DESIGN AND ANALYSIS OF CENTRALLY SUSPENDED CAGE-LESS LIMITED SLIP DIFFERENTIAL

A project report submitted in partial fulfillment of the requirements for the Award of the Degree of

BACHELOR OF ENGINEERING
IN
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ABSTRACT

Differential plays an important role in maintaining speeds and torque distribution between the wheels of the automobile. There are number of types of differentials based on the need are available in automobile industry such as Open differential , Limited slip differential (LSD), Locked differential, Torque vectoring differential (VTD).

In the present project work, limited slip differential is considered and the objective of the project is to find out tangential, axial and radial forces involved in meshing of the differential gears theoretically. Finite Element Analysis (FEA) is carried out on Final, Crown, Side and Ring gears made of 20MnCr5 material and running at a speed of 4000rpm and torque of 122N-m.

As a part of FEA, 3D modelling of gears was done in Solidworks 2017 and analysis was carried on ANSYS workbench 14.5. Using ANSYS, von-Misses stresses, deformation and factor of safety are computed for Final, Crown, Side and Ring gears. From the analysis, it is found that the forces and stresses obtained are below the allowable stress of the material considered in designing gears of the differential. Maximum von-Misses stress is for Ring gear and Minimum for Side gears.

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CHAPTER – I

INTRODUCTION

1.0 Major systems in automobile

Automobile is defined as wheeled vehicle driven by its own motive power unit that runs on ground and is used for transporting passengers and goods.

Major systems of an automobile are

1. Power unit
2. Control unit
3. Electrical unit
4. Suspension
5. Transmission

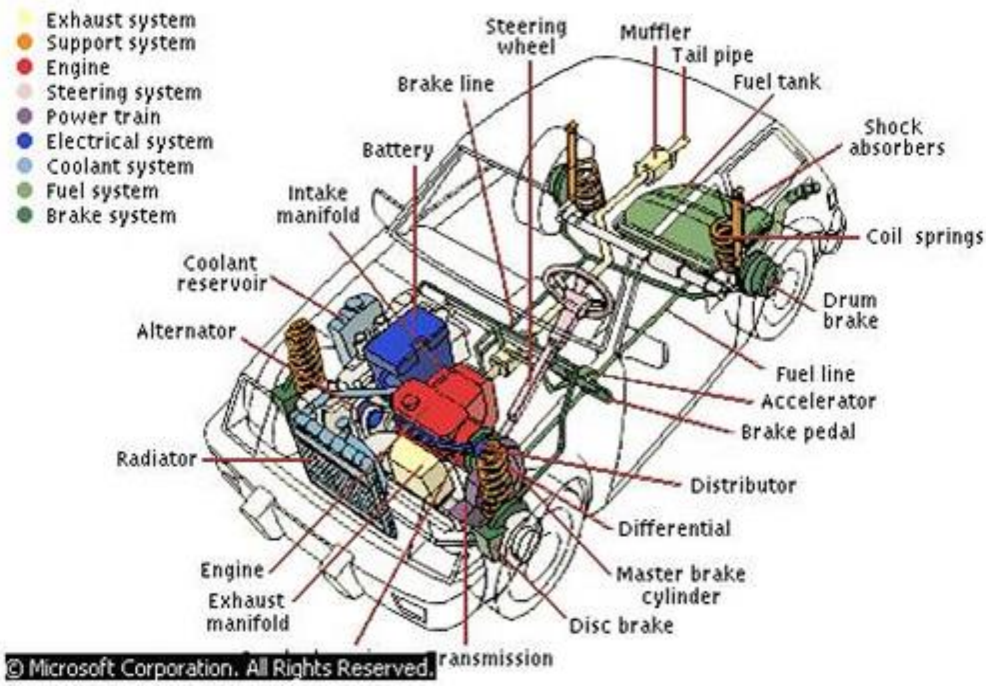


Fig 1.1 major systems in automobile

1.1 Power unit

It consists of engine that produces the required motive power. Engine is any machine that converts any form of energy into mechanical energy or motion.

In automobiles, engines convert chemical energy from fuels like petrol, diesel etc. into motive energy of the vehicle. There are many methods in doing this, however the cheapest and most effective method is to burn fuel and use this energy to move piston inside the engine.

Fuel can be burned using two methods:

a) External combustion: Fuel is burnt outside the combustion chamber.
   Example in Steam engines and these aren't used in passenger cars.
b) Internal combustion: Fuel is burnt inside the combustion chamber.
   Example in Engines used in passenger cars.

Again internal combustion engines can be divided into 2 types based on the method of combustionof fuel as

i. Spark ignition: Fuel is burnt by igniting a spark in the combustion chamber.
   Example in case of petrol engines.
ii. Compression ignition: Fuel is burnt by compressing it in the combustion chamber.
   Example in case of diesel engines.

Both spark and combustion ignition engines are divided into another 2 types based on number of strokes it takes to perform its 4 operations (Suction, compression, expansion and exhaust)

a. 2 stroke: Here, all 4 operations are completed in 2 strokes of piston i.e. suction and compression in a stroke and expansion and exhaust in another.
b. 4 stroke: Here, all 4 operations are completed in 4 strokes of piston each in a stroke

1.2 Control unit

It consists of Steering and Braking systems.

a) Steering system

The act of guiding and retaining any vehicle on its desired path is called as steering. Thus in simple words, steering is used by the driver to have directional control of the vehicle.

It is done by keeping the front wheels in desired direction by the driver using a steering wheel.

For steering action to be smooth, the front wheel geometry holds a pivotal role. For easy turning, the front tyres have to turn about a point which intersects the axis of the rear axle and that point is known as centre of rotation. To maintain this geometry and to reduce the amount of effort required to steer the vehicle, steering mechanisms consisting of rods, links, joints and gears are employed.

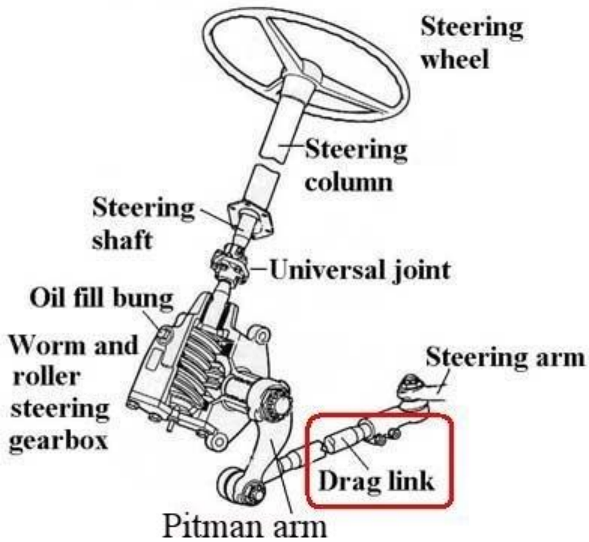


Fig 1.2 Components of Steering Mechanism

b) Braking system

Brakes are required to

i. To reduce the vehicle speed quickly while driving
ii. To hold the vehicle when needed
iii. To hold vehicle on slopes

In this process, kinetic energy of the vehicle is converted to heat and dissipated. Braking system is a combination of mechanical, hydraulic and electrical components.

Brakes are a closed system containing brake fluid, when pressure is applied at a point then the same pressure is transferred by the fluid in all directions.

This works on the principle of Pascal's law i.e. when there is an increase in pressure at any point in a confined fluid, there is an equal increase at every other point in the container i.e. pressures everywhere is same.

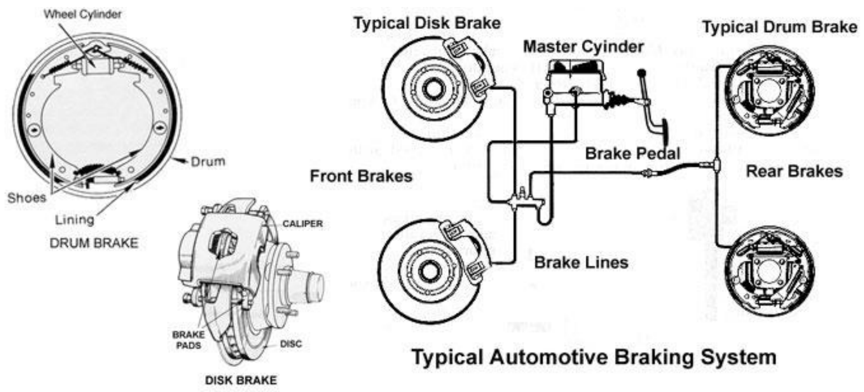


Fig 1.3 Braking System

1.3 Electrical unit

It consists of the starting system and auxiliaries.

Starting system:

When key is turned on to the start position in the ignition switch, current flows from the battery to the starter solenoid. The solenoid is an electromagnetic switch mounted on the starter motor. When the coils of the solenoid are energized, a plunger is pulled back .Attached to one end of this plunger is a lever, which is connected to the drive pinion and clutch assembly of the starter motor.

As the lever is pulled, the drive pinion is brought into mesh with the teeth of the flywheel. As the starter motor is energized, it produces a torque which turns the flywheel thereby cranking the engine.

Starter motor is small but powerful electric motor delivering high torque but in a small time.

When the driver releases the ignition switch from start position to run position, the starter solenoid deactivates. Its internal return springs cause the drive pinion to be pulled out of mesh with the flywheel and starter motor stops rotating.

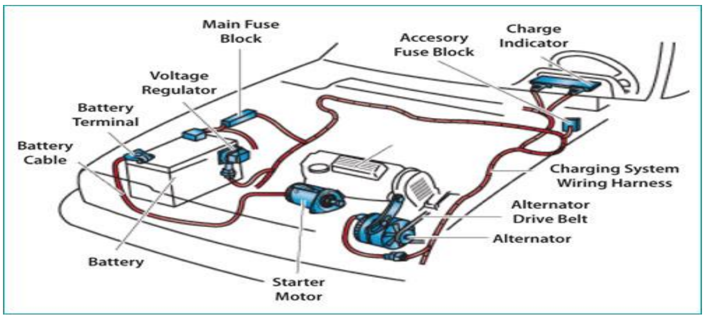


Fig 1.4 Electrical System

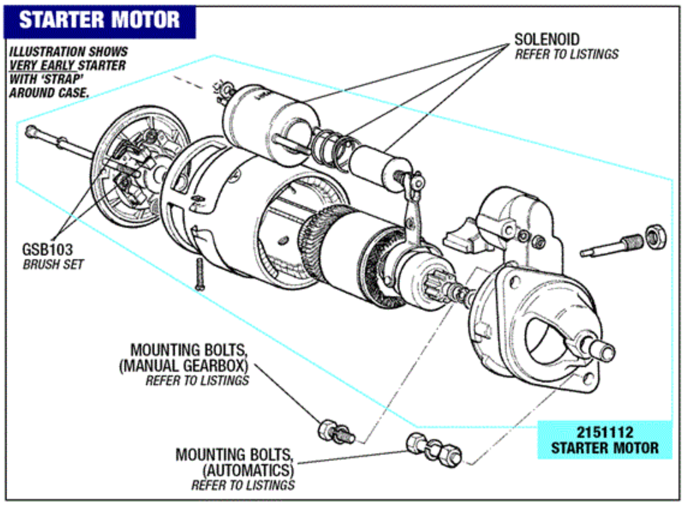


Fig 1.5 Starting System

1.4 Suspension system

It is a combination of springs, shock absorbers and linkages connected between the velocity the vehicle body and wheels allowing relative movement between them.

An appropriate suspension system will allow the vehicle to ride relatively undisturbed while travelling on rough, uneven surfaces or while maneuvering quick turns etc. by maintaining traction at all times.

Suspension helps to absorb the road shocks by providing resistance to vehicle bounce and sway during weight transfer.

It will resist the vehicle dive during braking and squat during acceleration, also the rolling effect caused due to centrifugal force, is compensated to major extent.

