) :

D = D_G + D_P = 319.5+213 = 532.5 mm

Actual N0. of Teeth on Bevel Gear (Z)

Z = Z_G+ Z_P = 60+40= 100

Module at Large End of Teeth (m)

m = D/Z = 532.5/100= 5.3 mm

Given

$V_{max} = R_{min} * \omega$

$= R_P * \omega_p$

114*5/18= $(100*10^{-3}) * \omega_p$

ω_p = 316.6 rad/s

$\omega_p = 2\pi N_P/60$

N_P = 2839.383 RPM

∴Velocity Ratio (VR) = $\dfrac{N_P}{N_G}$

N_G = 2839.383/1.5

1892.9 RPM

Speed of Pinion (N_P) =2839.383 RPM

Speed of Gear (N_G) =1892.922 RPM

Force Analysis of Bevel Gear

Rated Torque (M_t)= $\dfrac{60*10^6*kw}{2\pi N_p}$

$=\dfrac{60*10^6*5}{2\pi*3023.975}$

= 16816.332N-mm

Mean Radius (r_m) = $\dfrac{D_p}{2} - \dfrac{b\sin\gamma}{2}$

$= \dfrac{213}{2} - \dfrac{60.09*\sin 33.69}{2}$

=88.75mm

Tangential Component of Force (P_t)

$$P_t = \frac{M_t}{r_m} = 16816.32/88.33$$

P_t= 189.479 N

Radial Component of Force (P_r)

$P_r = P_t * \tan\alpha * \cos \Upsilon$

$P_r = 189.479 * \tan 20 * \cos 33.69$

P_r = 57.382 N

Axial Component of Force (P_a)

$P_a = P_t * \tan\alpha * \sin \Upsilon$

$P_a = 189.479 * \tan 20 * \sin 33.69$

P_a = 38.254 N

4.2 GREY CAST ITON

Beam Strength of Bevel Gear (S_b)

$$S_b = m * \sigma_b * b * y * [1 - \frac{b}{AO}]$$

σ_b = permissible bending stress = (S_{ut}/3) N/mm^2

Gray Cast Iron

Density	7.2e-06 kg/mm³

Structural

Isotropic Elasticity

Derive from	Young's Modulus and Poisson's Ratio
Young's Modulus	1.1e+05 MPa
Poisson's Ratio	0.28
Bulk Modulus	83333 MPa
Shear Modulus	42969 MPa
Isotropic Secant Coefficient of Thermal Expansion	1.1e-05 1/°C
Compressive Ultimate Strength	820 MPa
Compressive Yield Strength	0 MPa
Tensile Ultimate Strength	240 MPa
Tensile Yield Strength	0 MPa

Thermal

Isotropic Thermal Conductivity	0.052 W/mm·°C
Specific Heat Constant Pressure	4.47e+05 mJ/kg·°C

Electric

Isotropic Resistivity	9.6e-05 ohm·mm

Magnetic

Isotropic Relative Permeability	10000

Fig.4.1 Properties of grey cast iron

Y = lewis form factor

AO = Pitch cone distance (mm)

Module (m) = 5 mm

Equivalent teeth on pinion (z_{ep})

b = face width

$Y = \pi * y$

$y = 0.154 - \frac{0.912}{Zep}$

y =0.135

Y = π*0.135 = 0.424

S_{ut}= Ultimate tensile strength of grey cast iron = 295 N /mm^2

$S_b = 5.3*\frac{295}{3}*63.992 * 0.424* [1 - \frac{1}{3}]$

S_b = 9472.472N

∴Beam strength of bevel gear (s_b) =9472.472N

Wear Strength of Bevel Gear (S_w)

$$S_w = b * Q * d_p^{|} * k^{`}$$

b = face width

Q = Ratio factor

$$Q = \frac{2 * Z_G^{|}}{Z_G^{|} + Z_P^{|}}$$

$$Z_g^{|} = \frac{Z_g}{\sin\gamma}, \quad Z_P^{|} = \frac{Z_P}{\cos\gamma}$$

$d_p^{`}$= pitch circlr diameter of formative pinion(mm)

= D_P / COS ϒ

= 200/cos 33.69 = 240.369 mm

$$Z_P^{|} = \frac{Z_P}{\cos\gamma} = \frac{40}{cos 33.69} = 48.073$$

$$Z_g^{|} = \frac{Z_g}{\sin\gamma}, \quad = \frac{60}{sin 33.69} = 108.138$$

$$Q = \frac{2 * Z_G^{|}}{Z_G^{|} + Z_P^{|}} = \frac{2*108.138}{108.138+48.073} = 1.3846$$

K = Material constant (N/mm^2)

For 20^0 pressure angle

$$K = 0.16 * \left(\frac{BHN}{100}\right)^2$$ (BHN of grey cast iron is 235)

= 0.16*[235/100]^2

K = 0.8839 N/mm^2

S_W = 60.092*1.3846*255.9939*0.8836

= 20043.544N

∴wear strength of bevel gear (s_W) = 20043.544N

Effective Load on Gear Tooth

Rated Torque (M_t) = $\frac{60*10^6*kw}{2\pi N_p}$

$=\frac{60*10^6*5}{2\pi*2839.383}$

= 16816.332N-mm

Tangential Component Due to Power Transmission (P_t) :

$P_t = 2*\frac{M_t}{D_p}$ = 2*15789.311/200

= 157.893N

Velocity of Gear Tooth (v) :

$$v = \left(\frac{D_p * N_p * \pi}{60*10^3}\right)$$

$v = \frac{\pi*200*3023.975}{60*10\text{^}3}$

= 31.67

Service Factor (C_s) = 1.5 (by given properties **)**

Velocity Factor (C_v) =

$C_v = \frac{5.6}{5.6+\sqrt{v}}$

$C_v = \frac{5.6}{5.6+\sqrt{31.67}}$

= 0.498 (for generated tooth)

Effective Load Between Two Meshing Teeth

$$P_{eff} = \frac{C_S}{C_V} * \mathrm{P}_t$$

$P_{eff} = \frac{1.5}{0.498} * 157.893$

P_{eff} =475.581 N

Tolerance Factor(φ_p)

For pinion $\varphi_p = m + 0.25\sqrt{d_p^|}$

φ_p= 5+0.25$\sqrt{213}$

φ_p=8.973

for gear $\varphi_g = m + 0.25\sqrt{d_g^|}$

φ_g= 5+0.25 $\sqrt{319.5}$

φ_g=9.79

Errors of Pinion And Gear (e_p&e_g)

GRADE 6

e_p= 8+ 0.63* φ_p

e_p= 8+ 0.63* 8.55

e_p=13.3865 µm

e_g= 8+ 0.63* φ_g

e_g= 8+ 0.63*9.33

e_g=14.164µm

Sum of Errors Between Meshing Teeth (e)

e = e_p+ e_g

=13.3865+ 14.16479

e = 27.8224 µm

e =0.027822 mm

Deformation Factor(C)

C =5700 N/mm^2 (from properties)

Dynamic Load on Bevel Gear(P_d)

$$P_b = \frac{21 * \nu * Ceb + P_t}{21 * \nu + \left(Ceb + P_t\right)^{0.5}}$$

$P_d = \frac{21*31.67*(9338.722+157.8)}{21*31.67+(9338.722+157.8)^{\wedge}0.5}$

P_d =9197.925N

Effective Load(Peff)

Peff = Cs* Pt+ Pd

Peff = 1.5* 157.8+9197.925

Peff = 9434.773 N

Avoid Failure Gear Tooth Due to Bending

$$S_b = P_{eff} * f.s$$

f.s = Sb/ Peff

=9472.472/9434.773

f.s = 1.003

Avoid Failure Gear Tooth Due to Pitching

$$S_w = P_{eff} * f.s$$

f.s = S_w/P_{eff}

= 20043.544/9434.773

f.s = 2.124

DESIGN IS SAFE

4.3 TITANIUM ALLOY (Ti64)

Beam Strength of Bevel Gear (S_b)

Titanium Alloy

Density	4.62e-06 kg/mm³
Structural	
Isotropic Elasticity	
Derive from	Young's Modulus and Poisson's Ratio
Young's Modulus	96000 MPa
Poisson's Ratio	0.36
Bulk Modulus	1.1429e+05 MPa
Shear Modulus	35294 MPa
Isotropic Secant Coefficient of Thermal Expansion	9.4e-06 1/°C
Compressive Ultimate Strength	0 MPa
Compressive Yield Strength	930 MPa
Tensile Ultimate Strength	1070 MPa
Tensile Yield Strength	930 MPa
Thermal	
Isotropic Thermal Conductivity	0.0219 W/mm·°C
Specific Heat Constant Pressure	5.22e+05 mJ/kg·°C
Electric	
Isotropic Resistivity	0.0017 ohm·mm
Magnetic	
Isotropic Relative Permeability	1

Fig.4.2 Properties of Titanium Alloy

$$S_b = m * \sigma_b * b * y * [1 - \frac{b}{AO}]$$

σ_b = permissible bending stress = ($S_{ut}/3$) N/mm^2

Y = lewis form factor

AO = Pitch cone distance (mm)

Module (m) = 5.33 mm

Equivalent teeth on pinion (z_{ep})

b = face width

$$Y = \pi * y$$

$y = 0.154 - \frac{0.912}{Zep}$

y =0.135

Y = π*0.135 = 0.424

S_{ut}= Ultimate tensile strength of titanium alloy = 1170 N /mm^2

$S_b = 5.32 * \frac{1170}{3} * 63.998 * 0.424 * [1 - \frac{1}{3}]$

S_b = 37568.61N

∴Beam strength of bevel gear (s_b) =37568.6N

Wear Strength of Bevel Gear (S_w)

$$S_w = b * Q * d_p^{|} * k^{`}$$

b = face width

Q = Ratio factor

$$Q = \frac{2 * Z_G^{|}}{Z_G^{|} + Z_P^{|}}$$

$$Z_g^{|} = \frac{Z_g}{\sin\gamma}, \quad Z_P^{|} = \frac{Z_P}{\cos\gamma}$$

$d_p^{`}$= pitch circlr diameter of formative pinion(mm)

= D_P / COS ϒ

= 200/cos 33.69 = 240.369 mm

$$Z_P^{|} = \frac{Z_P}{\cos\gamma} = \frac{40}{cos 33.69} = 48.073$$

$$Z_g^{|} = \frac{Z_g}{\sin\gamma}, = \frac{60}{sin 33.69} = 108.138$$

$$Q = \frac{2 * Z_G^{|}}{Z_G^{|} + Z_P^{|}} = \frac{2*108.138}{108.138+48.073} = 1.3846$$

K = Material constant (N/mm^2)

For 20^0 pressure angle

$K = 0.16*\left(\frac{BHN}{100}\right)^2$ (BHN of titanium alloy is 390)

$= 0.16*[390/100]^2$

$K = 2.4336\ N/mm^2$

$S_w = 63.9982*1.3846*285.993*2.4336$

= 55203.678N

∴wear strength of bevel gear (s_w) = 55203.678N

Effective Load on Gear Tooth

Rated Torque (M_t) $= \frac{60*10^6*kw}{2\pi N_p}$

$= \frac{60*10^6*5}{2\pi*3023.975}$

= 15789.311 N-mm

Tangential Component Due to Power Transmission (P_t)

$P_t = 2*\frac{M_t}{D_p}$ = 2*15789.311/200

= 157.893N

Velocity of Gear Tooth (ν)

$$\nu = \left(\frac{D_p*N_p*\pi}{60*10^3}\right)$$

$\nu = \frac{\pi*200*3023.975}{60*10^\wedge 3}$

= 31.67

Service Factor (C_s) = 1.5 (by given properties)

Velocity Factor(C_v) =

$C_v = \frac{5.6}{5.6+\sqrt{\nu}}$

$C_v = \frac{5.6}{5.6+\sqrt{31.67}}$

= 0.498 (for generated tooth)

Effective Load Between Two Meshing Teeth

$$P_{eff} = \frac{C_S}{C_V} * P_t$$

$P_{eff} = \frac{1.5}{0.498} * 157.893$

P_{eff} =475.581 N

Tolerance Factor(φ_p)

For pinion $\varphi_p = m + 0.25\sqrt{d_p^{|}}$

φ_p= 5+0.25$\sqrt{200}$

φ_p=8.535

for gear $\varphi_g = m + 0.25\sqrt{d_g^{|}}$

φ_g= 5+0.25 $\sqrt{300}$

φ_g=9.33

Errors of Pinion and Gear (e_p&e_g)