Suspension system improves vehicle stability by better vehicle weight distribution and enhanced traction at all tires in all conditions like sudden braking and acceleration, driving in bumps or uneven surfaces and high speed cornering.

It enhances the passenger comfort by isolating the cabin from road shocks.

It also protects the vehicle components and luggage from damage or wear due to rough roads.

The suspension system is primarily a spring and a shock absorber or damper.

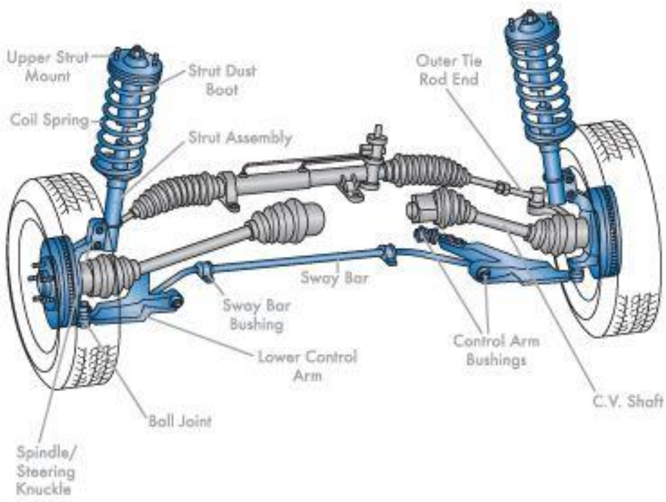


Fig 1.6 Suspension System

1.5 Transmission

Major resistances acting against the movement of any vehicle are:

a) Rolling resistance

b) Air drag

c) Gradient resistance

Rolling resistance is primarily the frictional resistance between the road surface and the tyre. It is low at good tarmacs and high at poor surfaces.

Air resistance is the resistance which arises due to the opposed effect of air density on movement of the vehicle, it is low at low speeds and high at high speeds.

However, this increase isn't linear with speed. It increases a squared component i.e. if speed doubles, the air drag increases by four times

Gradient resistance is the opposition to vehicle movement caused due to the angle at which the driving surface is inclined. Steeper the road, higher is the resistance.

The sum of all resistances requires a powertrain performance according to the dynamic condition of the vehicle. Engine has a limited range of sufficient power, torque and speed.

Hence, a system is required to adopt and adjust to power, torque and speed variations required in the actual physical conditions. It is transmission.

Functions of a transmission are:

i. Changes engine speed and torque as per user requirement
ii. Overcomes inertial force of the vehicle from standstill to motion as high torque is needed at low wheel speed.
iii. Changes relatively high speed and low torque of engine to low speed and high torque.
iv. If car is in motion and is accelerating, less torque and high power is required.

Therefore, transmission has several gears with different gear ratios.

Transmission system majorly consists of:

a. An engagement and disengagement system/ device called Clutch.
b. Torque and speed variation system/ device called Gear box.
c. One of the most important parts of transmission is the Differential.

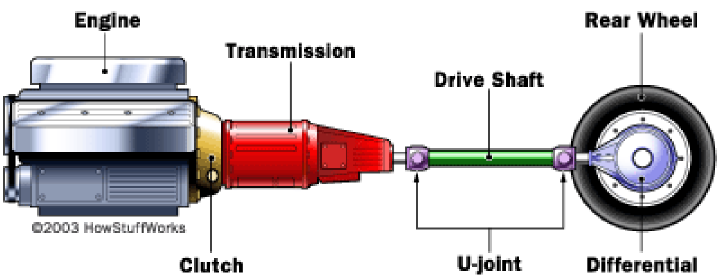


Fig 1.7 Transmission

1.6 Differential

All wheels travel at the same distance during straight driving however, the situation changes drastically during cornering.

During cornering, individual wheels have to travel different distances, Not only for left and right, but also between front and rear. For the non-driven axle, this isn't an issue as the wheels can turn freely.

But for the driven axle, this is a huge problem as with a single rigid axle the only way to equalize the difference in travel would be to let one wheel slip.

This can't be permitted as this would lead to

i. Immense load on axle
ii. Uncomfortable driving
iii. Tyre wear
iv. Loss in vehicle control

The differential allows engine torque to be applied to both drive axles, which rotate at varying speeds during cornering and while traveling over bumps and dips in the road. The differential also changes the direction of engine torque by 90° from the rotation of the driveshaft lengthwise with the vehicle.

These two purposes of a differential can be summarized as follows:

a) To change the direction of engine torque
b) To allow the drive wheels to rotate at different speeds

A differential is a mechanical addition and subtraction assembly. By splitting the engine torque to the drive wheels when the vehicle is turning a corner, the torque forces cause the side gear and pinion gears to subtract torque from one side and add torque to the opposite side.

A differential also contains two side or axle gears. The inside bore of these gears is splined and mates with splines on the ends of the axles. The differential pinion gears and side gears are in constant mesh. The pinion gears are mounted on a pinion gear shaft, which is mounted in the differential casing. As the case turns with the ring gear, the pinion shaft and gears also turn. The pinion gears deliver torque to the side gears.

Working Principle: The amount of power delivered to each driving wheel by differential is expressed as a percentage. When the vehicle moves straight ahead, each driving wheel rotates at 100% of the differential case speed. When the vehicle is turning, the inside wheel might be getting 90% of the differential case speed. At the same time, the outside wheel might be getting 110% of the differential case speed.

Differentials are of 2 types based on its construction:

1) Open differential
2) Limited slip differential(LSD)
3) Locked differential
4) Torque vectoring differential(VTD)

1.6.1 Open differential

Open differentials deliver equal torque to both wheels at all times. As the case rotates (driven by the engine through the ring and pinion gears), the cross-shaft applies a drive force to the spider gears.

The two side gears apply reaction forces that counter this drive force. Because the spider gear is free to rotate about the cross-shaft, the two reaction forces are equal.

As the side gear applies a force to the spider gear, the spider gear applies an equal and opposite force to the side gear. It is this force, on the side gear, that supplies the torque to the axle that drives the wheel. Because the force on each side gear is equal, the torque supplied to each wheel is also equal. This is true regardless of whether one wheel is rotating faster than the other or at the same speed.

While traveling straight-ahead, assuming that a vehicle has equal traction at both wheels, differential action does not occur. In other words, when traveling straight on smooth road surfaces the ring gear, carrier, and the drive axles are traveling at the same speed; they rotate as a unit and the spider gears are not rotating on the cross shaft.

While traveling around corners, the ability of the differential pinion gears to spin on their shafts allows each axle to rotate at a different speed. Case speed is always equal to the average speed of the two side gears. Since the ring gear rotates with the case and each side gear rotates with its axle; when a vehicle corners, the outside wheel gains the same number of RPMs that the wheel on the inside loses, while ring gear RPM remains constant.

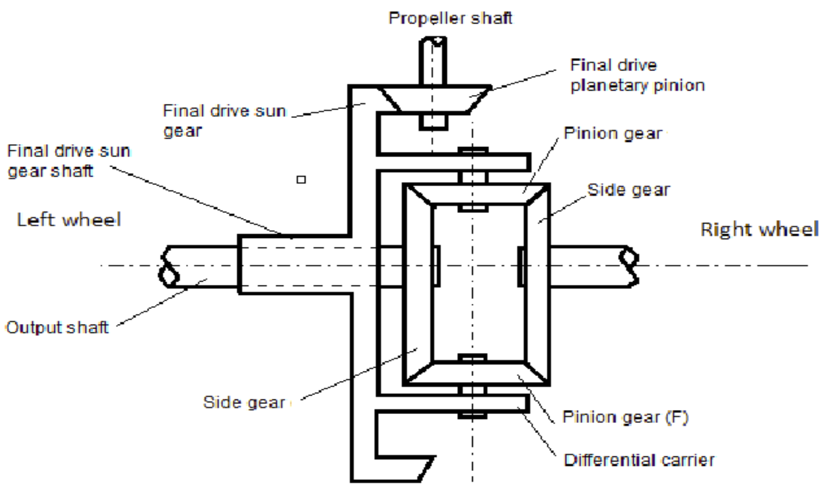


Fig 1.8 Open Differential

1.6.2 Limited-slip differential (LSD)

When a vehicle equipped with a standard differential spins a tire, the opposite wheel does not receive enough torque to move the vehicle.

To solve this problem, most manufacturers use differentials that direct more power to the side gear attached to the spinning axle.

Many differentials do this by forcing the side gear against the revolving case. This bypasses differential action, allowing the case to drive the side gear directly.

A limited-slip differential distributes torque to both wheels equally or unequally, allowing the wheels to turn at the same or at different speeds.

The only means of having the standard differential apply different amounts of torque to each axle is to have the case drive the side gear directly, bypassing the pinion gears. One means of accomplishing this is to literally "push" the side gear out of mesh with the pinion gears against the rotating case.

Limited slip differentials are of 6 types based on different designs proposed:

i. Clutch pack/ pressure disk LSD
ii. Cone clutch LSD
iii. Hydraulic locking LSD
iv. Torsen LSD
v. Viscous LSD
vi. Helical gear LSD

i. Clutch pack/ pressure ring LSD

Apart from its basic components a Limited slip differential has got a series of friction and steel plates packed between the side gear and the casing. Friction discs are having internal teeth and they are locked with the splines of the side gear. So the friction discs and the side gear will always move together.

Steels plates are having external tabs and are made to fit in the case groove. So they can rotate with the case.

If any of the clutch pack assembly is well pressed, the frictional force within them will make it move as a single solid unit. Since steel plates are locked with the case and friction discs with the side gear, in a well pressed clutch pack casing and the clutch pack will move together or motion from the casing is directly passed to the corresponding axle.

Space between the side gears is fitted with a pre-load spring. Pre load spring will always give a thrust force and will press clutch pack together.

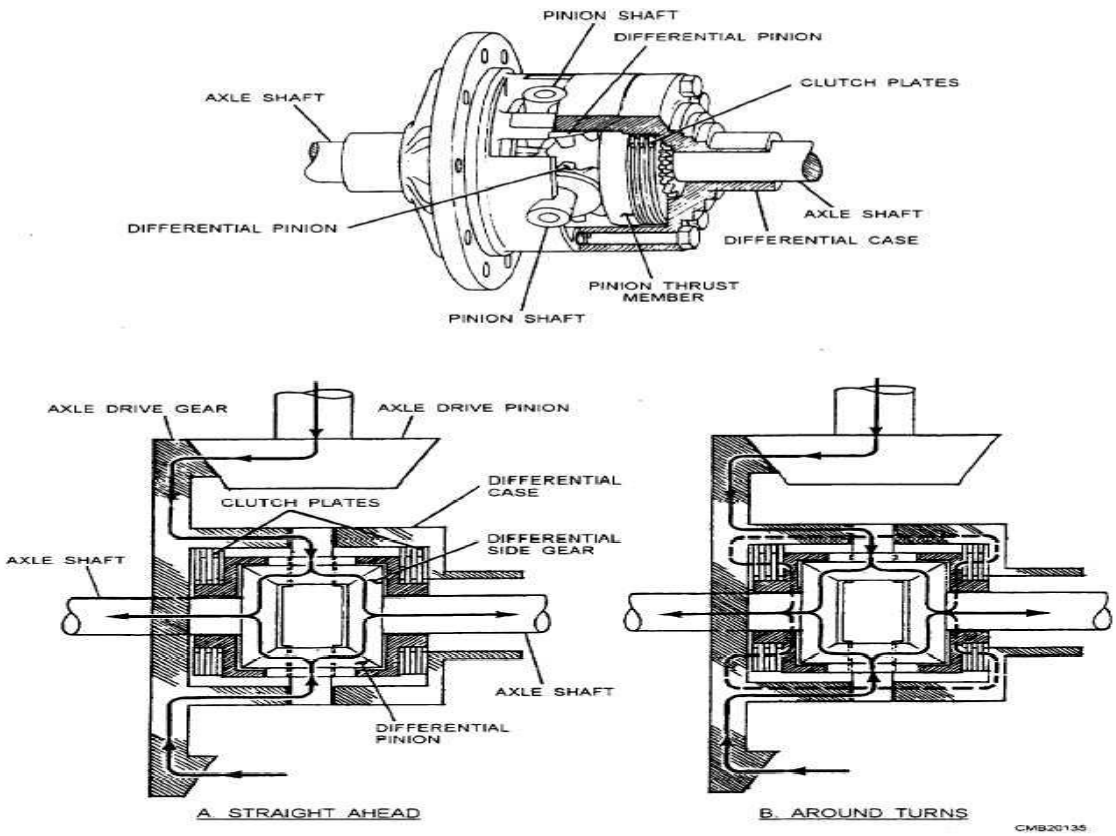


Fig 1.9 Limited Slip Differential

ii. Cone clutch LSD

A cone clutch limited slip differential uses the friction produced by cone-shaped axle gears to provide improved traction. These cones fit behind and are splined to the axle shafts.

With the axles splined to the cones, the axles tend to rotate with the differential case. Coil springs are situated between the side gears to wedge the clutches into the differential case.

Under rapid acceleration or when one wheel loses traction, the differential pinion gears drive the cones; push outward on the cone gears. This action increases friction between the cones and case, driving the wheels with even greater torque.

When a vehicle goes around a comer, the inner drive wheel must slow down. The unequal speed between the side gears will cause the side gear pinions to walk around the side gears. This walking action causes the outer axle shaft to rotate faster than the differential case. Because the cones have spiral grooves cut into their clutch surfaces, the inner cone will draw itself into the case and lock tight and the outer cone clutch will back itself out of the case. This action allows the outer drive axle to free wheel. The end result is the majority of the engine torque is sent to the inner drive wheel.

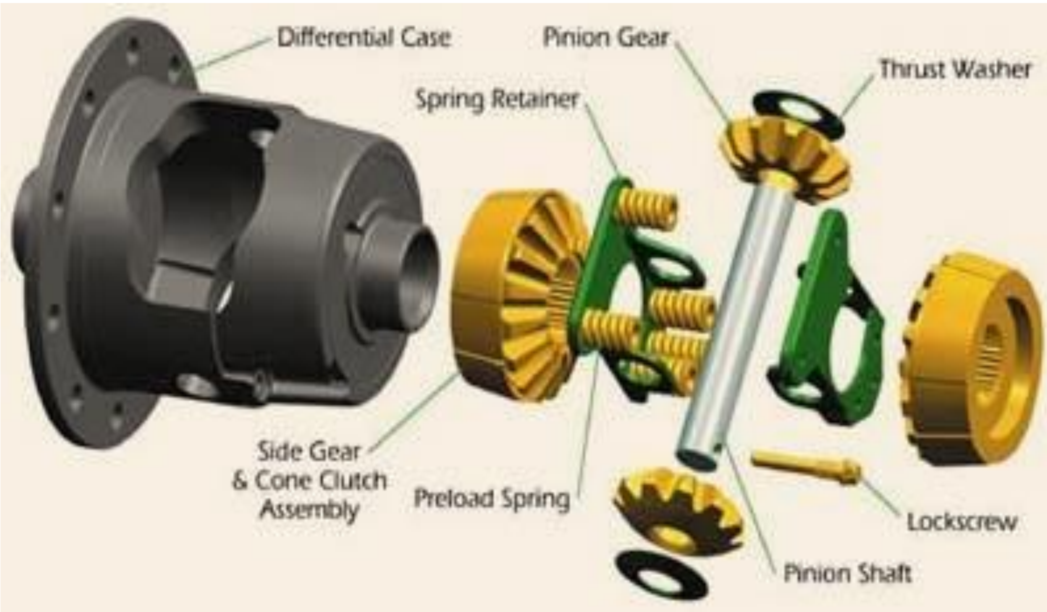


Fig 1.9 Cone Clutch LSD

iii. Hydraulic locking LSD

An improved hydraulically locking limited slip differential assembly for a drivetrain of a motor vehicle having a fluid pump external to a differential carrier and arranged for preventing slip between the wheels by selectively pressurizing a differential clutch internal to the carrier, and a controller arranged for selectively activating the fluid pump.

iv. Torsen differential

The Torsen unique torque sensing ability keeps engine power going to the ground during changing traction conditions. It functions an open differential as long as the amount of torque transmitted to each rear wheel remains equal.

When a tyre begins to lose traction, it instantly senses the change in torque being applied to the ground.

The excess torque that cannot be delivered to the ground by the tyre that is beginning to lose traction is delivered to the opposite tyre, which has better traction and can take more torque.

The Torque bias ratio (TBR) is the ratio indicating how much more torque it can send to tyre with more available traction, than is used by the tyre with less traction. This represents the locking effect of the differential.

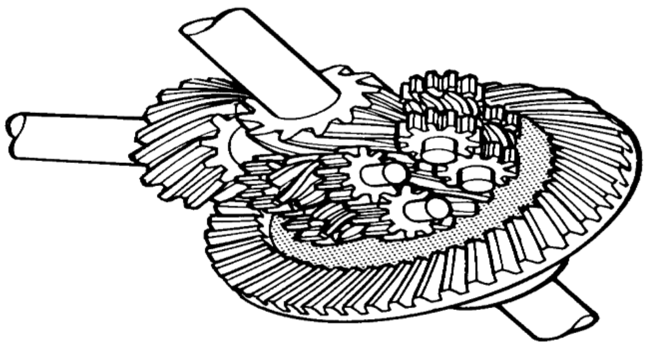


Fig 1.10 Torsen Differential

v. Viscous LSD

Viscous limited slip differentials use a viscous coupling that allows for torque to transfer to the wheel with more gripping action. It is an alternative method to a clutch pack differential, though by design it is not as effective at locking the two drive shafts.

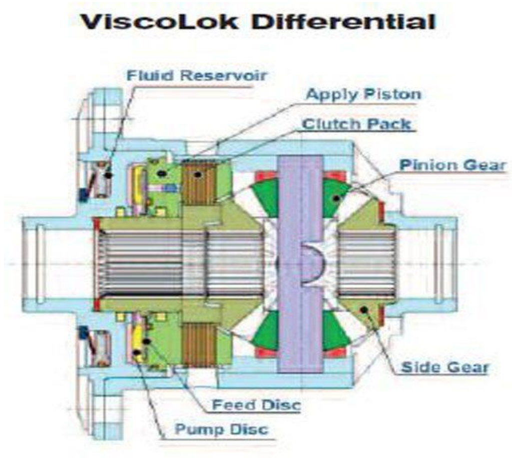


Fig 1.11 Viscous LSD

vi. Helical gear LSD

It maximizes wheel traction and enhances driving and handling characteristics. It operates as a standard or open differential under normal driving conditions, allowing one wheel to spin faster or slower as necessary.

When a wheel encounters a loss of traction or the terrain changes, the gear separation forces take effect and transfer torque to the high-traction wheel. The helical-shaped gears mesh with increasing force until wheel spin is slowed or completely stopped. When the vehicle exits the low traction situation, the differential resumes normal operation.

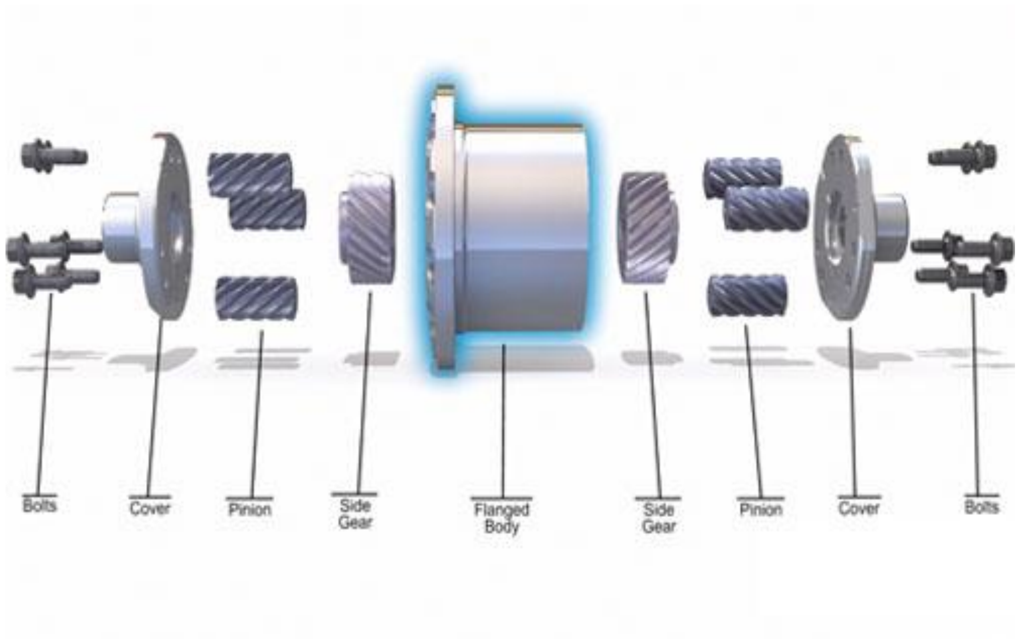


Fig 1.12 Helical Gear LSD

1.6.3 Locked differential

A locking differential, differential lock, diff lock or locker is a variation on the standard automotive differential. A locking differential may provide increased traction compared to a standard or "open" differential by restricting each of the two wheels on an axle to the same rotational speed without regard to available traction or differences in resistance seen at each wheel.

A locking differential is designed to overcome the chief limitation of a standard open differential by essentially "locking" both wheels on an axle together as if on a common shaft. This forces both wheels to turn in unison, regardless of the traction available to either wheel individually.

When the differential is unlocked (open differential), it allows each wheel to rotate at different speeds (such as when negotiating a turn), thus avoiding tire scuffing. An open (unlocked) differential always provides the same torque (rotational force) to each of the two wheels, on that axle.

So although the wheels can rotate at different speeds, they apply the same rotational force, even if one is entirely stationary, and the other spinning i.e. equal torque but unequal rotational speed.

By contrast, a locked differential forces both left and right wheels on the same axle to rotate at the same speed almost under all circumstances, regardless of traction differences on either wheel.

Therefore, each wheel can apply as much rotational force as the traction under it will allow, and the torques on each side-shaft will be unequal i.e. unequal torque but equal rotational speeds. Exceptions apply to automatic lockers.

A locked differential can provide a significant traction advantage over an open differential, but only when the traction under each wheel differs significantly.

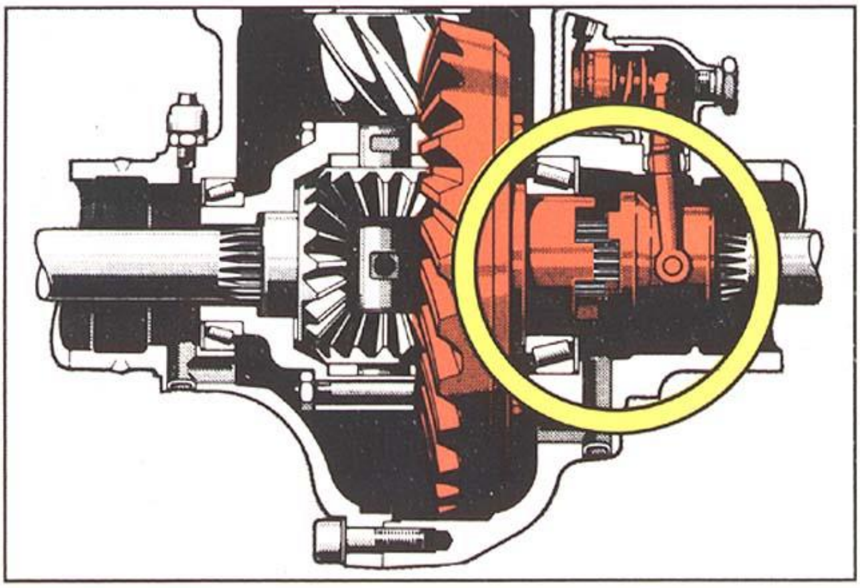


Fig 1.13 Locked Differential

1.6.4 Torque vectoring differential

The main goal of torque vectoring is to independently vary torque to each wheel. Differentials generally consist of only mechanical components.