GRADE 6

e_p= 8+ 0.63* φ_p

e_p= 8+ 0.63* 8.55

e_p=13.3865 µm

e_g= 8+ 0.63* φ_g

e_g= 8+ 0.63*9.33

e_g=13.8779µm

Sum of Errors Between Meshing Teeth (e)

e = e_p+ e_g

=13.3865+ 13.8779

e = 27.2644 µm

e =0.0272644 mm

Deformation Factor(C)

C =9542N/mm^2 (from properties)

Dynamic Load on Bevel Gear(P_d)

$$P_b = \frac{21*\nu*Ceb+P_t}{21*\nu+\left(Ceb+P_t\right)^{0.5}}$$

$$P_d = \frac{21*31.67*(191540.65)}{21*31.67+(191540.65)^{\wedge}0.5}$$

P_d =14326.98N

Effective Load(Peff)

Peff = Cs* Pt+ Pd

Peff = 1.5* 157.8+14326.98

Peff = 14563.815

Avoid Failure Gear Tooth Due to Bending

$$S_b = P_{eff} * f.s$$

f.s = Sb/ Peff

= 37568.62/14563.815

f.s = 2.27

Avoid Failure Gear Tooth Due to Pitching

$$S_w = P_{eff} * f.s$$

f.s =Sw/P eff

= 55203.678/14563.815

f.s = 3.341

DESIGN IS SAFE

4.4 STRUCTURAL STEEL

Structural Steel

Fatigue Data at zero mean stress comes from 1998 ASME BPV Code, Section 8, Div 2, Table 5-110.1

Density	7.85e-06 kg/mm³

Structural

Isotropic Elasticity

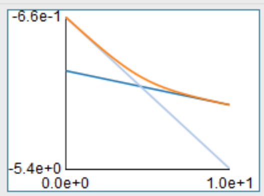
Derive from	Young's Modulus and Poisson's Ratio
Young's Modulus	2e+05 MPa
Poisson's Ratio	0.3
Bulk Modulus	1.6667e+05 MPa
Shear Modulus	76923 MPa
Isotropic Secant Coefficient of Thermal Expansion	1.2e-05 1/°C
Compressive Ultimate Strength	0 MPa
Compressive Yield Strength	250 MPa
Strain-Life Parameters	

Fig.4.3 Properties of Structural steel

Beam Strength of Bevel Gear (S_b)

$$S_b = m * \sigma_b * b * y * [1 - \frac{b}{AO}]$$

σ_b = permissible bending stress = ($S_{ut}/3$) N/mm²

Y = lewis form factor

AO = Pitch cone distance (mm)

Module (m) = 5 .3mm

Equivalent teeth on pinion (z_{ep})

b = face width

$$Y = \pi * y$$

y = 0.154-$\frac{0.912}{Zep}$

y =0.135

Y = π*0.135 = 0.424

S_{ut}= Ultimate tensile strength of grey cast iron = 360 N /mm^2

$S_b = 5.32*\frac{360}{3}*63.392 * 0.424* [1-\frac{1}{3}]$

S_b = 11496.56N

∴Beam strength of bevel gear (s_b) =11496.56N

Wear Strength of Bevel Gear (S_w)

$$S_w = \text{b}*Q*d_p^{|}*k^{`}$$

b = face width

Q = Ratio factor

$$Q = \frac{2*Z_G^{|}}{Z_G^{|}+Z_P^{|}}$$

$$Z_g^{|} = \frac{Z_g}{\sin\gamma}, \quad Z_P^{|} = \frac{Z_P}{\cos\gamma}$$

d_p`= pitch circlr diameter of formative pinion (mm)

= D_P / COS ϒ

= 200/cos 33.69 = 240.369 mm

$$Z_P^{|} = \frac{Z_P}{\cos\gamma} \quad = \frac{40}{cos 33.69} \quad = 48.073$$

$$Z_g^{|} = \frac{Z_g}{\sin\gamma}, \quad = \frac{60}{sin 33.69} \quad = 108.138$$

$$Q = \frac{2*Z_G^{|}}{Z_G^{|}+Z_P^{|}} = \frac{2*108.138}{108.138+48.073} = 1.3846$$

K = Material constant (N/mm^2)

For 20^0 pressure angle

$K = 0.16*\left(\frac{BHN}{100}\right)^2$ (BHN of grey cast iron is 235)

= 0.16*[235/100]^2

K = 0.8839 N/mm^2

S_w = 63.392*1.3846*240.369*0.8836

= 18633.94

∴wear strength of bevel gear (s_w) = 18633.9406N

Effective Load on Gear Tooth

Rated Torque (M_t) = $\frac{60*10^6*kw}{2\pi N_p}$

$=\frac{60*10^6*5}{2\pi*3023.975}$

= 15789.311 N-mm

Tangential Component Due to Power Transmission (P_t)

$P_t = 2*\frac{M_t}{D_p}$ = 2*15789.311/200

= 157.893N

Velocity of Gear Tooth (v)

$$v = \left(\frac{D_p*N_p*\pi}{60*10^3}\right)$$

$v = \frac{\pi*200*3023.975}{60*10\text{^}3}$

= 31.67

Service Factor (C_s) = 1.5 (by given properties)

Velocity Factor(C_v) =

$C_v = \frac{5.6}{5.6+\sqrt{v}}$

$C_v = \frac{5.6}{5.6+\sqrt{31.67}}$

= 0.498 (for generated too..th)

Effective Load Between Two Meshing Teeth

$$P_{eff} = \frac{C_S}{C_V} * P_t$$

$P_{eff} = \frac{1.5}{0.498} * 157.893$

P_{eff} =475.581 N

Tolerance Factor(φ_p)

For pinion $\varphi_p = m + 0.25\sqrt{d_p^|}$

φ_p= 5+0.25$\sqrt{200}$

φ_p=8.535

for gear $\varphi_g = m + 0.25\sqrt{d_g^|}$

φ_g= 5+0.25 $\sqrt{300}$

φ_g=9.33

Errors of Pinion And Gear (e_p&e_g)

GRADE 6

e_p= 8+ 0.63* φ_p

e_p= 8+ 0.63* 8.55

e_p=13.3865 µm

e_g= 8+ 0.63* φ_g

e_g= 8+ 0.63*9.33

e_g=13.8779µm

Sum of Errors Between Meshing Teeth (e)

e = e_p+ e_g

=13.3865+ 13.8779

e = 27.2644 μm

e =0.0272644 mm

Deformation Factor(C)

C =11200N/mm^2 (from properties)

Dynamic Load on Bevel Gear(P_d)

$$P_b = \frac{21 * \nu * Ceb + P_t}{21 * \nu + \left(Ceb + P_t\right)^{0.5}}$$

$$P_d = \frac{21*31.67*(201278.131)}{21*31.67+(201278.138)^{0.5}}$$

P_d =12019.41N

Effective Load(Peff)

Peff = Cs* Pt+ Pd

Peff = 1.5* 157.8+12019.41

Peff = 12346.245

Avoid Failure Gear Tooth Due to Bending

$$S_b = P_{eff} * f.s$$

f.s = Sb/ Peff

= 11496/123456

f.s = 0.93

Avoid Failure Gear Tooth Due to Pitching

$$S_w = P_{eff} * f.s$$

f.s =Sw/P eff

= 18633/12346

f.s = 1.51

DESIGN IS NOT SAFE ∴

4.5 ALUMINIUM ALLOY (Al 6160)

Aluminum Alloy

General aluminum alloy. Fatigue properties come from MIL-HDBK-5H, page 3-277.

Density	2.77e-06 kg/mm^3

Structural

Isotropic Elasticity

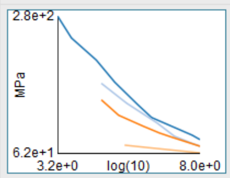
Derive from	Young's Modulus and Poisson's Ratio
Young's Modulus	71000 MPa
Poisson's Ratio	0.33
Bulk Modulus	69608 MPa
Shear Modulus	26692 MPa
Isotropic Secant Coefficient of Thermal Expansion	2.3e-05 1/°C
Compressive Ultimate Strength	0 MPa
Compressive Yield Strength	280 MPa
S-N Curve	
Tensile Ultimate Strength	310 MPa
Tensile Yield Strength	280 MPa

Fig. 4.4.Properties of Aluminium Alloyc

Beam Strength of Bevel Gear (S_b) :

$$S_b = m * \sigma_b * b * y * [1 - \frac{b}{AO}]$$

σ_b = permissible bending stress = ($S_{ut}/3$) N/mm^2

Y = lewis form factor

AO = Pitch cone distance (mm)

Module (m) = 5 mm

Equivalent teeth on pinion (z_{ep})

b = face width

$$Y = \pi * y$$

y = 0.154-$\frac{0.912}{\text{Zep}}$

y =0.135

Y = π*0.135 = 0.424

S_{ut}= Ultimate tensile strength of AL ALLOY = 1200 N /mm^2

S_b = 5.3*1200/3*63.992 * 0.424* [1-$\frac{1}{3}$]

S_b = 3851.91N

∴Beam strength of bevel gear (s_b) =3851.91N

Wear Strength of Bevel Gear (S_w)

$$S_w = b * Q * d_p^{|} * k^{`}$$

b = face width

Q = Ratio factor

$$Q = \frac{2 * Z_G^{|}}{Z_G^{|} + Z_P^{|}}$$

$$Z_g^{|} = \frac{Z_g}{\sin\gamma}, \quad Z_P^{|} = \frac{Z_P}{\cos\gamma}$$

d_p`= pitch circlr diameter of formative pinion (mm)

= D_P / COS ϒ

= 200/cos 33.69 = 240.369 mm

$$Z_P^{|} = \frac{Z_P}{\cos\gamma} = \frac{40}{cos 33.69} = 48.073$$

$$Z_g^{|} = \frac{Z_g}{\sin\gamma}, \quad = \frac{60}{sin 33.69} = 108.138$$

$$Q = \frac{2 * Z_G^{|}}{Z_G^{|} + Z_P^{|}} = \frac{2*108.138}{108.138+48.073} = 1.3846$$

K = Material constant (N/mm^2)

For 20^0 pressure angle

$K = 0.16*\left(\frac{BHN}{100}\right)^2$ (BHN of AL ALLOY is 95)

= 0.16*[95/100]^2

K = 0.144 N/mm^2

S_w = 63.9982*1.3846*181.7*0.1444

= 4121675.024N

∴wear strength of bevel gear (s_w) = 4121675.024N

Effective Load on Gear Tooth

Rated Torque (M_t) = $\frac{60*10^6*kw}{2\pi N_p}$

$=\frac{60*10^6*5}{2\pi*3023.975}$

= 15789.311 N-mm

Tangential Component Due to Power Transmission (P_t)

$P_t = 2*\frac{M_t}{D_p}$ = 2*15789.311/200

= 157.893N

Velocity of Gear Tooth (ν) :

$\nu = \left(\frac{D_p*N_p*\pi}{60*10^3}\right)$

$\nu = \frac{\pi*200*3023.975}{60*10^{\wedge}3}$

= 31.67

Service Factor (C_s) = 1.5 (by given properties **)**

Velocity Factor(C_v) =

$C_v = \frac{5.6}{5.6+\sqrt{\nu}}$

$C_v = \frac{5.6}{5.6+\sqrt{31.67}}$

= 0.498 (for generated tooth)

Effective Load Between Two Meshing Teeth

$$P_{eff} = \frac{C_S}{C_V} * \text{ P}_t$$

$P_{eff} = \frac{1.5}{0.498} * 157.893$

P_{eff} =475.581 N

Tolerance Factor(φ_p)

For pinion $\varphi_p = m + 0.25\sqrt{d_p^{|}}$

φ_p= 5+0.25$\sqrt{200}$

φ_p=8.535

for gear $\varphi_g = m + 0.25\sqrt{d_g^{|}}$

φ_g= 5+0.25 $\sqrt{300}$

φ_g=9.33

Errors of Pinion And Gear (e_p&e_g)

GRADE 6

e_p= 8+ 0.63* φ_p

e_p= 8+ 0.63* 8.55

e_p=13.3865 µm

e_g= 8+ 0.63* φ_g

e_g= 8+ 0.63*9.33

e_g=13.8779µm

Sum of Errors Between Meshing Teeth (e)

$e = e_p + e_g$

$=13.3865 + 13.8779$

$e = 27.2644\ \mu m$

$e = 0.0272644$ mm

Deformation Factor(C)

C =4968N/mm^2 (from properties)

Dynamic Load on Bevel Gear (P_d)

$$P_b = \frac{21 * v * Ceb + P_t}{21 * v + \left(Ceb + P_t\right)^{0.5}}$$

$$P_d = \frac{21*31.67*(8437.94+157.8)}{21*31.67+(8437.97+157.8)^{\wedge}0.5}$$

P_d =592039.26

Effective Load (Peff)

Peff = Cs* Pt+ Pd

Peff = 1.5* 157.8+592039.26

Peff = 888216.79N

Avoid Failure Gear Tooth Due to Bending

$$S_b = P_{eff} * f.s$$

f.s = Sb/ Peff

= 38531.91/888216.79

f.s = 0.043

Avoid Failure Gear Tooth Due to Pitching

$$S_w = P_{eff} * f.s$$

f.s =Sw/P eff

= 4121675.024/888216.79

f.s = 4.64

DESIGN IS NOT SAFE.