A torque vectoring differential requires an electronic monitoring system in addition to standard mechanical components. This electronic system tells the differential when and how to vary the torque.

Due to the number of wheels that receive power, a front or rear wheel drive differential is less complex than an all-wheel drive differential. The impact of torque distribution is the generation of yaw moment arising from longitudinal forces and changes to the lateral resistance generated by each tyre. Applying more longitudinal force reduces the lateral resistance that can be generated. The specific driving condition dictates what the trade-off should be to either damp or excite yaw acceleration. The function is independent of technology and could be achieved by driveline devices for a conventional powertrain, or with electrical torque sources. Then comes the practical element of integration with brake stability functions for both fun and safety.

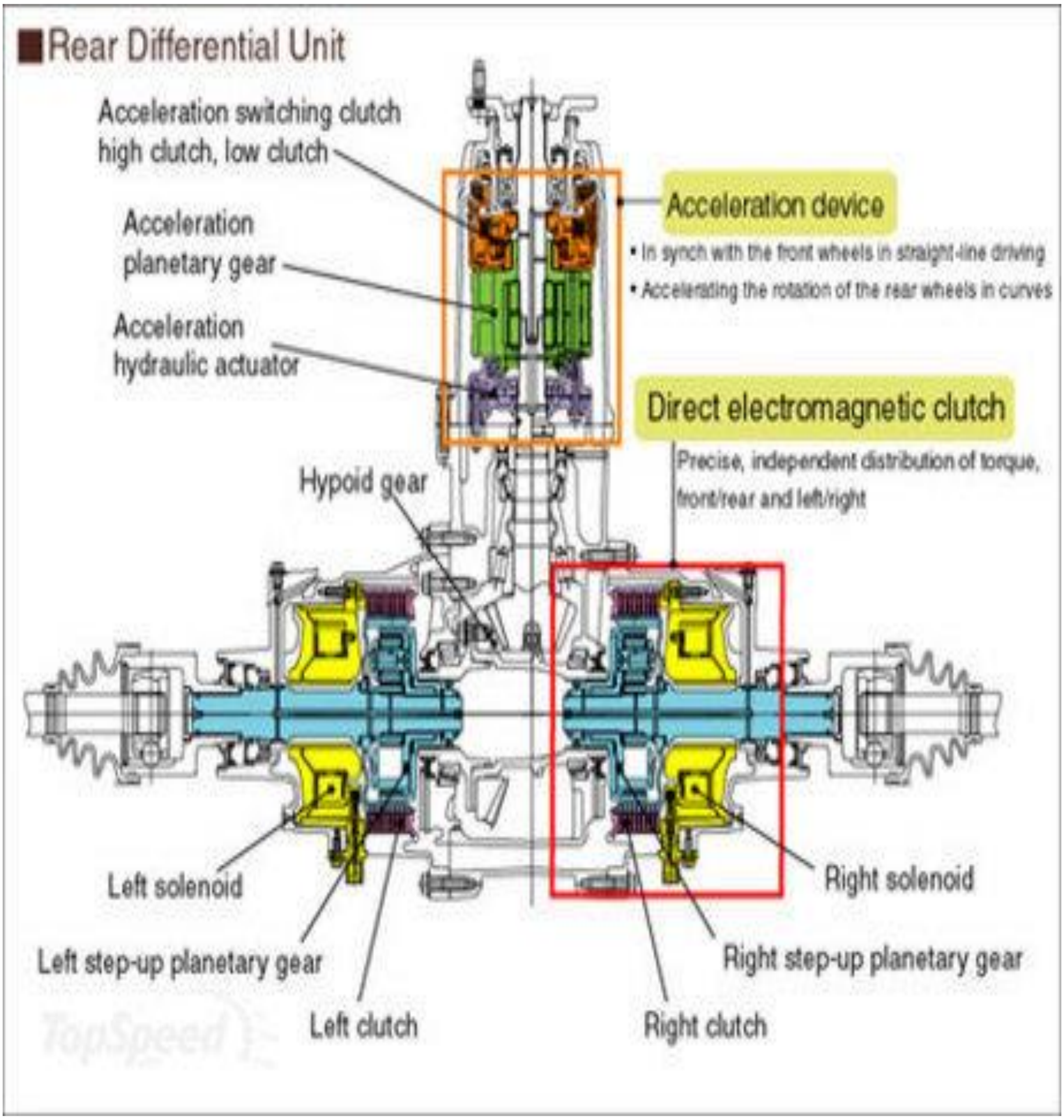


Fig 1.14 Torque Vectoring Differential (VTD)

CHAPTER –II

LITERATURE REVIEW

This chapter discusses about works carried out by various researchers on various types of automobile differentials.

Prof. A. S. Todkar, R. S. Kapare [1] designed a differential locking system that can be engaged or disengaged either manually or automatically.They concluded that automatic engagement of the differential when the loss of traction condition is encountered thereby validating the function of the automatic mode of the differential locking system and manual override using push button system for semi-automatic mode of differential locking. Prof. A V Hari babu , Dr B Durga Prasad [2] identified the true design and the extended service life, long term stability can be assured on a CATIA generated geometric model of pinion of rear axle differential and they concluded that above calculated Von-mises stresses, deformation of pinion, stress values are less than ultimate standard value of steel and deformations are very less under the maximum torsional moment. Therefore, the design is safe in condition.

Prof .Amir Khan, Sankalp Verma [3] obtained the performance parameters of modified automobile central differential and to compare with Transfer gear box and they concluded that by Using the FEA output a design was created that met all the stipulated functional requirements and possessed a reasonable factor of safety while still saving significant mass and rotational inertia as compared to the old differential or transfer gearbox. Prof .Daniel et al.[4] carried out mechanically design and analysis on assembly of gears in a differential gearbox when they transmit power at different speeds and obvserved the structural analysis results, using Aluminum alloy the stress values are within the permissible stress value. So using Aluminum Alloy is safe with decreased vibrations and increased mechanical efficiency.

Prof .Shashank et al.[5] analyzed the differential gears assembly and its housing for the vibrational effect on a system in which the life of the gears is determined within different frequency range and they designed 3-D deformable-body model of differential gearbox and its housing was developedthrough SOLIDWORKS. The results obtained were then compared with the AGMA theoretical stress values.

The results are in good congruence with the theoretical values, which suggest that the model designed is correct N.Vijayababu, CH.Sekhar [6] had conducted analysis to verify the best material for the gears in the gear box at higher speeds by analyzing stress, displacement and also by considering weight reduction and observed to verify the best material for the gears in the gear box at higher speeds by analyzing stress, displacement and also by considering weight reduction.

Mayank bansal et al. [7] studied the structural and dynamic behaviour of the gears in the differential gearbox assembly made up of the different composite material as compared to the conventional metallic materials and concluded the structural and dynamic behaviour of the gears in the differential gearbox assembly made up of the different composite material as compared to the conventional metallic materials. Iresh Bhavi et al. [8] conducted the static, fatigue, modal and harmonic analysis of Spiral bevel gears pair used as drive pinion and crown wheel in the differential gearbox and obtained the static, fatigue, modal and harmonic analysis of Spiral bevel gears pair used as drive pinion and crown wheel in the differential gearbox

Markus et al. [9] compared the static, fatigue, modal and harmonic analysis of Spiral bevel gears pair used as drive pinion and crown wheel in the differential gearbox and conducted the integrated control strategy with an eLSD is however the solution with the most advantages when combining all results and is therefore also the best solution. Richard A James[10] designed a lightweight aluminum differential housing to replace the cast-iron housing used in the Torsen® T-1 and concluded that cast-iron housing of the Torsen T- 1 differential was evaluated and redesigned as a two-piece aluminum unit.

G. Srikanth Reddy, M. Promod Reddy[11] designed and analised gears assembly in differential gear box and concluded that By theoretical comparison between steel and Ni-Cr steel, Ni-Cr steel is better in differential gear box manufacturing because of its high strength. Though the cost of Ni-Cr is high, it has long life compared to Steel. Sarika S. Bhatt [12] determined the deflection and to estimate the stress distribution in the housings and A successful understanding of the characteristic structural behavior of the gearbox housing under different loading conditions is attained by conducting the finite element analysis. The analysis serves as the groundwork to improve the design of the gearboxes.

Muhammad Safuan Bi Md Salleh[13] carried out mechanically design and analysis on assembly of gears in gearbox and concluded that the cast iron material is replaced to Aluminum Alloy for reducing weight

Ameya M. Mahadeshwar et al. [14] designed and analyised a Differential anti reverse mechanism and produced DARM with the provision of the engagement and disengagement of the mechanism. Nilton Raposo et al. [15] designed a differential unit and analyze using FEM methods and illustrated the entire design methodology into the differential and driveline assembly. Suraj Aru et al [16] designed a cage-less differential and analyze its use on an All-Terrain Vehicle (ATV)and illustrated the entire design methodology of a cage less differential.

Ronak P et a.l [17] developed theoretical model of bevel gear and to determine the effect of meshing gear tooth stresses by taking material case hardened alloy steel (15Ni4Cr1) and conclude that margin of safety is high even though after reducing 2 tooth on pinion. Reduction of teeth also reduces the weight of the pinion. Karande Mahesh et al. [18] conducted study and experiment with differential locking system and obtained Results from the experimentation predict the effective locking which is required by the differential when the vehicle wheel stuck in to any obstacle like mud.

Aleks Vrcek [19] investigated molybdenum based friction material for clutch application and concluded that Molybdenum coating shows stable friction characteristics for wet clutch applications in combination with a suitable lubricant, when the surface of the coating is additionally surface finished reducing surface roughness. Amol B Rindhe et al.[20] had studied its design procedure along with finite element analysis some important parameter will be obtained and noticed that there are various forces, torque, angles of inclination, material's effect on components of Drive Shaft and trying to make it better by changing some parameter or material to composite material and test the same.

S. P. Chaphalkar Subhash Khetre [21] analyzed and design the rear axle shaft of steel material and studied the modal analysis , using the numerical (FEA) approach, it has been found that the relative error between these two approaches are very minute. RahI Jain and Pratik Goyal[22] analyzed the stresses in mating teeth of spur gear to find maximum contact stress in the gear teeth

It is concluded that the gearbox can bear the maximum stresses and deformation well under the safety zone. As a result, based on this finding if the contact stress minimization in the primary concern and if the large power is to be transmitted then spur gears with the higher model is preferred.

N. Siva Teja[23] performed mechanical design of differential gear box and analysis of gears in gear box and found that both grey cast iron and aluminum alloy are preferable for performing the application of differential gearbox in automobiles. But, when it comes to weight for light utility vehicles Aluminum Alloy is preferred. T.Loknadh , DR. P.S.S. vasu[24] designed and structurally analyze of crown wheel gear used in gear box and to verify the best material (steel and Aluminum Alloys) for the crown wheel gear in the gear box at higher speed by analyzing static and dynamic conditions and design of crown wheel gear is safe under the above operating loading conditions. But the steel material model factor of safety (2.9) is better than the aluminum material model FOS (1.64). Hence we can conclude that the steel material design is better than the aluminum material design.