Table 1 Beam Strength and Wear Strength of different materials

S.no	Material	Beam Strength (N)	Factor of Safety	Wear Strength (N)	Factor of Safety
1	Grey Cast Iron	9472.472	1.003	20043.544	2.14
2	Titanium Alloy (Ti64)	37568.61	2.27	55203.678	3.341
3	Structural Steel	11496.56	0.93	18633.94	1.51
4	Aluminium Alloy (Al 6061)	38531.9	0.43	4121675.024	4.64

From the theoretical Calculations it is found that factor of safety for Existing material i.e., Grey Cast Iron and some alternating materials like Titanium Alloy (Ti64), Structural Steel, Aluminium Alloy (Al 6061) based on Beam Strength and Wear Strength and it is found out that factor of safety for Titanium alloy is higher than other materials

CHAPTER 5

ANALYSIS

5 ANALYSIS

5.1 ANSYS:

With the emerging importance of CFD and finite element analyses, it is of great necessity that engineering students get a good base of knowledge on one of the most used software packages in the industry of simulation, ANSYS.

ANSYS is a finite element analysis package used widely in industry to simulate the response of a physical system to structural loading, and thermal and electromagnetic effects. ANSYS uses the finite-element method to solve the underlying governing equations and the associated problem-specific boundary conditions.

FEM, A computer-based analysis technique for calculating the strength and behavior of model during the given limits. In the FEM the model is represented as finite elements and is joined at special points which are called as nodes. Finite element analysis is the numerical solution of the mechanical components that are acquired by discretizing the mechanical elements into a small finite number of building blocks (known as elements) and by investigation those mechanical components for their acceptability and reliability. FEM is the simple technique as compared as the theoretical methods to discover the stress developed in a pair of gears. Models for numerical analysis have been prepared in SOLIDWORK and these have been bring in into ANSYS as IGES files for further analysis. The proportions of gear obtained from theoretical analysis have been used for preparing geometric model of gear. The condition for analysis has been assumed as static

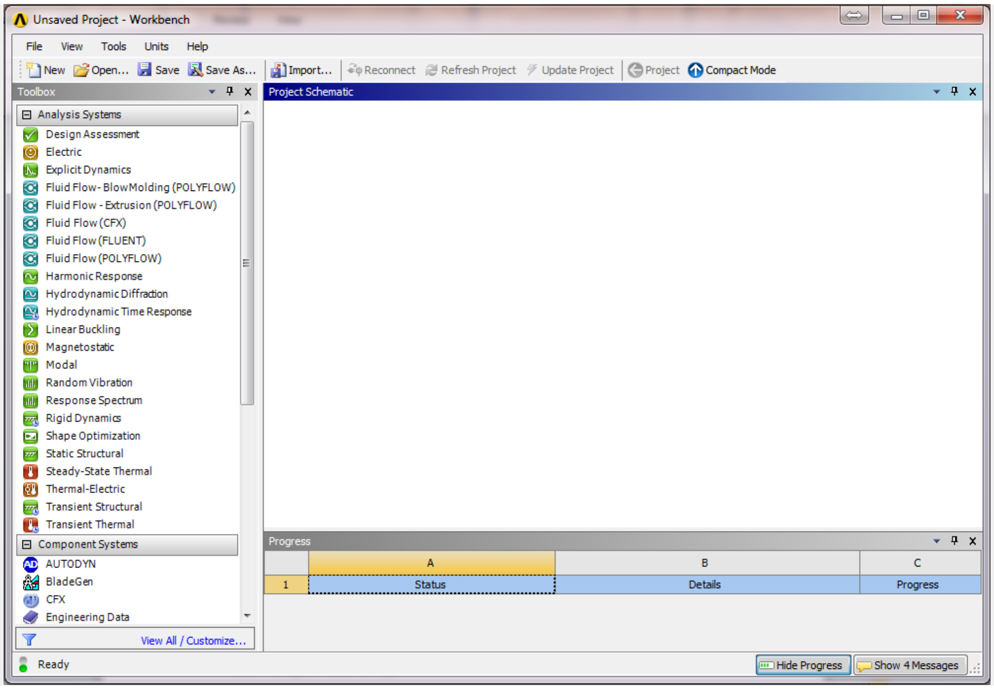


Fig 5.1 Ansys Workbench

This manual includes the procedure of solving the (static structural, Fluent) problems

Each one of the analysis systems has its own procedure. However, there are some common stages in all of the systems:

The ANSYS installation has many packages included. For this, we will be

using ANSYS Workbench.

Start menu > ANSYS 15.0 > Workbench 15.0

5.2 STEPS

5.2.1 Step1: Selection of Workbench

In ANSYS Workbench window:

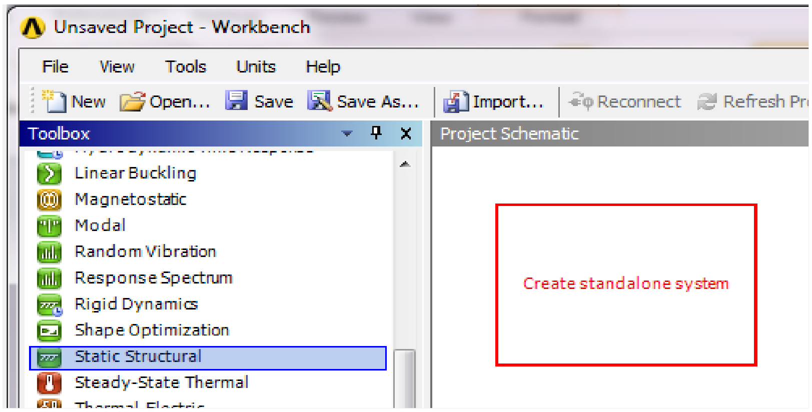


Fig 5.2 Selection of Workbench

- Drag (Static Structural) to the Project Schematic inside the red square

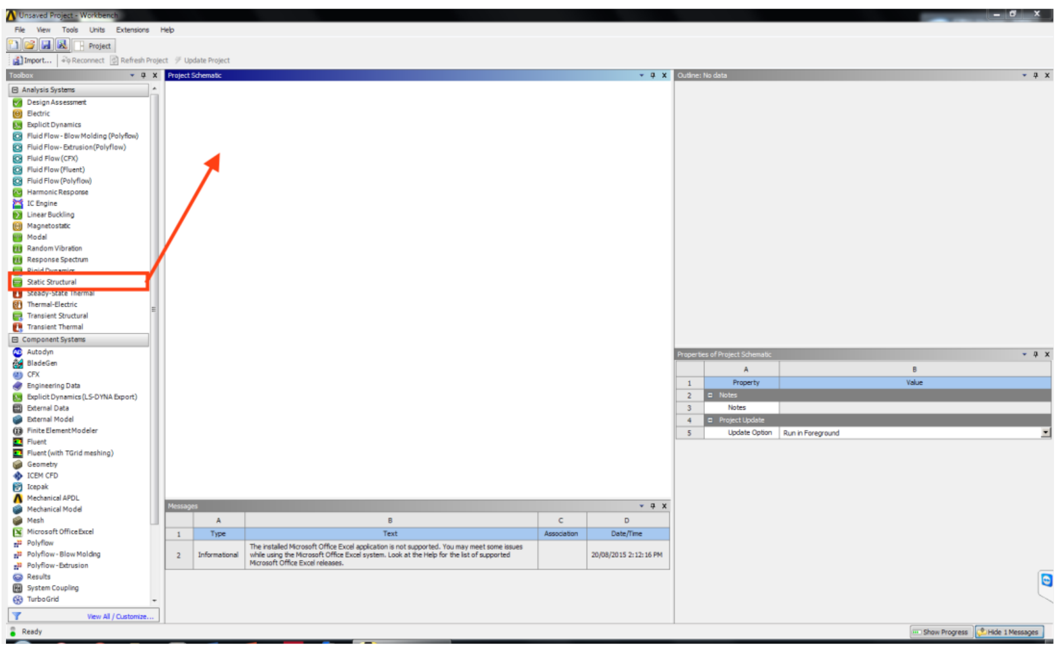


Fig 5.3 Static Structual Worbench

- Double Click on (Engineering Data) to configure and add the materials that would be used in the analysis along with their properties.
- ** The shown window will appear where a new material can be added >> (click here to add a new material)>> add (material for the beam)

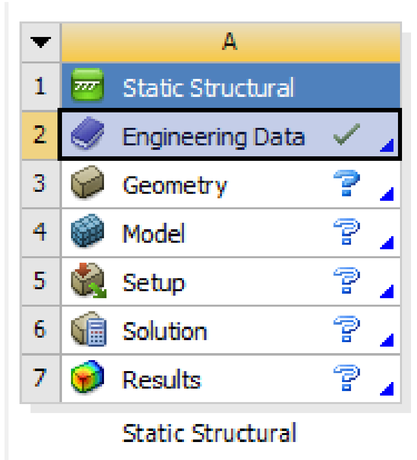


Fig 5.4 Engineering Data

5.2.2 Step 2: Engineering Data

- Double--click Engineering Data. What you see in this window may differ from thescreenshot below. In here, you can add a new material by defining a new materialentry for Mild Steel. We want to define the material as an isotropic elastic one.

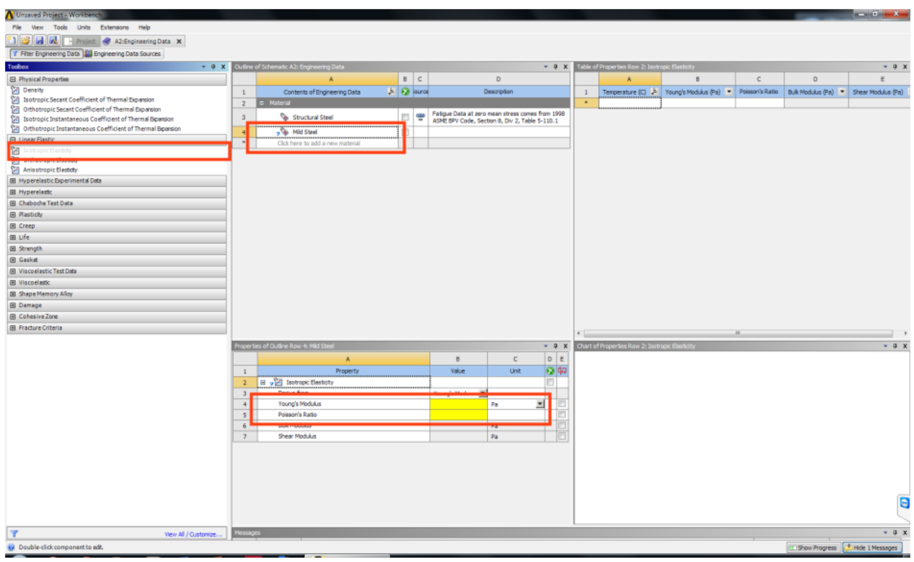


Fig 5.5 Selection of material in Engineering Data

- In the Toolbox, the material properties can be added from "Density" or "Isotropic Elasticity". Double Clicking on the mentioned options will open new fields in the outline where the fields have to be filled with the values of the properties.

- Note: Try to find the desired material in the "Engineering Data Source" Library before adding a new material. Click on the icon >> select the type of the material and the materials will appear in a list. If you want to add a material to your project list, click on After you are done with adding all the materials needed in the project, click on "Return to Project"
- The Engineering Data field should be marked with a indicating that the process of adding materials properties has been done.

5.2.3 Step 3: Geometry

- ** Right Click on (Geometry) >> Import Geometry >> Browse >> Locate the geometry file
- Note: Simple geometry can be constructed in Ansys Geometry window itself. However, complex geometry should be imported from 3D modeling software like Solidworks, as it has been done in this exercise.