Rikard Mäki [25] investigated on transmission fluids used in wet clutches in all-wheel drive systems, featuring a wet multi-plate clutch and concluded that thermal effects have a significant influence on the torque transferred by the differential under limited slip conditions. It is therefore necessary to have a temperature dependent boundary friction model. Gabriele Barbaraci et al. [26] designed a MR sports car differential , with the aim of the total mass Reduction and torque maintenance and concluded that Possibilities offered by MR fluids for the implementation of a limited slip differential are studied in this paper.

A.J. Tremlett et al. [27] computed the minimum time optimal torque bias profile through a lane change man oeuvre and compare with traditional open and fully locked differential strategies, in addition to considering related vehicle stability and agility metrics and described minimum time optimal control method for determining the ideal torque biasing profile of an LSD.considering A RWD racing saloon racing vehicle , carrying out a traditional lane change man oeuvre

John W. H. Trout [28] analyzed and test the Quaife torque biasing differential and investigated the operation of the Quaife differential, modeled its Mathematics, and determined how the preload torque affects its biasing ratio.

S. H. Gawande[29] performed mechanical design of crown wheel and pinion in differential gear box of MFWD (FWA) Axle and concluded that this work detailed manual and computer aided designing of crown gear and pinion is carried out and is a safe design. Marc ollé Bernades [30] derived a model for an open differential in order to qualify and quantify the difference of torque between left and right side and concluded that this analysis has given an understanding of which are the causes of the torque steer effects due to the differential in those driving situations when it affects the torque distribution.

From the above literature, it is understood that earlier researches carried out works on analysis related to von-Misses stresses, deformation, vibrational effects, harmonic effect, deflection, maximum contact stresses, fatigue & static analysis, torque distribution and replaced conventional materials of the differential with new materials for better results.

The present project work comprises design of cage-less centrally suspended limited slip differential using Solid works 2017 and calculating the forces involved in limited slip differential. Carried out Static structural analysis to find von-Moisses stresses and deformation of all differential gears by replacing differential gear materials to 20MnCr5 using ANSYS 14.5.

CHAPTER III

DESIGN CALCULATIONS FOR CAGE-LESS LIMITED SLIP DIFFERENTIAL

This chapter describes about design procedure in calculating various forces involved in Cage-less limited slip differential.

The following notations are followed in the design calculations.

Number of teeth on pinion	- Z_p
Number of teeth on gear	- Z_g
Pitch angle of pinion	- γ
Module	- m
Diameter of pinion	- D_p
Diameter of gear	- D_g
Pressure angle	- α
Face width	- b
Mean radius	- R_m
Tangential load	- P_t
Radial load	- P_r
Axial/Thrust load	- P_a
Torque	- M_t
Pitch of gear	- Γ
Formative spur gear tooth	-Z_p'
Ultimate strength	-σ_u
Allowable bending stress	-σ_b
Working stress	-σ_w
Beam strength	-S_b
Lewis form factor	-Y
Cone distance	-A_o
Wear strength	-S_w
Ratio factor	-Q
Material constant	-K
Brinell hardness number	-BHN
Sum of errors between	
Two meshing gear teeth	-e
Deformation factor	-C
Effective load	-P_{eff}
Dynamic load	-P_d
Speed of pinion	-N_p
Factor of safety against bending failure	-FS_b
Factor of safety against pitting failure	-FS_w
Factor of safety	-Fs

3.1 Design of differential gears

3.1.1 For Final drive gear and ring gear

While designing Final drive gear, the following flow is adopted.
Final drive gear specifications

m=5, Z_p=40, Z_g=60, $\alpha=20^0$, b=25mm

Manual design calculation is as follows:
$\tan\gamma = Z_p/Z_g$
$\Rightarrow \gamma = \tan^{-1}(40/60)$
$\Rightarrow \gamma = 33.69^0$

Module (m) = D_p/Z_p
$\Rightarrow D_p = m \times Z_p$
$\Rightarrow D_p = 5\times40$
$\Rightarrow D_p$=200mm

Mean radius of Final drive gear (R_m)= $[D_p/2 - b\sin\gamma/2]$
$= [200/2 - 25\sin(33.69^0)/2]$
= 93.066mm

Components of tooth force on gears are as follows
Tangential load (P_t) = M_t/R_m
$= 122\times10^3/93.066$
= 1310.89N
Radial load (P_r) =$P_t.\tan\alpha.\cos\gamma$
$=1310.89\times\tan(20^0)\cos(33.69^0)$
=396.99N

Axial (or)
Thrust load (P_a) = $P_t.\tan\alpha.\sin\gamma$
$= 1310.89\times\tan(20^0)\sin(33.69^0)$
= 264.66N

For gear pitch angle (Γ) = 90-γ
= 90-33.69
$= 56.31^0$

Mean Radius of Ring gear (R_m) = $[D_p/2 - b\sin\gamma/2]$
$= [300/2 - 40\sin(56.31^0)/2]$
=133.358mm

Design for Ring gear is same as that of Final drive gear as they are meshing.

Torque on Ring gear (M_t) = $P_t\times R_m$
$=1310.89\times133.358$
= 174.817N-m

3.1.2 For two Crown gears and two Side gears

While designing Crown gear, the following flow is adopted.

Crown gear specifications

$m=5$, $Z_p=20$, $Z_g=20$, $\alpha=20^0$, $b=25$

Manual design calculation is as follows:

From the ring gear, same torque will be transmitted to the crown gear.

Module (m) = D_p/Z_p

$=> D_p = m \times Z_p$

$=> D_p = 20\times5$

$=> D_p = 100mm$

$Tan\gamma = Z_p/Z_g$

$=> \gamma = Tan^{-1}(20/20)$

$=> \gamma = 45^0$

Mean radius (R_m)= $[D_p/2 - b\sin\gamma/2]$

$= [100/2 - 25\sin(45^0)/2]$

$= 41.16116mm$

Tangential load (P_t) = $M_t/2\times R_m$

$= 174.817\times10^3/2\times41.16116$

$= 2126.127N$

Radial load (P_r) = $P_t.\tan\alpha.\cos\gamma$

$= 2126.127\times\tan(20^0)\times\cos(45^0)$

$= 547.237N$

Axial (or) Thrust load (P_a) = $P_t.\tan\alpha.\sin\gamma$

$= 2126.127\times\tan(20^0)\times\sin(45^0)$

$= 547.237N$

Design of the Side gears is same as that of Crown gear.

Torque on crown gear (M_t) = $P_t\times(R_m/2)$

$= 2126.127\times(41.16116/2)$

$= 43.756N\text{-}m$

3.2 Calculations for Beam and Wear strength of differential gears

3.2.1 For Final drive gear and Ring gear

Torque transmitted (M_t) = 122N-m

$Z_p=40$, $Z_g=60$, $b=25$, $N_p=4000rpm$

Material taken for all gears is 20MnCr5

$\sigma_y=850Mpa$, $\sigma_u=1300Mpa$, BHN=600

Allowable bending stress (σ_b) = $\sigma_u/3$

$=1300/3$

$=433.33$Mpa

Pitch angle (γ)=$\tan^{-1}(40/60)$

$=33.69^0$

Formative spur gear tooth (Z_p')=$Z_p/\cos\gamma$

$=40/\cos(33.69^0)$

$=46.073$

Diameter of pinion (D_p) $=m\times Z_p$

$=5\times40$

$=200$mm

Diameter of gear (D_g) $=m\times Z_g$

$=5\times60$

$=300$mm

Cone distance (A_o) =Sqrt$[(D_p/2)^2+(D_g/2)^2]$

=Sqrt$[(100)^2+(150)^2]$

$=316.227$mm

By interpolating from table 3.1

$Y=0.399+[(0.468-0.399)\times(48.073-45)]/(50-45)$

$=0.4045$

Beam strength (Sb) $=m\times b\times\sigma_b\times Y[1-(b/A_o)]$

$= 5\times25\times433.33\times0.4045[1-(25/316.227)]$

$=20{,}178.32$N

Ratio factor (Q) = $[(2\times Z_g)/(Z_g+Z_p\tan\gamma)]$

$= [(2\times60)/(60+\{40\times\tan(33.69^0)\}]$

$=1.3846$

Material constant (K) $=0.16\times(BHN/100)^2$

$=0.16\times(600/100)^2$

$=5.76$

Wear strength (Sw) = $(0.75\times b\times Q\times D_p\times K)/\cos\gamma$

$= (0.75\times b\times Q\times D_p\times K)/\cos(33.69^0)$

$= 35954.53$N

From table 3.3, for module 5,e=0.0125

For 20^0 full depth gears, C=11400N-mm^2 & P_t=1310.89N

Velocity (v) $= (\pi \times D_p \times N_p)/(60\times10^3)$
$= (\pi\times200\times4000)/(60\times10^3)$
$= 41.887$m/s

Dynamic load (Pd) $= [21\times v\times(\{C\times e\times b\}+P_t)]/[(21\times v)+\text{Sqrt}(\{C\times e\times b\}+P_t)]$
$= \dfrac{[21\times41.887\times(\{11400\times0.0125\times25\}+1310.89)]}{[(21\times11400)+\text{Sqrt}(\{11400\times0.0125\times25\}+1310.89)]}$
$= 4515.07$N

Effective load (P_{eff}) $= (C_s\times P_t)+P_d$
$= (1\times1310.89)+4515.07$
$= 5825.962$N

Factor of safety against bending failure (FS_b) $= S_b/P_{eff}$
$=20{,}178.32/5825.963$
$=3.46$

Factor of safety against pitting failure (FS_w) $= S_w/P_{eff}$
$=35{,}954.53/5825.962$
$=6.1714$

3.2.2 For two Crown gears and two Side gears

$Z_p=20$, $Z_g=20$, $m=5$, $\alpha=20^0$, $b=25$
BHN=600, $\sigma_u=1300$

Pitch angle (γ) $= \tan^{-1}(20/20)$
$=45^0$

Formative spur gear tooth (Z_p') $=Z_p/\text{Cos}\gamma$
$=20/\text{Cos}(45^0)$
$=28.28^0$

Allowable bending stress (σ_b) $=\sigma_u/3$
$=1300/3$
$=433.33$

Diameter of pinion (D_p) $= m\times Z_p$
$= 5\times20$
$= 100$mm

Diameter of gear (D_g) $= m\times Z_g$
$=5\times20$
$=100$mm

Cone distance (A_o) $= \text{Sqrt}[(D_p/2)^2+(D_g/2)^2]$
$= \text{Sqrt}[(100/2)^2+(100/2)^2]$

$= 70.71mm$

From table 3.1, Y=0.32

Beam strength (S_b) = $m \times b \times \sigma_b \times Y[1-(b/A_o)]$

$= 5 \times 25 \times 43.33 \times 0.32[1-(25/70.71)]$

$=11{,}204.9296N$

Ratio factor (Q) = $[(2 \times Z_g)/(Z_g+Z_p \tan\gamma)]$

$= [(2 \times 20)/(20+20 \times \tan 45^0)]$

$=1$

Material constant (K) = $0.16 \times (BHN/100)^2$

$= 0.16 \times (600/100)^2$

$= 5.76$

Wear strength (S_w) = $(0.75 \times b \times Q \times D_p \times K)/\cos\gamma$

$= (0.75 \times 25 \times 1 \times 100 \times 5.76)/\cos 45^0$

$=10{,}800N$

Velocity (v) = $(\pi \times D_p \times N_p)/(60 \times 10^3)$

$= (\pi \times 100 \times 4000)/(60 \times 10^3)$

$= 20.94m/s$

From table 3.2

e=0.0125mm, C=11400N/mm^2 ,P_t=2126.127N

Dynamic load (Pd) = $[21 \times v \times (\{C \times e \times b\}+P_t)]/[(21 \times v)+Sqrt(\{C \times e \times b\}+P_t)]$

$$= \frac{[21 \times 20.94 \times (\{11400 \times 0.0125 \times 25\}+2126.127)]}{[(21 \times 20.94)+Sqrt(\{11400 \times 0.0125 \times 25\}+2126.127)]}$$

$= 4855.786N$

Effective load (P_{eff}) = $(C_s \times P_t)+P_d$

$= (1 \times 2126.127)+4855.786$

$= 6981.913N$

Factor of safety against bending failure (FS_b) = S_b/P_{eff}

$= 11{,}204.9296/6981.913$

$= 1.6048$

Factor of safety against pitting failure (Fs_w) = S_w/P_{eff}

$= 10{,}800/6981.913$

$=1.5468$

3.3 Calculations for Factor of safety

3.3.1 For two Crown & two Side gears

$P_{eff}=m\times b\times\sigma_w\times Y\times[1-(b/A_o)]$

$=>4855.786 =5\times25\times\sigma_w\times0.320\times[1-(25/70.710)]$

$=> \sigma_w=187.78$Mpa

Factor of safety (Fs)= σ_u/ σ_w

=1300/187.788

=6.922

3.3.2 For Final & Ring gear

$P_{eff}=m\times b\times\sigma_w\times Y\times[1-(b/A_o)]$

$=>5825.962 =5\times25\times\sigma_w\times0.409\times[1-(25/316.22)]$

$=> \sigma_w=125.26$ Mpa

Factor of safety (Fs)= σ_u/σ_w

=1300/125.26

=10.37

Table 3.1 Lewis form factor

Sl.No	No. of teeth (z)	Lewis form factor (Y)
1	15	0.289
2	20	0.320
3	30	0.358
4	40	0.389
5	50	0.408
6	60	0.421

Table 3.2 Error for two meshing gears

Module (m) (mm)	Class 1	Class 2	Class 3
Up to 4	0.050	0.025	0.0125
5	0.056	0.025	0.0125
6	0.064	0.030	0.0150
7	0.072	0.035	0.0170
8	0.080	0.038	0.0190
9	0.085	0.041	0.0205
10	0.090	0.044	0.0220

CHAPTER-IV

MODELING AND ANALYSIS

This chapter describes about generating 3D geometry using SOLID WORKS and evaluations of forces using ANSYS

4.1 Introduction to Solidworks

Solidworks main idea is user to create drawing directly in 3D or solid form. From this solid user can assemble it directly on their workstation checking clashes and functionality of it. Creating drawing is pretty easy just drag and drop the solid to drawing block.

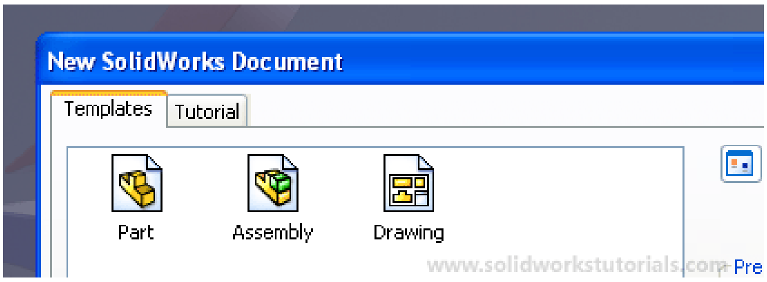


Fig.4.1 New Solidworks document

4.1.1 Part

Part is created by sketch

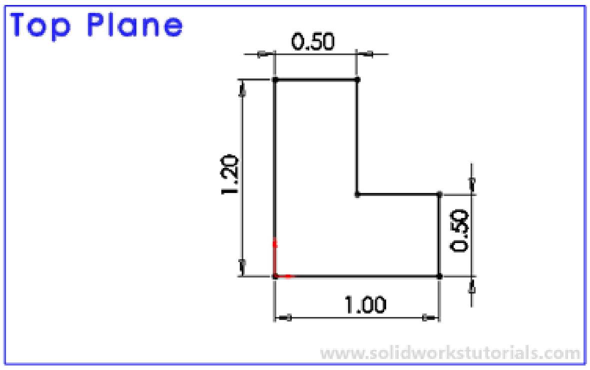


Fig.4.2 2D drawing

Sketch is the base to define your part, form and feature.

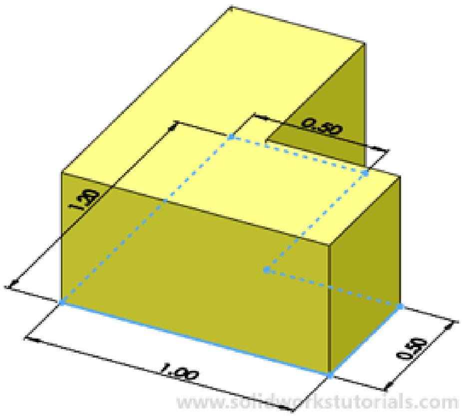


Fig.4.3 3D modeling

Before you start creating sketches you must select plane or face where the sketch will be place on.

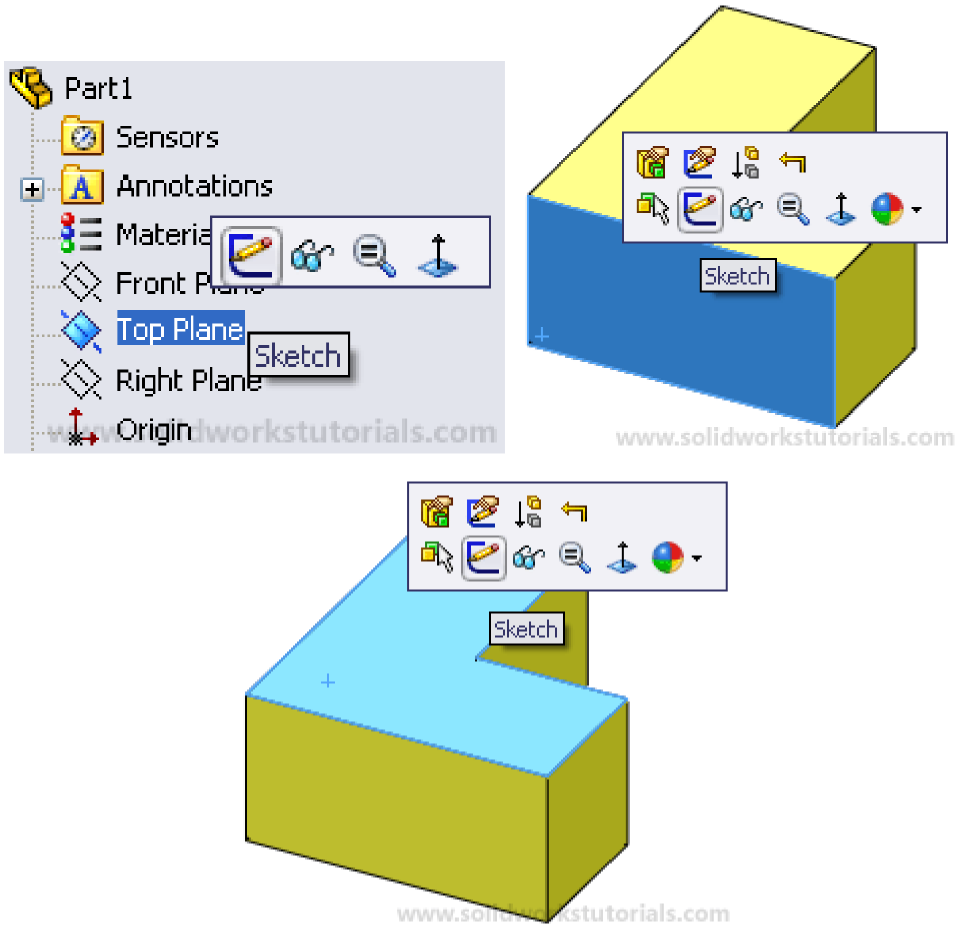


Fig. 4.4 ,4.5 .4.6 Sketching

After select plane or face the sketch will be, sketch on it!

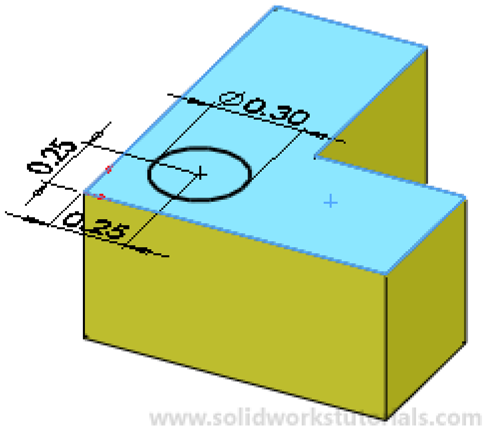


Fig.4.7 Smart dimensioning

When you done with sketch, adding features it is your next step. Select Feature>Extruded Cut

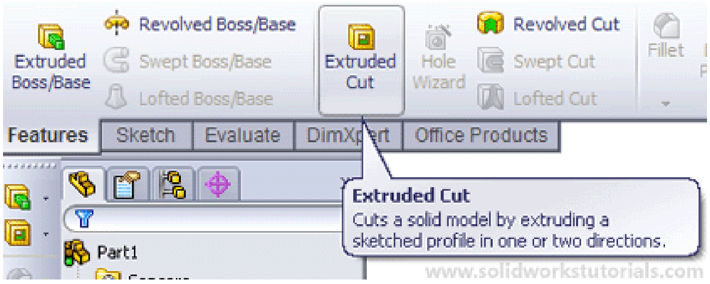


Fig.4.8 Extruded cut

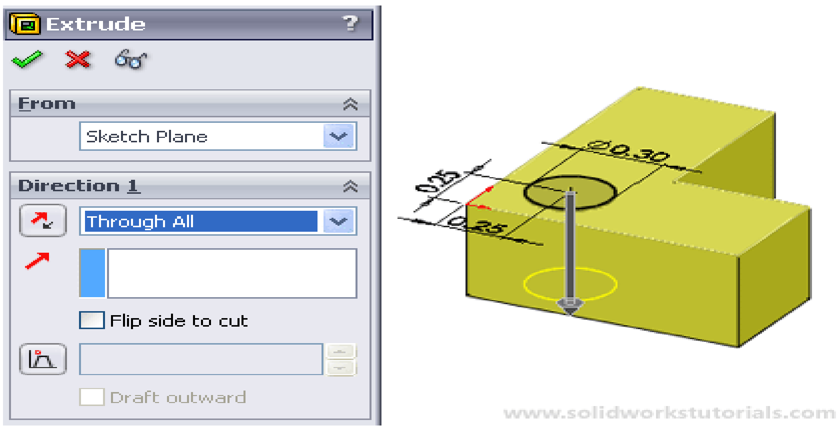


Fig.4.9, 4.10 Through all Hole

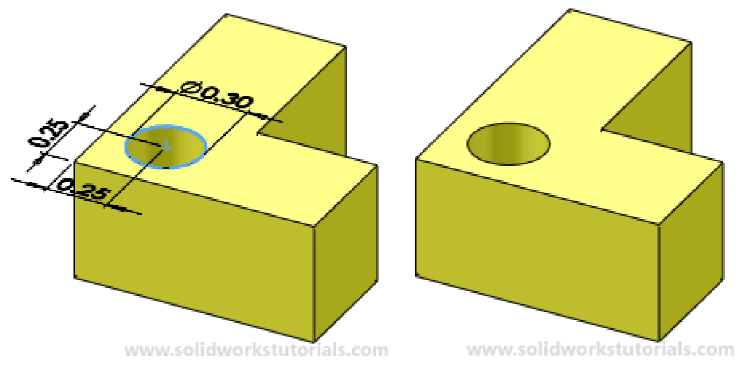


Fig. 4.11 & 4.12 Through all Hole

4.1.2 Assembly

Assembly is how all parts works together in assembly, checking for clashes and it functionality. First all parts inserted in assembly by Insert Component tool**.** When all parts inserted into workspace, Mate is command to define how parts mate with each other