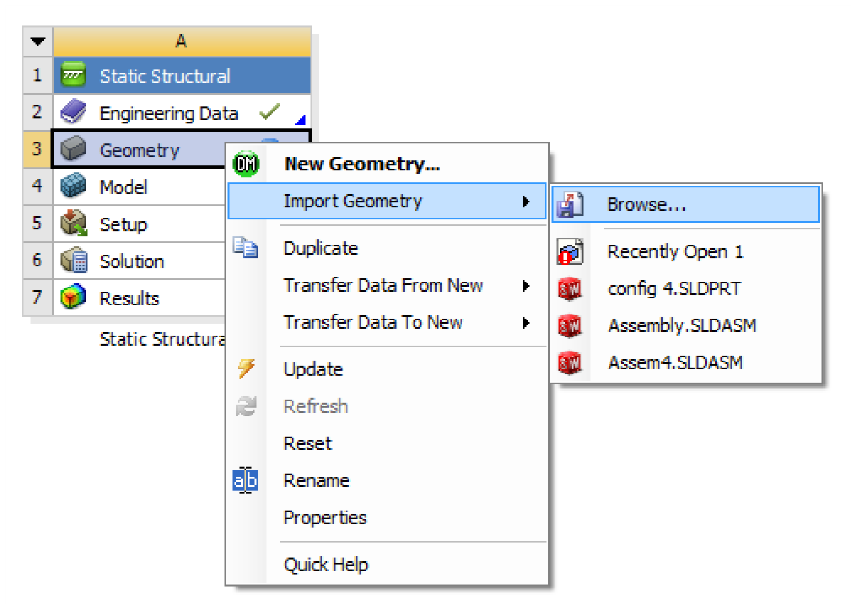


Fig 5.6 Selection of Geometry

- on the Tree Outline on the left side >> Right Click on "Import" >> Generate. Hence, the geometry will appear in the graphics window. After this step, close the geometry window.

- Double click on “Model”
- on the outline window, expand the “Geometry” tree by clicking on “+”, this tree should show you all the parts in the project (will be clear when there are multiple parts in the project). Moreover, the tree helps in assigning different material to different parts or managing the contact type between two parts (Frictional, Frictionless, etc).

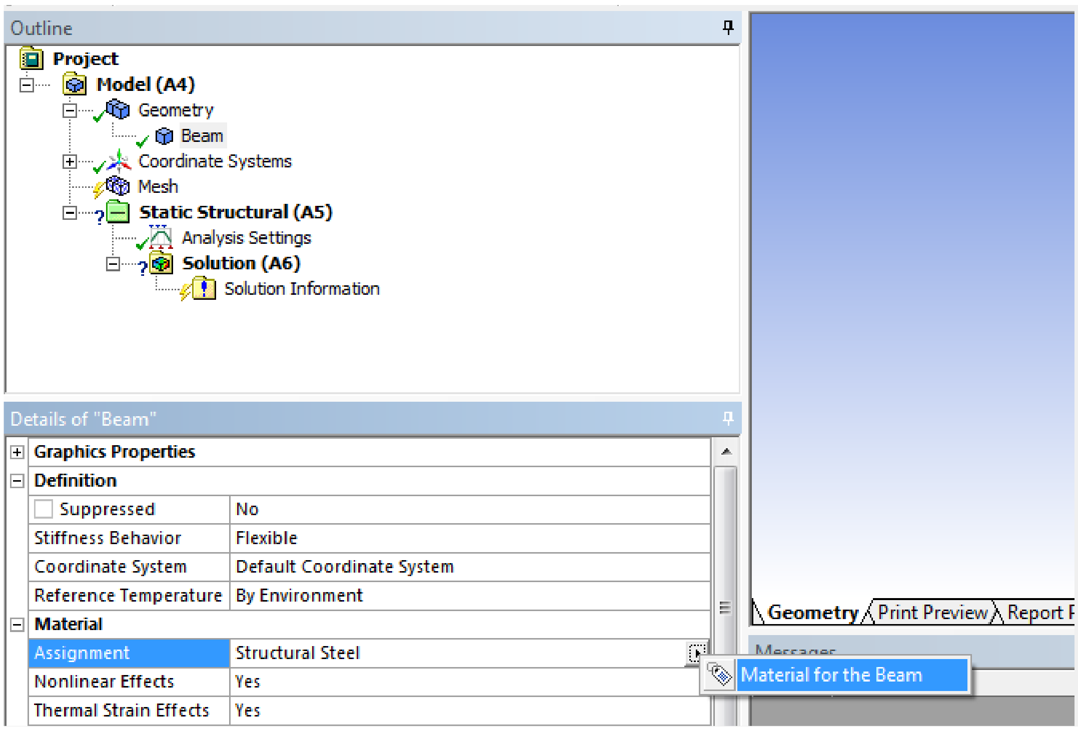


Fig 5.7 Workbench Tree

5.2.4 Step 4: Mesh generation

- In the analysis of the gear assembly, it is mandatory to study of its structural behaviour at different load and condition. 3–D model of the gear assembly were made in solidwork and were carried out in ansys analysis software as an iges file format . thereafter importing the model in ansys the suitable material was applied to the model and then meshing were done in ansys by which the whole body is divided into small tetrahedral element connected by nodes

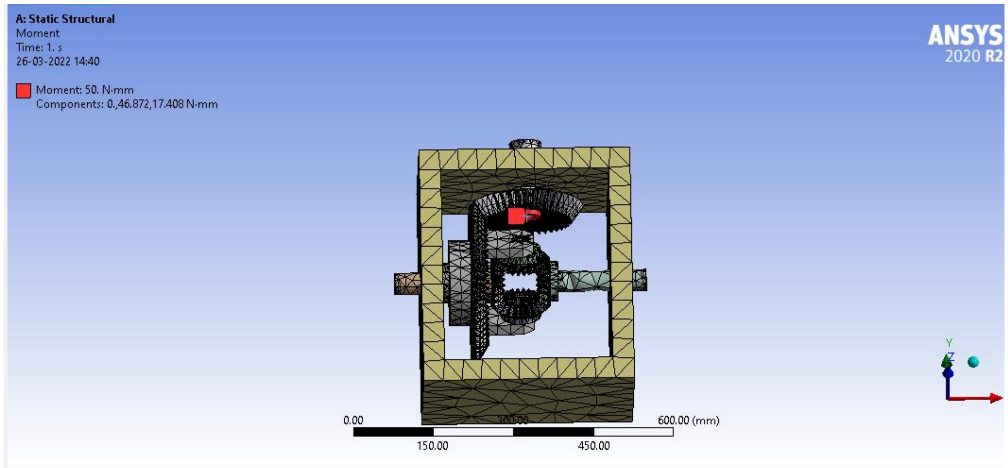


Fig 5.8 Mesh Generation

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Steps

- on the outline window, click on "Mesh". For generating the mesh with the default size, click on from the top bars. For advanced mesh options, adjust the settings from "Details of Mesh" window.
- Note: The default mesh is usually a very basic grid with no attention given to the details of the geometry. Advanced mesh details can be added by choosing the geometrical detail and inserting "sizing" as it is shown in the figure. The details can be chosen using the selecting icons.

5.2.5 Step 5: Adding forces

- Add the force from the "Loads" list. In the "Details of Force" window, change "Defined By" to "Components" and then set the "Y" direction force to be " - 1000 N" as it is shown in the figure.

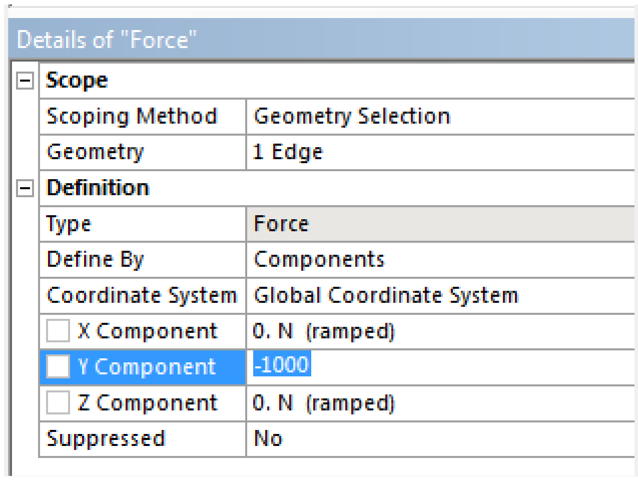


Fig 5.9 Adding Moments to the assembly

5.2.6 Step 6: Solution

- To define the desired solution parameters, click on "Solutions" and define all the parameters needed to be found. The parameters can be chosen from the lists shown in the figure.
- After defining the investigation parameters, click to get the results. To show the results of the different parameters, use the list under "solutions" in the "Outline" window.