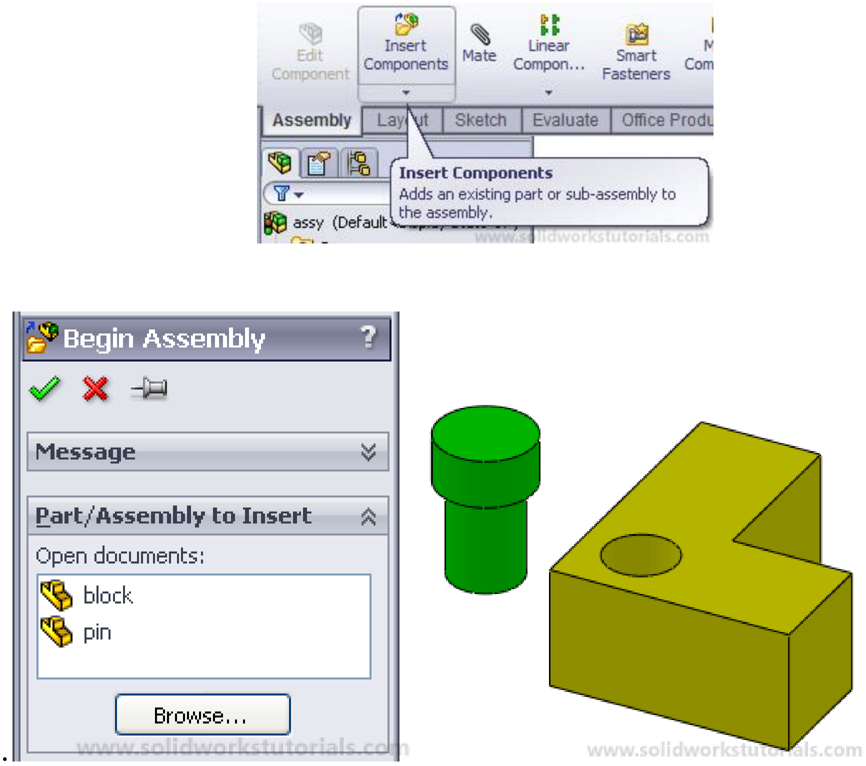


Fig.4.13, 4.14, 4.15 Assembling

Let’s mate this block and pin together, click Mate and select pin face & hole face,OK.

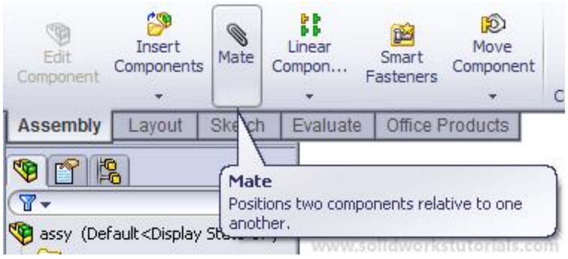
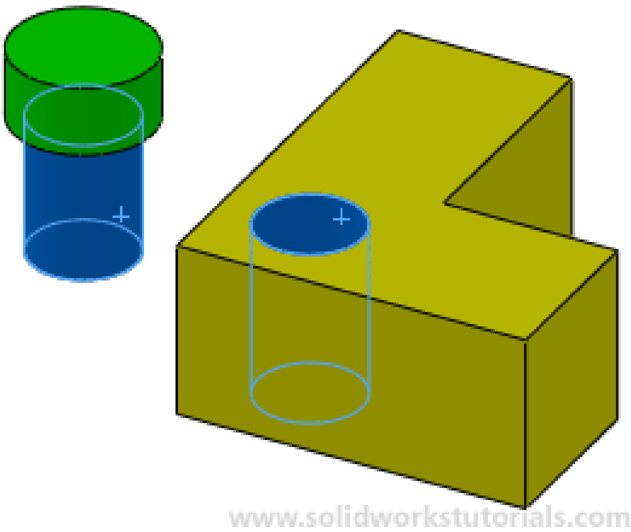


Fig.4.16 Mating



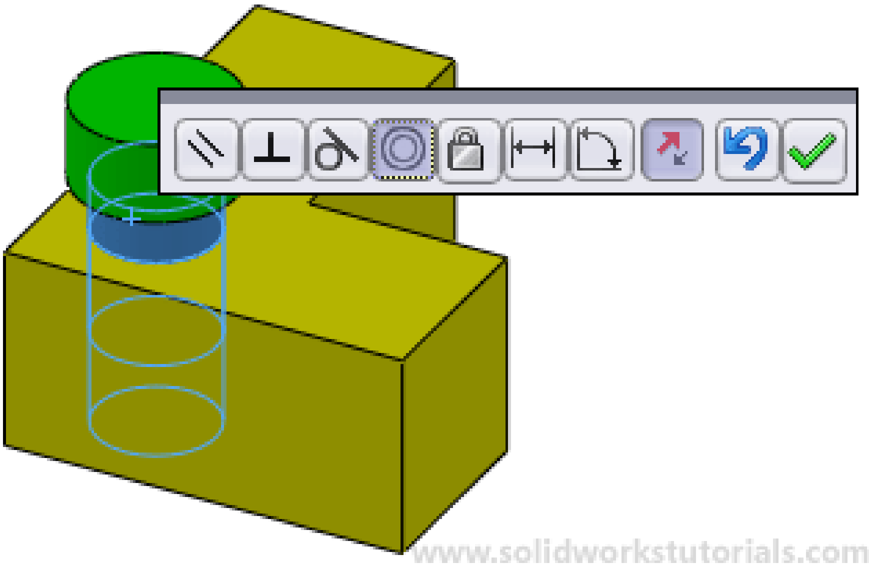


Fig.4.17 & 4.18 Assembly

4.1.3 Drawing

Drawing is use for detailing part by adding dimension to it. To create drawing first you need to select drawing block.

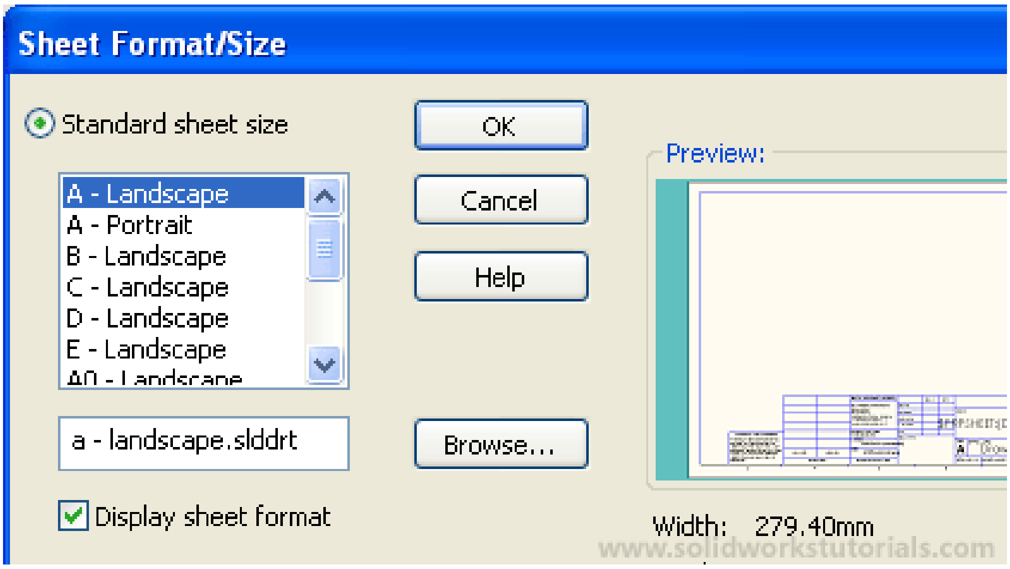


Fig.4.19 Sheet format

When block inserted, select click view palette to add drawing view.

Fig.4.20 View palette

Choose the part you wish to make drawing.

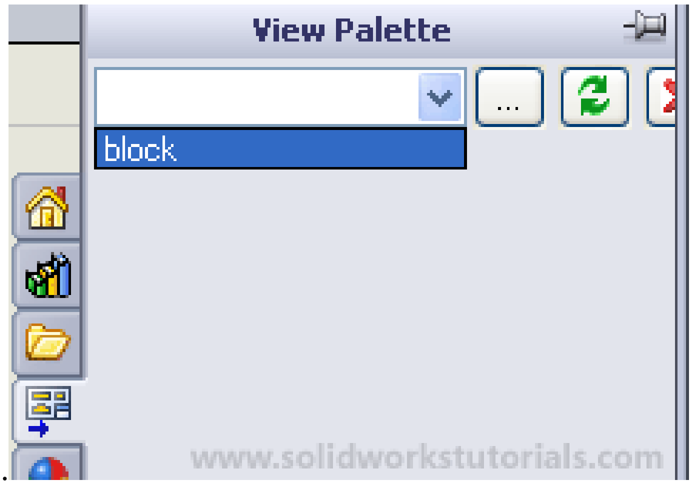


Fig.4.21 Blocking of view palette

Now just drag and drop the part view on drawing block and add dimensions.