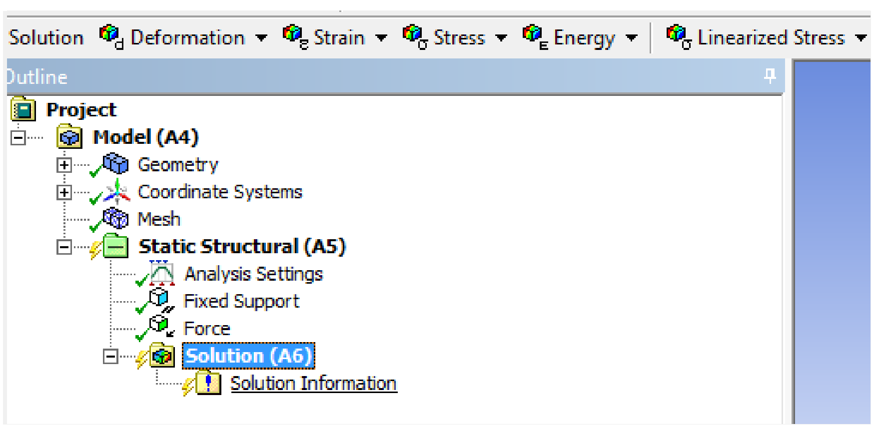


Fig 5.10 Generating Solution

5.3 SOLUTION FOR ALUMINIUM ALLOY

5.3.1 Moment 100 N-MM Al Alloy

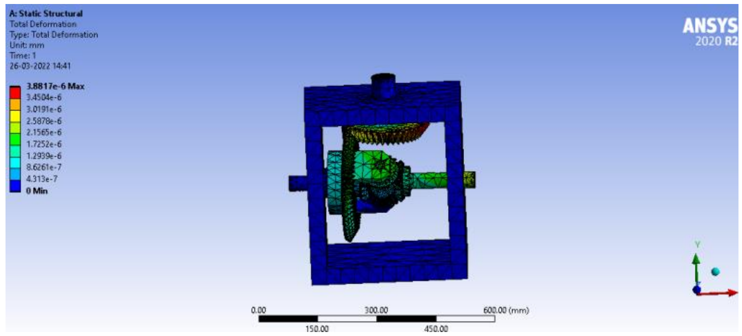


Fig 5.11Total Deformation for 100 N-mm moment

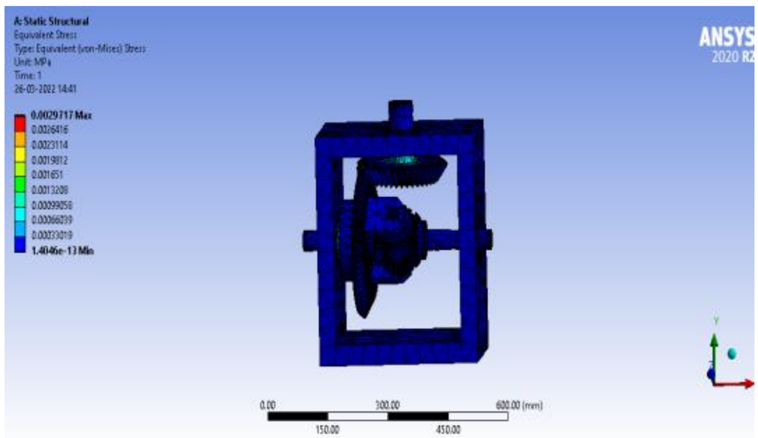


Fig 5.12Equivalent Von mises Stresses at 100N-mm

5.3.2 Moment at 500 N-MM Al alloy

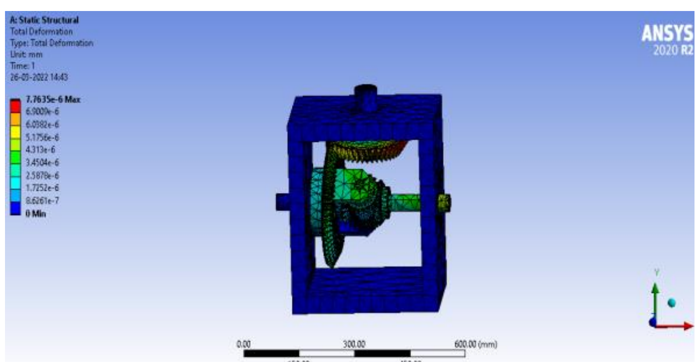


Fig 5.13 Total Deformation for 500 N-mm moment

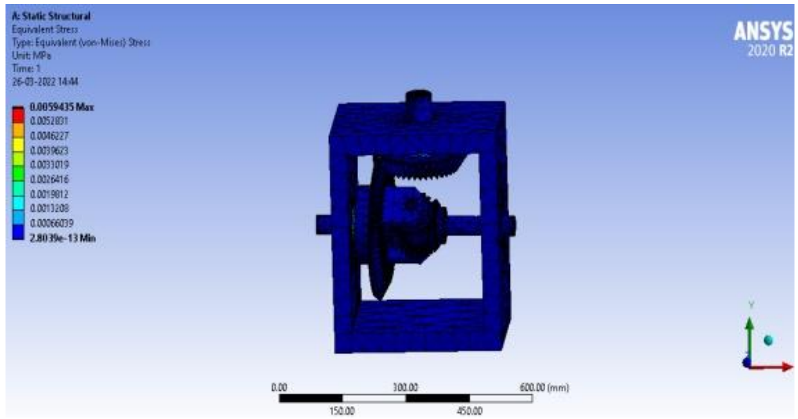


Fig 5.14Equivalent Von mises Stresses at 500N-mm

5.3.3 Moment at 1000N-MM Al alloy

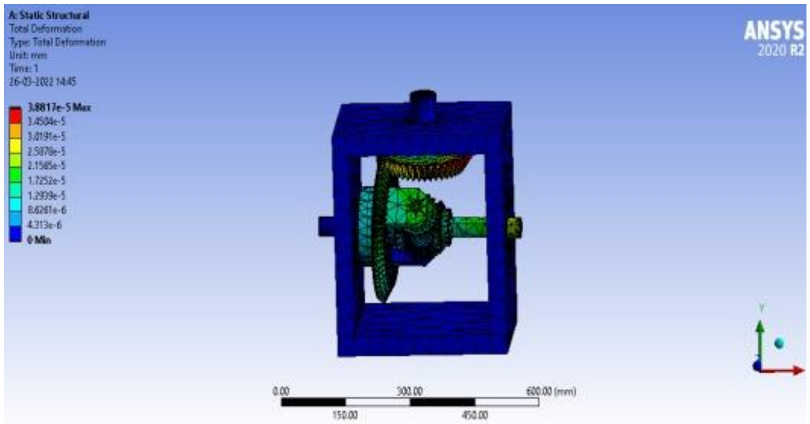


Fig 5.15 Total Deformation at 1000 N-mm moment

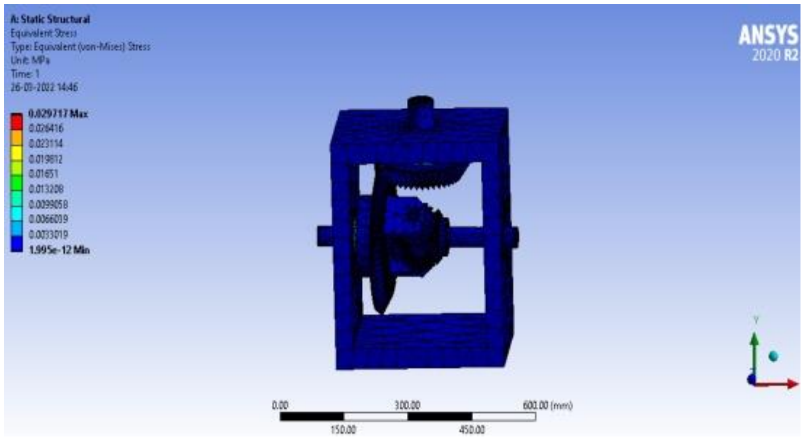


Fig 5.16Equivalent Von mises Stresses at 1000N-mm

5.3.4 Moment 1500 N-MM

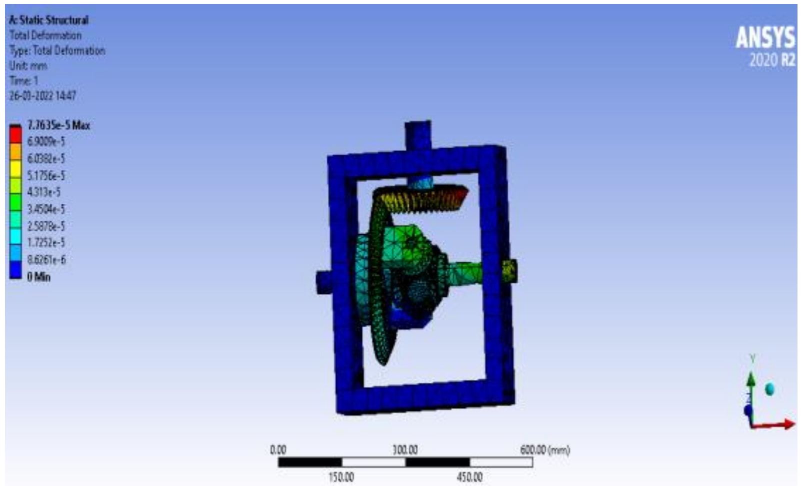


Fig 5.17 Total Deformation at 1500 N-mm moment

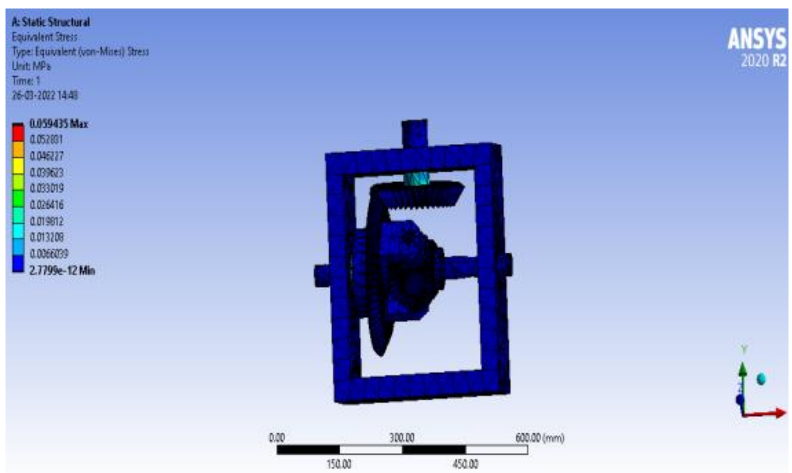


Fig 5.18Equivalent Von mises Stresses at 1500N-mm

5.4 SOLUTION FOR TITANIUM ALLOY

5.4.1 Moment at 100 N-MM

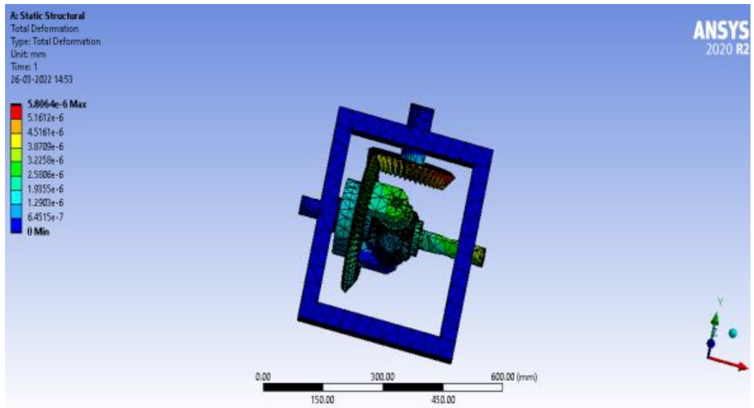


Fig 5.19 Total Deformation for 100 N-mm moment

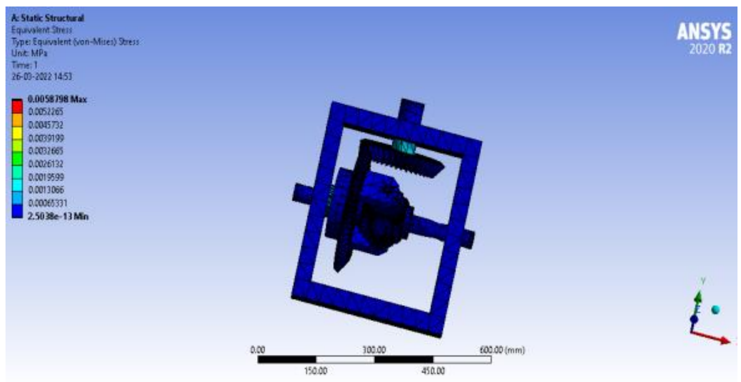


Fig 5.20Equivalent Von mises Stresses at 100N-mm

5.4.2 Moment at 500 N-MM

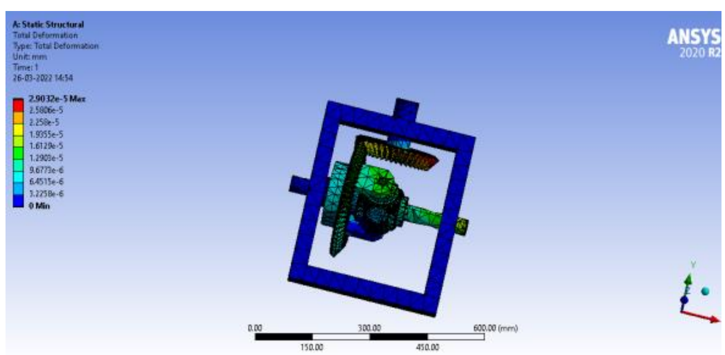


Fig 5.21 Total Deformation for 500 N-mm moment

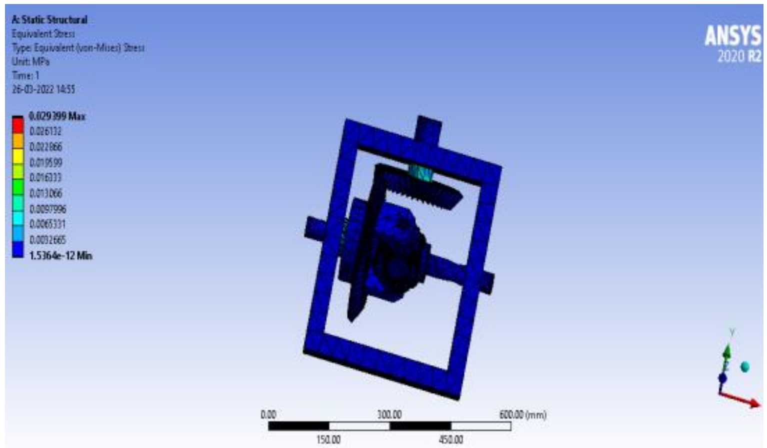


Fig 5.22Equivalent Von mises Stresses at 500N-mm

5.4.3 Moment at 1000 N-MM

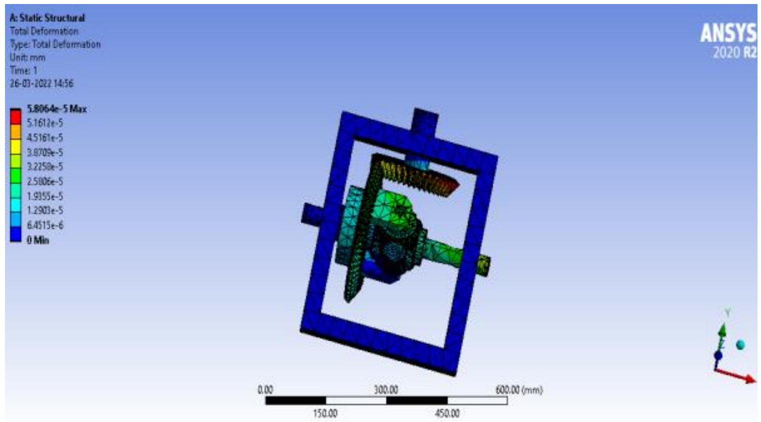


Fig 5.23 Total Deformation at 1000 N-mm moment

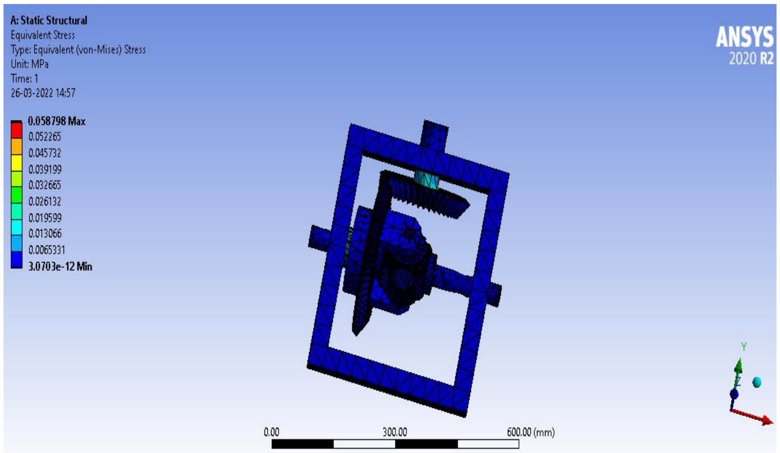


Fig 5.24Equivalent Von mises Stresses at 1000N-mm

5.4.4 Momentat 1500 N-MM

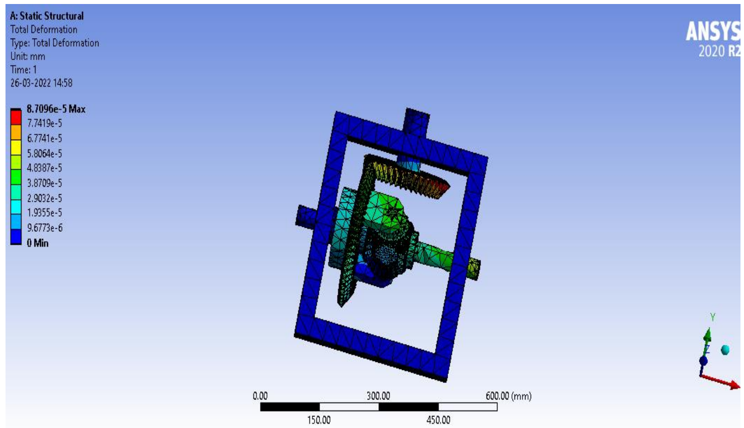


Fig 5.25 Total Deformation at 1500 N-mm moment

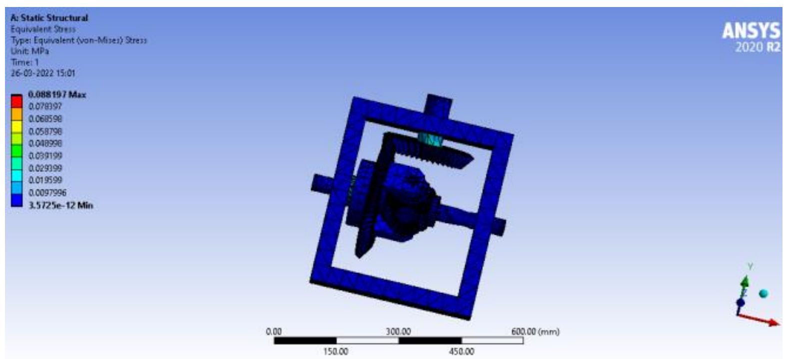


Fig 5.26Equivalent Von mises Stresses at 1500N-mm

5.5 SOLUTION FOR GREY CAST IRON

5.5.1 Moment at 100 N-MM

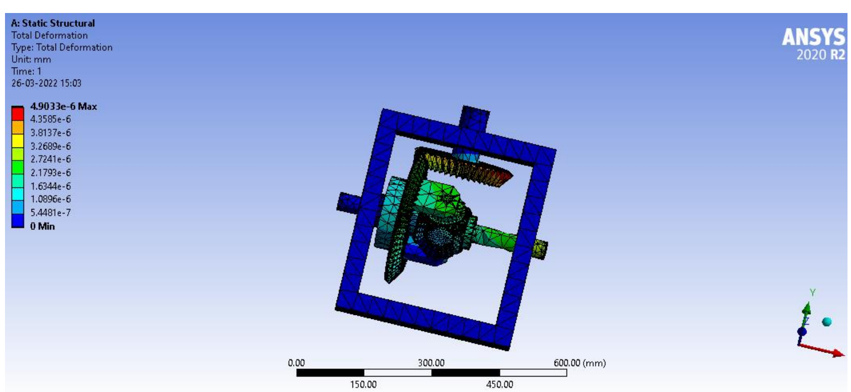


Fig 5.27 Total Deformation for 100 N-mm moment

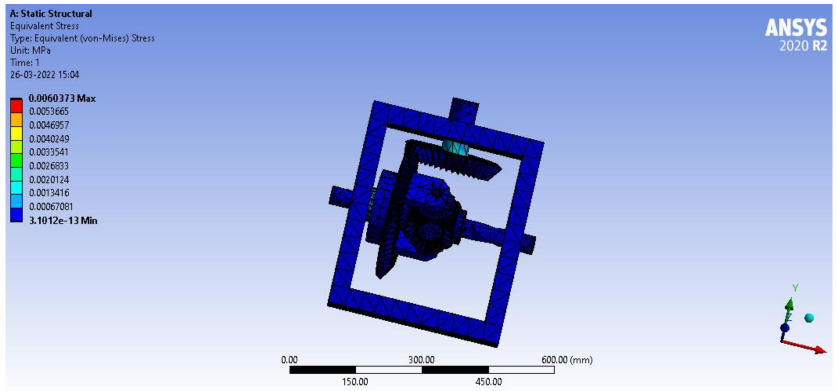


Fig 5.28Equivalent Von mises Stresses at100 N-mm moment

5.5.2 Moment at 500 N-MM

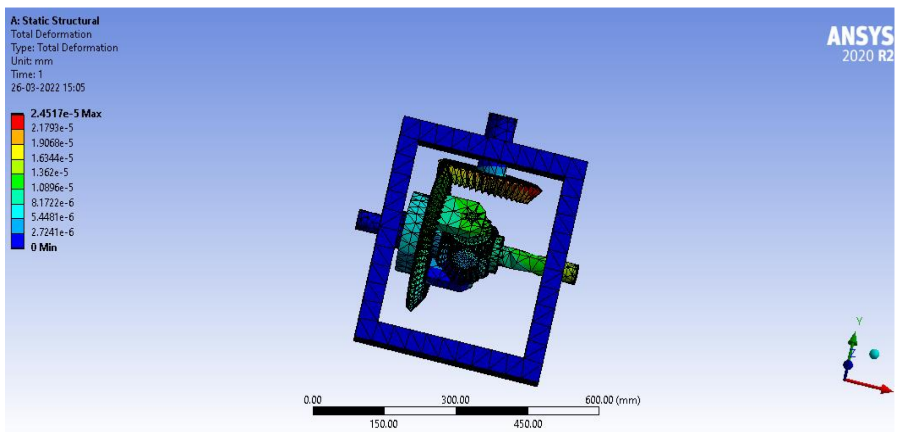


Fig 5.29 Total Deformation for 500 N-mm moment

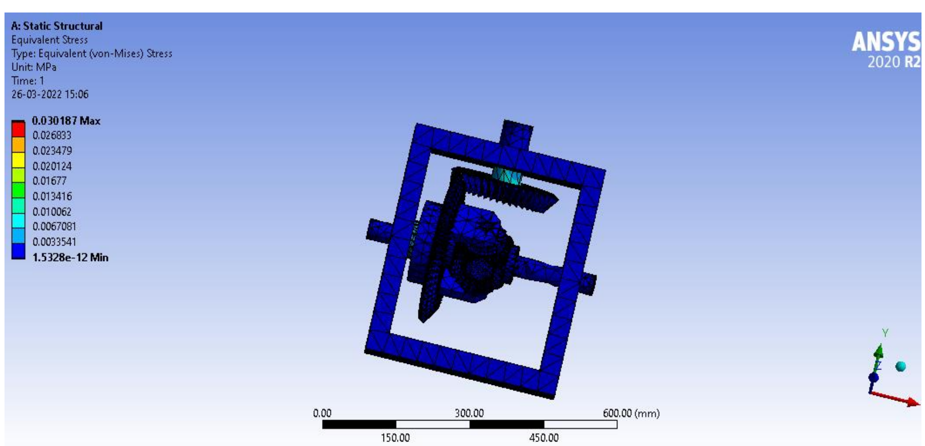


Fig 5.30Equivalent Von mises Stresses at 500 N-mm moment

5.5.3 Moment at 1000 N-MM

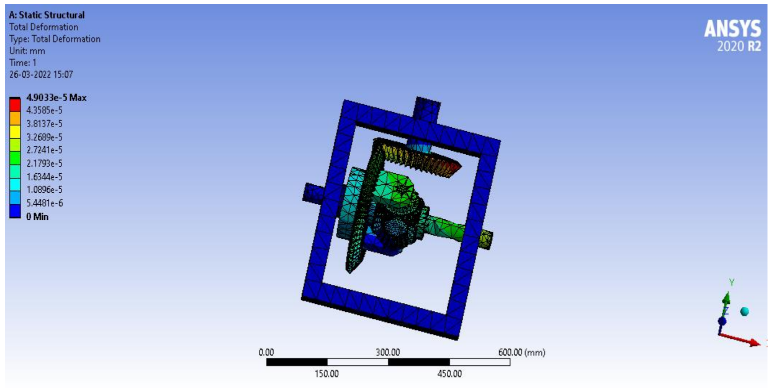


Fig 5.31 Total Deformation for 1000 N-mm moment

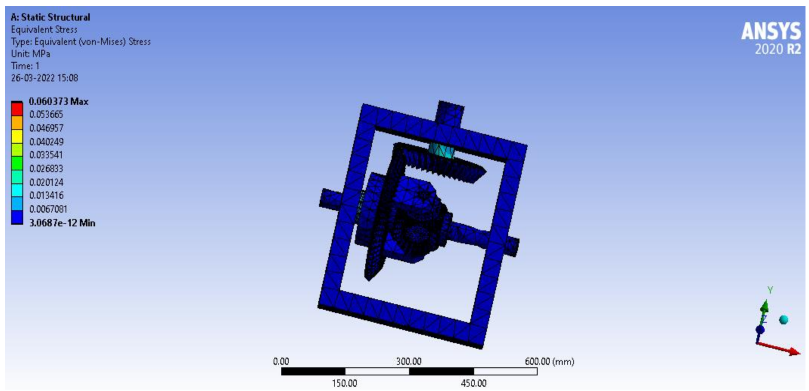


Fig 5.32Equivalent Von mises Stresses at 1000 N-mm moment

5.5.4 Moment at 1500 N-MM

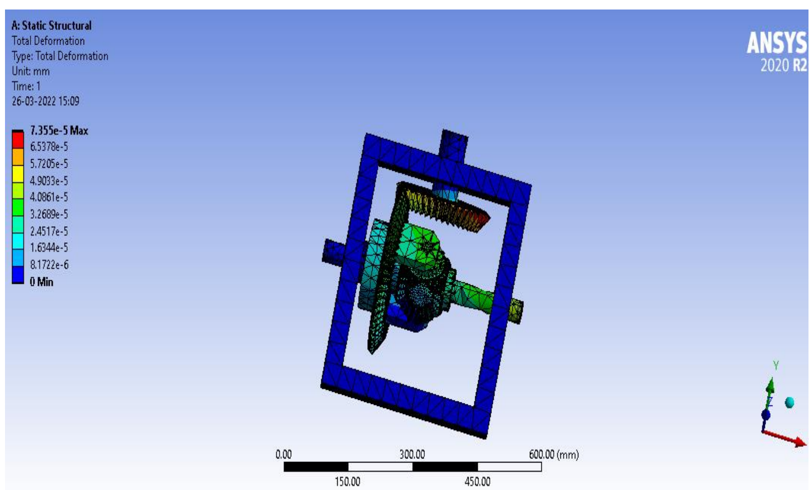


Fig 5.33 Total Deformation for 1500 N-mm moment

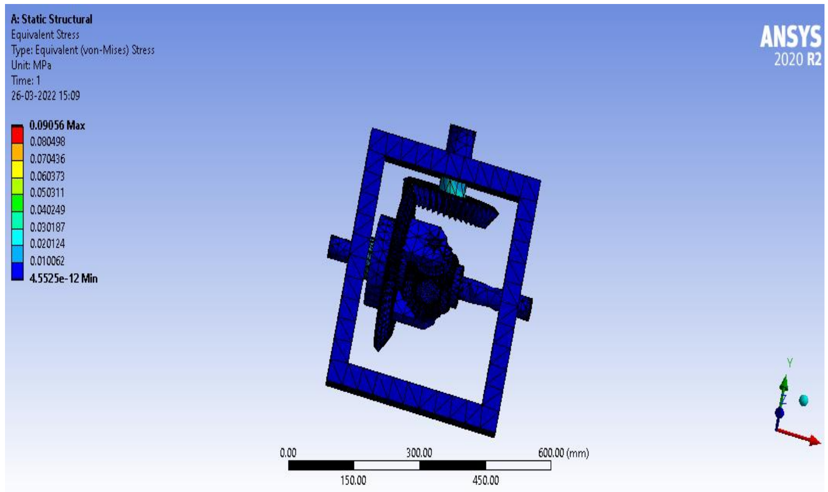


Fig 5.34Equivalent Von mises Stresses at 1500 N-mm moment

5.6 SOLUTION FOR STRUCTURAL STEEL

5.6.1 Moment at 100N-MM

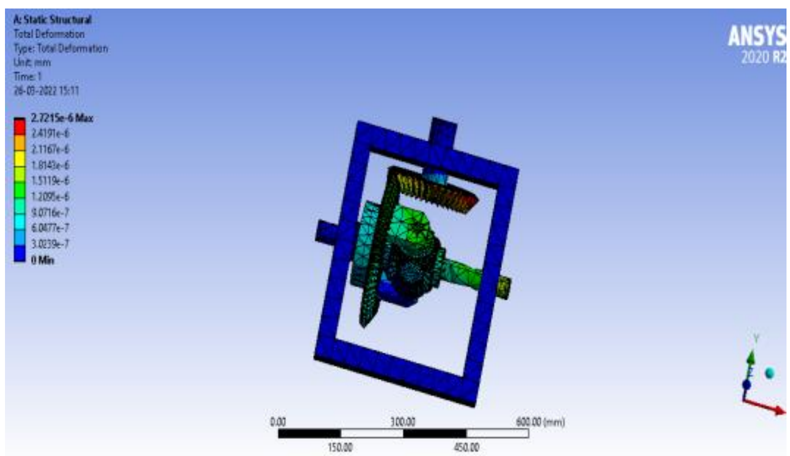


Fig 5.35 Total Deformation for 100 N-mm moment

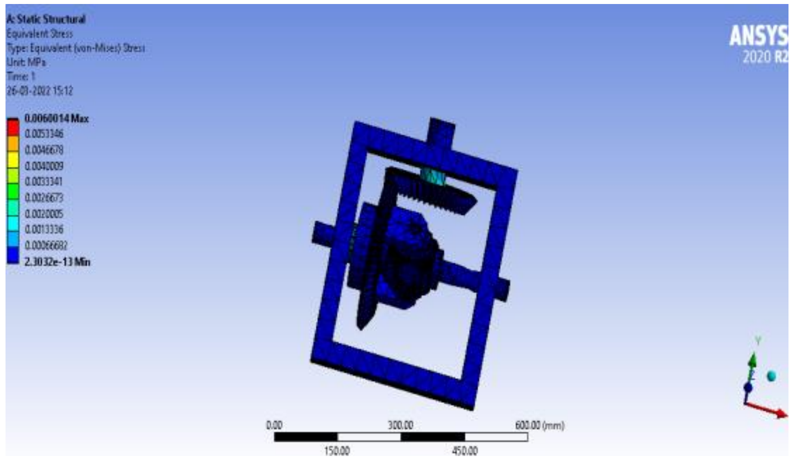


Fig 5.36Equivalent Von mises Stresses at 100 N-mm moment

5.6.2 Moment at 500N-MM

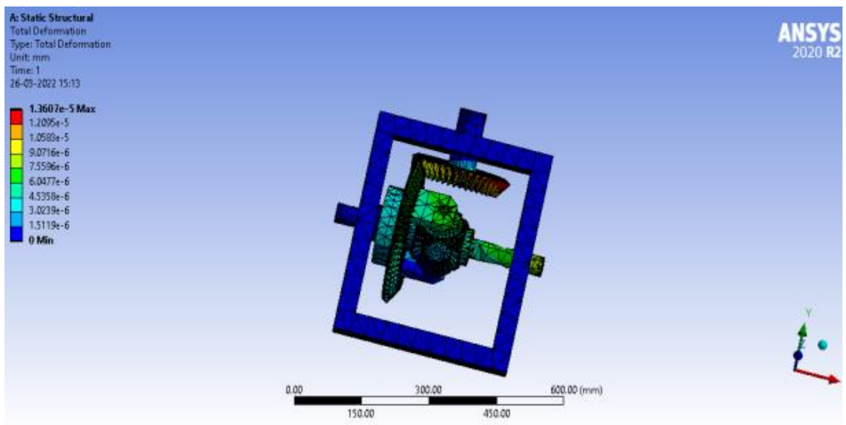


Fig 5.37 Total Deformation for 500 N-mm moment

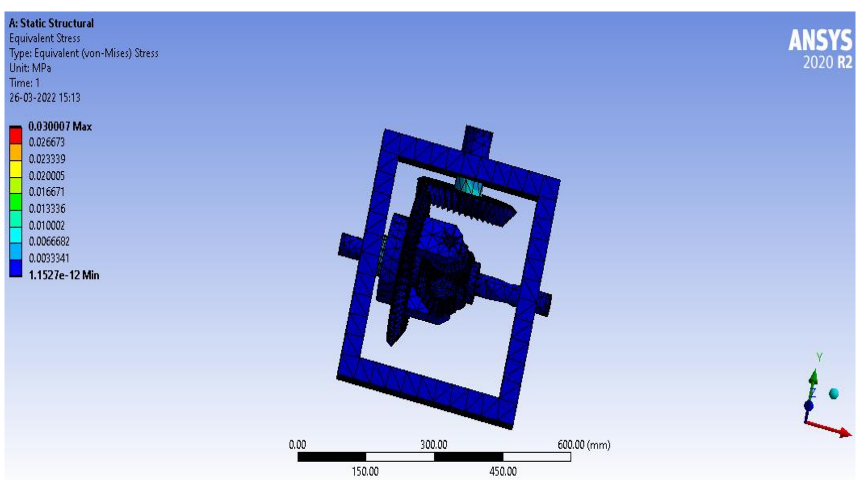


Fig 5.38Equivalent Von mises Stresses at 500 N-mm moment

5.6.3 Moment at 1000 N-MM

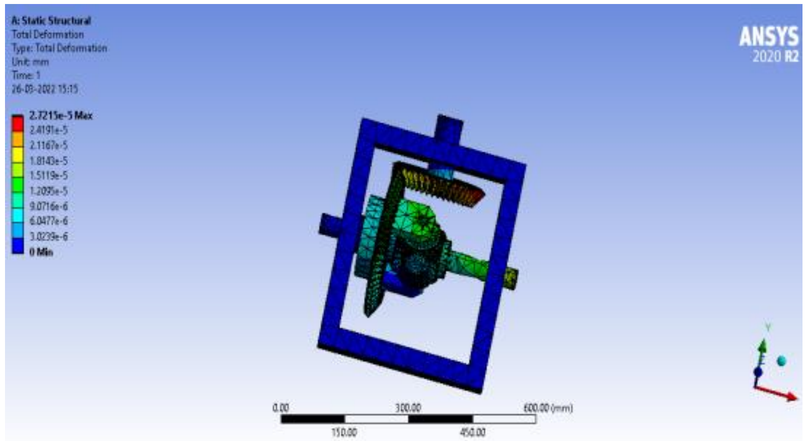


Fig 5.39 Total Deformation for 1000 N-mm moment

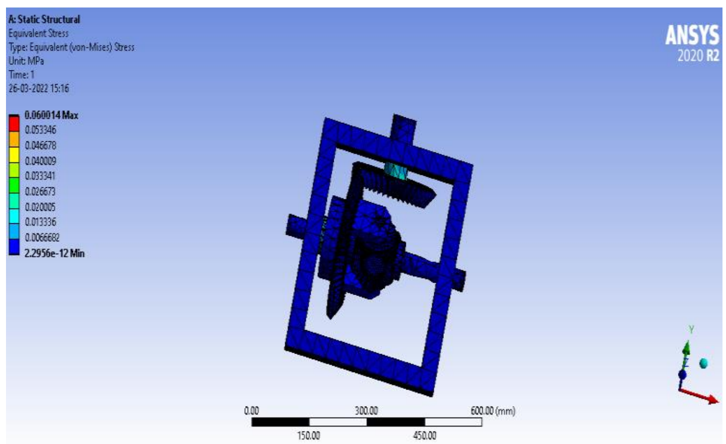


Fig 5.40Equivalent Von mises Stresses at 1000 N-mm moment

5.6.4 Moment at 1500 N-MM

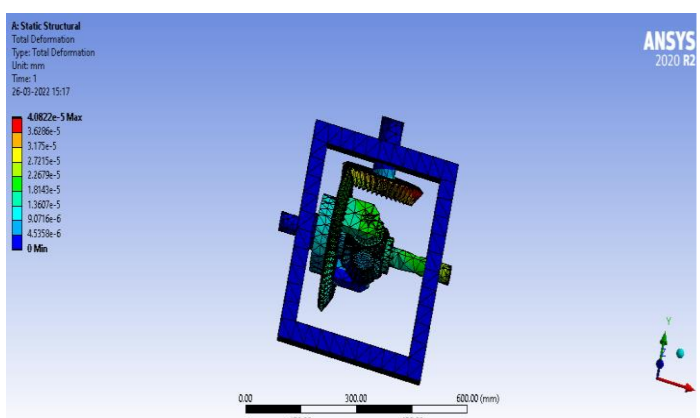


Fig 5.41 Total Deformation for 1500 N-mm moment

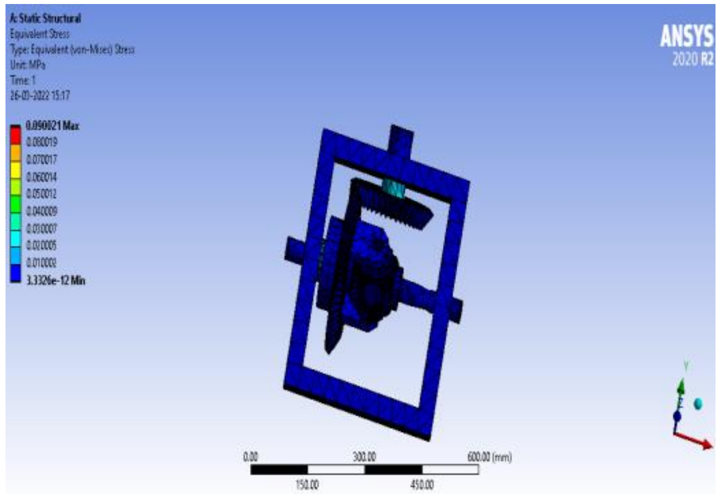


Fig 5.42Equivalent Von mises Stresses at 1500 N-mm moment

5.7 OBSERVATION AND RESULTS

After performing the analysis using ANSYS software total deformation and von mises stresses are observed at different moments on the differential.

The obtained resultant von mises stresses and total deformation are tabulated in the following tables.

Table 2 Moment at 100 N-MM

Material	Vonmises stress (Mpa)	Total Deformation (mm)
Al Alloy	7.76E-05	0.059435
Structural steel	5.81E-05	0.0058798
GCI	4.90E-06	0.0060373
Ti alloy	2.72E-06	0.0060014

Table 3 Moment at 500 N-MM

Material	Vonmises stress (Mpa)	Total deformation
Al alloy	3.88E-05	0.029717
Structural steel	2.90E-05	0.029399
GCI	2.45E-05	0.030187
Ti alloy	1.36E-05	0.030007

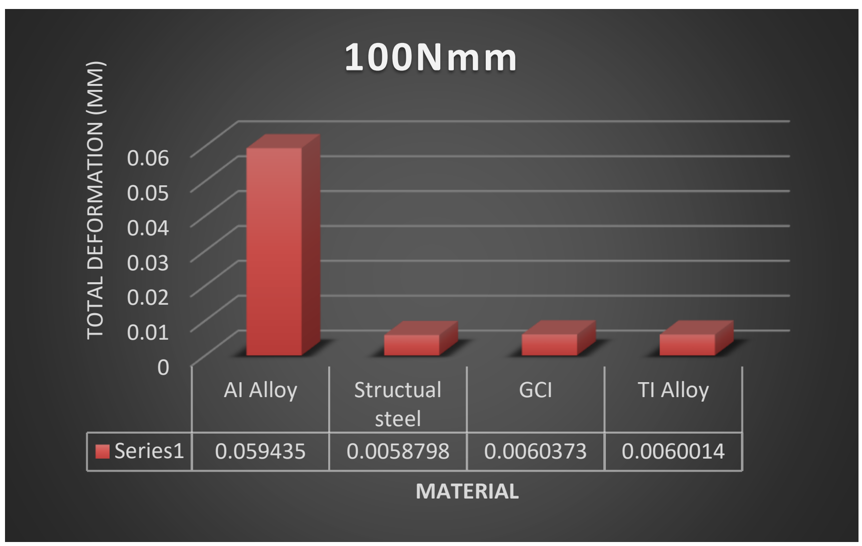
Table 4 Moment at 1000 N-MM

Material	Vonmises stress (Mpa)	Total Deformation (mm)
Al alloy	7.76E-05	0.059435
Structural steel	5.81E-06	0.058798
GCI	4.90E-05	0.060373
Ti alloy	2.72E-05	0.060014

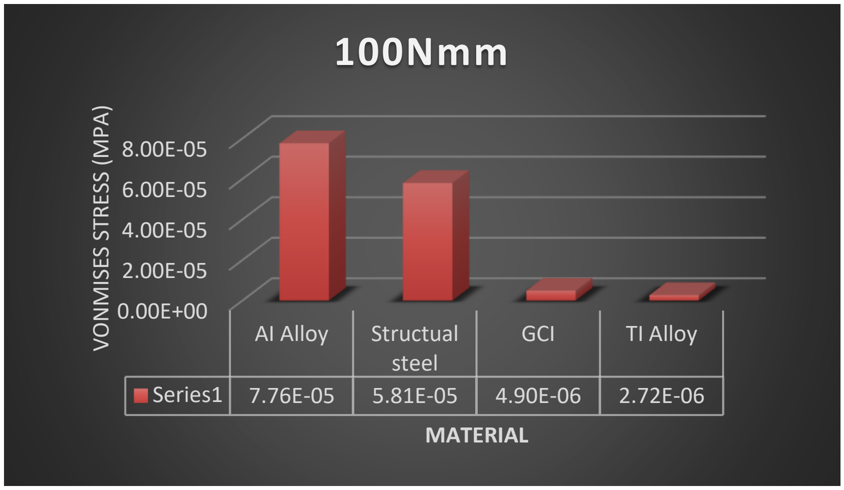
Table 5 Moment at 1500 N-MM

Material	Vonmises stress (Mpa)	Total Deformation (mm)
Al alloy	0.00011645	0.089152
Structural steel	8.71E-05	0.088197
GCI	7.36E-05	0.09056
Ti alloy	4.08E-05	0.090021

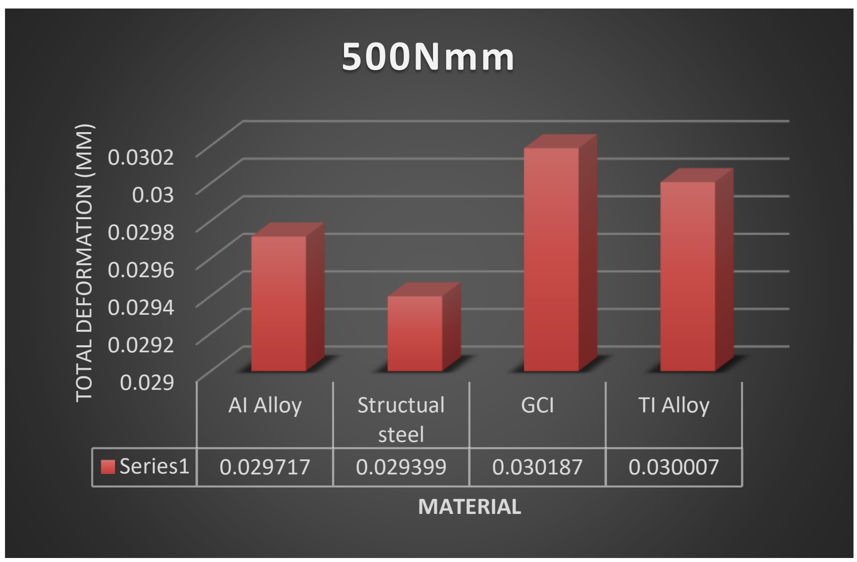
GRAPHS



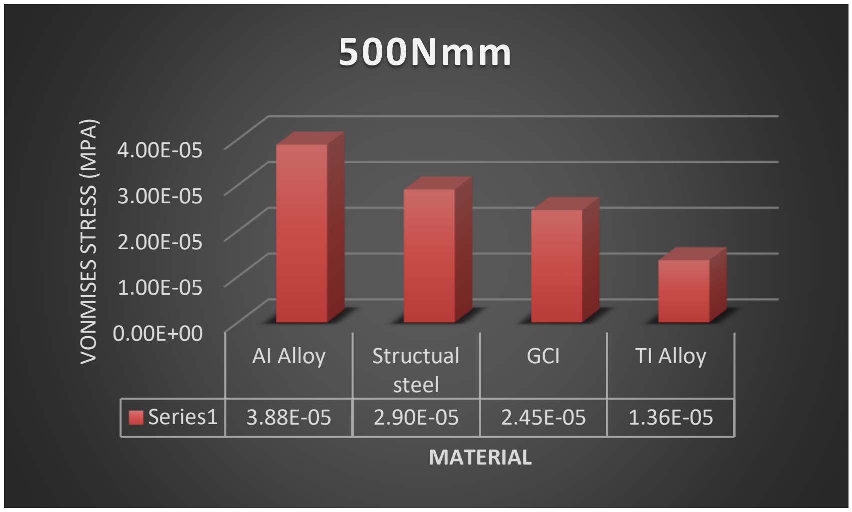
Graph 5.1 Total Defromation Vs Material at 100 Nmm



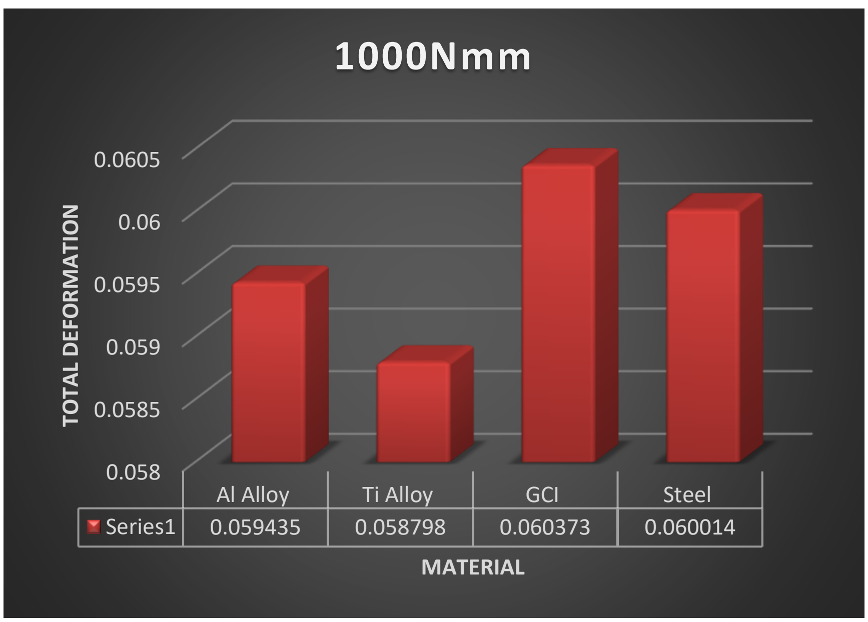
Graph 5.2 Vonmises Stress Vs Material at 100 Nmm



Graph 5.3Total Defromation Vs Material at 500 Nmm

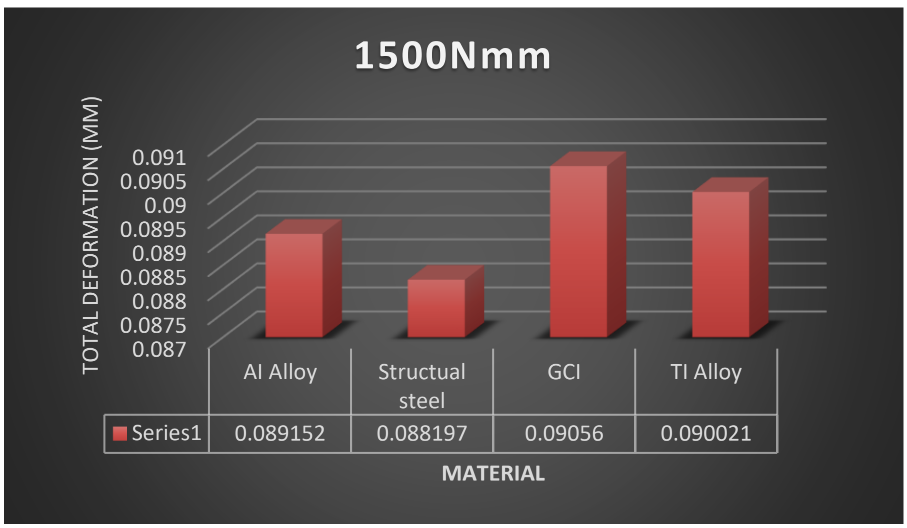


Graph 5.4 Vonmises Stress Vs Material at 500 Nmm

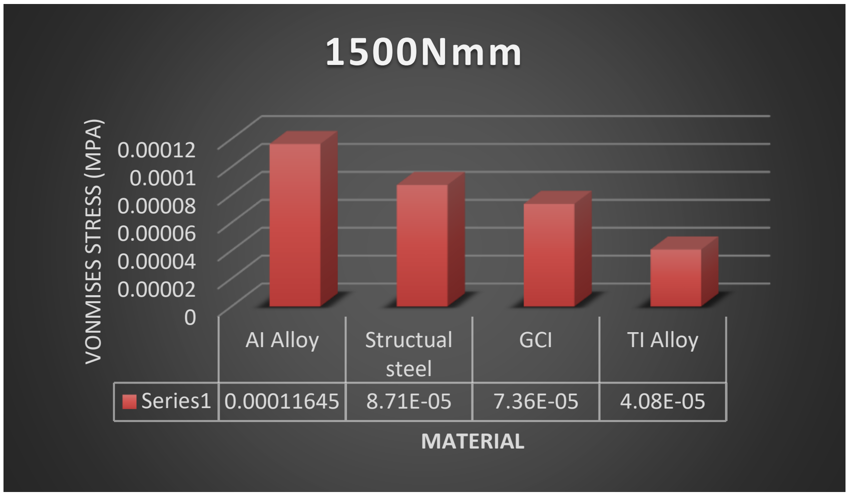


Graph 5.5 Vonmises Stress Vs Material at 1000 Nmm

Graph 5.6 Vonmises Stress Vs Material at 1000 Nmm



Graph 5.7 Total Defromation Vs Material at 1500 Nmm



Graph 5.8 Vonmises Stress Vs Material at 1500 Nmm

Graphs are drawn based on vonmises stresses and total deformation observed at 100Nmm, 500Nmm, 1000Nmm and 1500Nmm. Based on the graphs and the tabulated values it is observed that Vonmises Stress and Total deformation got lower for Titanium alloy(Ti64) rather than other than other materials such as Structural Steel, Aluminium alloy(Al 6160) and Grey Cast Iron for the given moments.

CHAPTER 6

CONCLUSION

6 CONCLUSION

The behaviour of the differential present in an automobile is studied by performing design calculations on the differential and performing static analysis using Ansys workbench 2020. The individual parts of a bevel gear differential were modelled by using solid works and the assembly was imported to ansys workbench. It was observed from design calculations based on input power , GCI material and Ti alloy exhibits good beam strength and wear strength values and the values of factor of safety for these two materials are observed to be greater than one. The values obtained at moment at 100 N-mm for Ti alloy are 5.81E-05 Mpa(vonmises stress) and 0.0058798 mm (total deformation)

After performing, static analysis on differential with varying torques applied on the input shaft, i.e (100Nmm, 500Nmm, 1000Nmm,1500Nmm) Ti alloy exhibits lower values of vonmises stress and deformation compared to remaining materials followed by grey cast iron.

Hence , Ti alloy(Ti 64) material may have chance to replace the existing grey cast iron materials used in gear boxes. Further, manufacturing methods and the amount of inclusion of Ti alloy in the base material has to be studied in detail.

Chapter 7

Future Scope

7 FUTURE SCOPE

- Cost analysis can be performed for better usage of Ti alloy as replacement material for the gears present in differential.
- Dynamic and thermal analysis can be performed for the complete assessment of the entire gear box assembly.
- The Composition of Ti in the base alloy selection is very important for reduction of manufacturing cost of gears of a differential. Further experimental analysis is needed in this area.

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