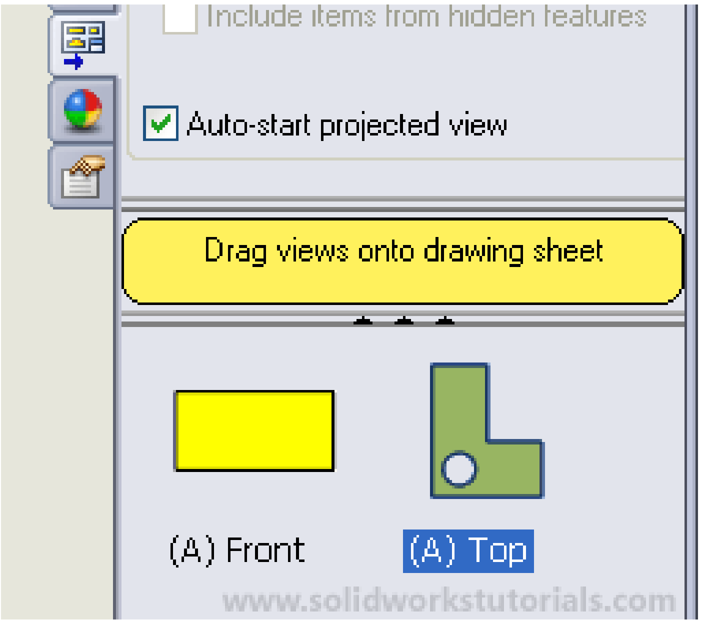


Fig.4.22 Views

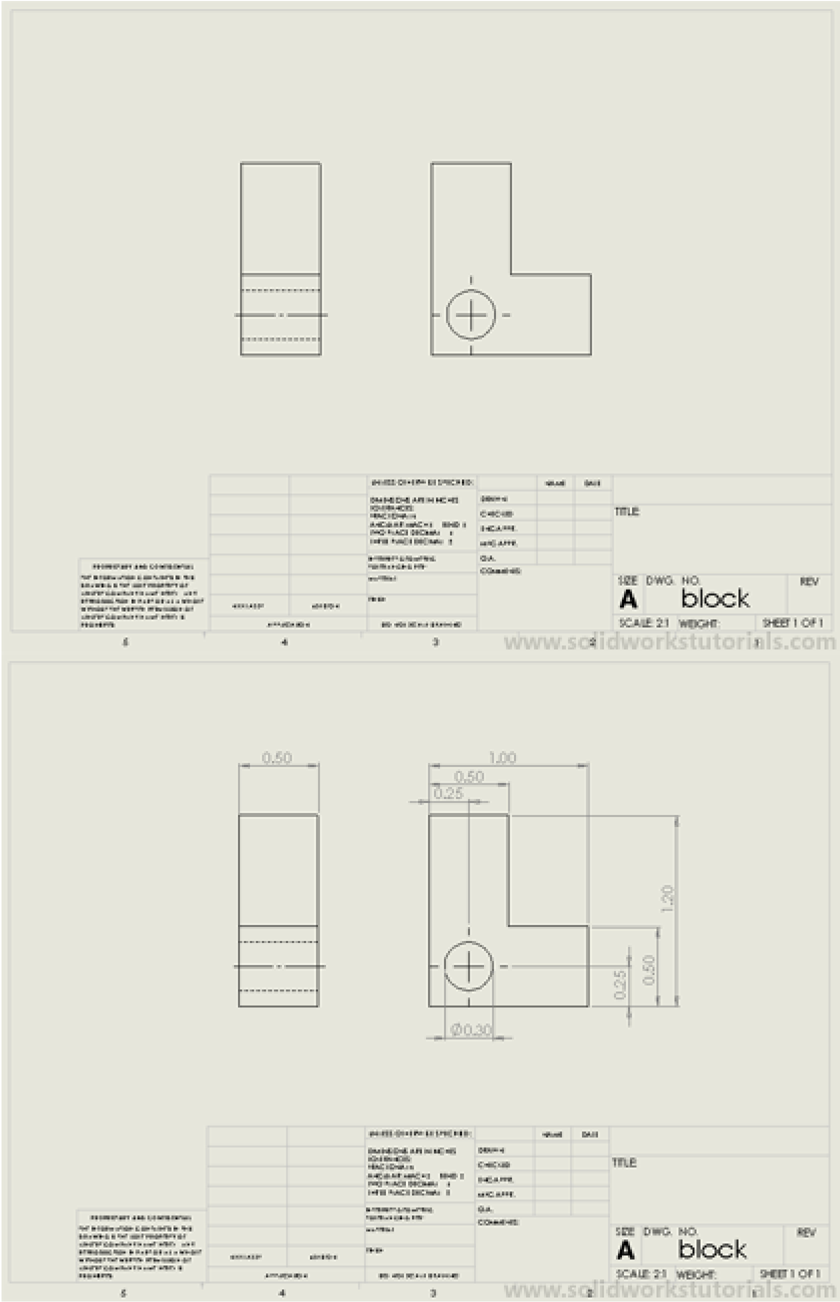


Fig.4.23 Drawing

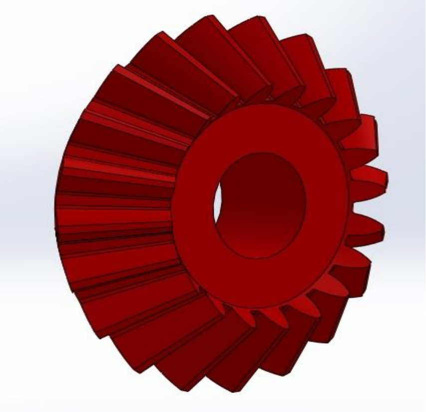
Summary

Solidworks works by it user creating part in 3D or solid form. Three solidworks components are Part, Assembly and Drawing. Part define by it sketch and selected feature. Assembly is how all parts assemble in one unit, parts assemble by user adding mate between parts. Drawing is for detailing and adding dimensions to part.

4.2 Modelling Of Cage-Less Limited Slip Differential

Cage-less limited slip differential consists of the following parts.

4.2.1 Crown gear

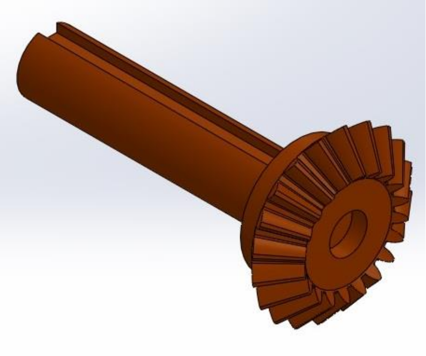


Quantity:2

These gears are mounted onto the Crosspin over bush and transmits power fromthe Ring gear to Side gears.

Material: 20MnCr5

4.2.2 Side gear

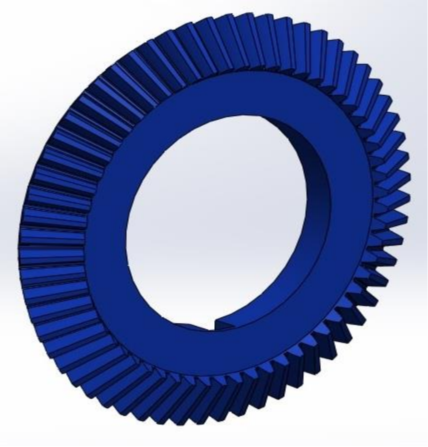


Quantity:2

These gears are directly to the wheels and transmit power from crown gears to the wheels.

Material:20MnCr5

4.2.3 Ring gear

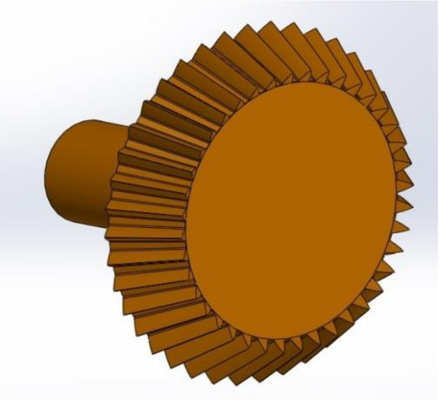


Quantity:1

This gear is connected to Crosspin connecting crown gears and transmits power from the final drive gear to the crown gears.

Material:20MnCr5

4.2.4 Final drive gear

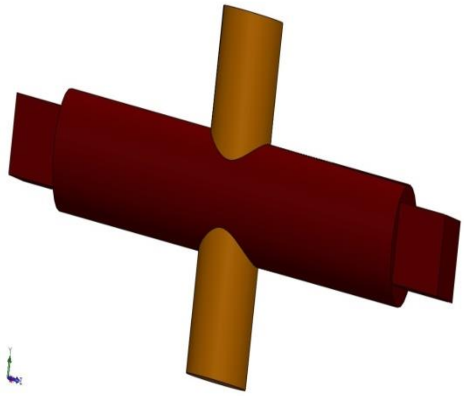


Quantity:1

This gear is directly connected to the final drive and transmits power from the final drive to the ring gear.

Material:20MnCr5

4.2.5 Crosspin

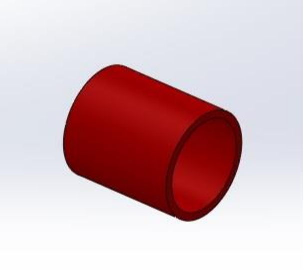


Quantity:1

It keeps the crown gears and side gears intact.

Material:20MnCr5

4.2.6 Bush



Quantity:2

It connects crown gears to the attachment.

Material:20MnCr5

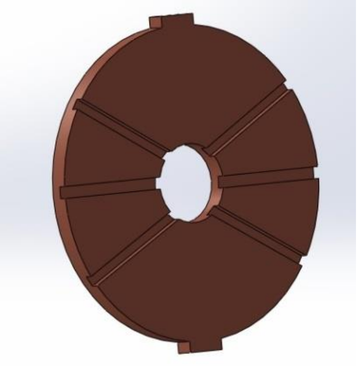
4.2.7 Gear Housing

Quantity:2

It houses all the crown gears and side gears and keeps all gears in position.

Material:20MnCr5

4.2.8 Pressure plate



Quantity:4

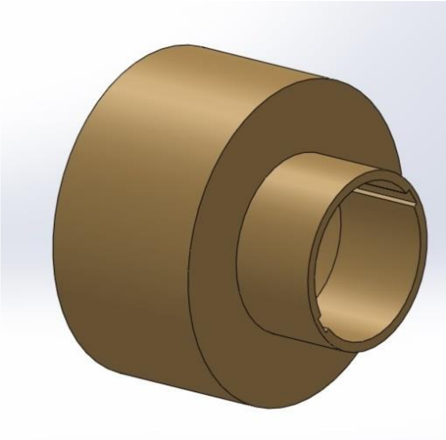
It applies pressure on the friction plate engaging all plates in position and retains oil.

4.2.9 Friction plate

Quantity:5

It acts as a friction lining and keeps plates engaged.

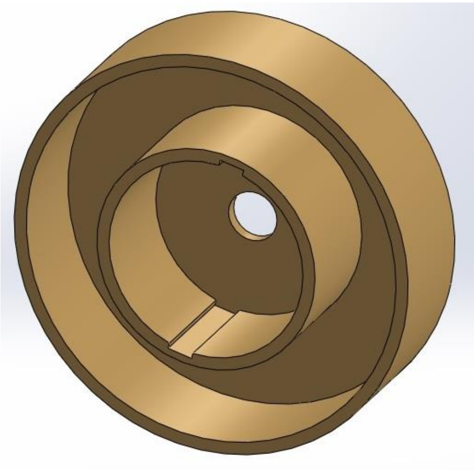
4.2.10 Housing 1



Quantity:1

It houses entire assembly.

4.2.11 Housing 2



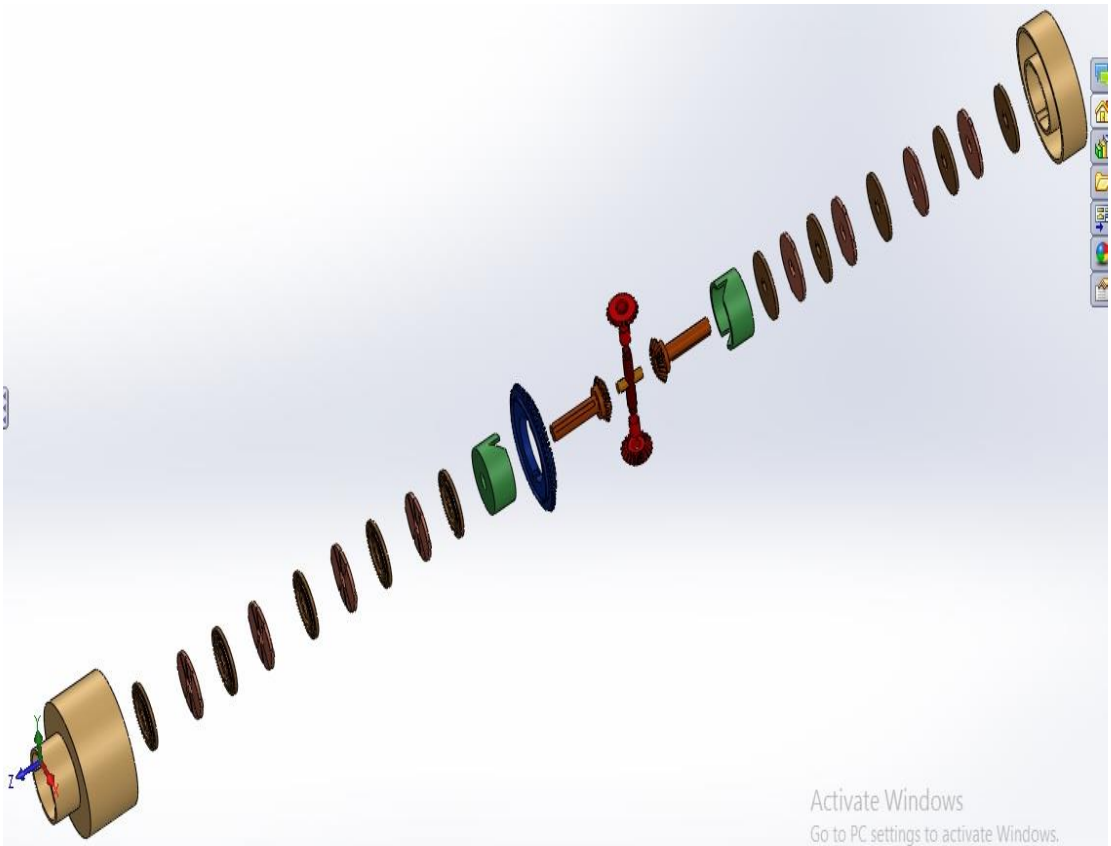
Quantity:1

It houses entire assembly

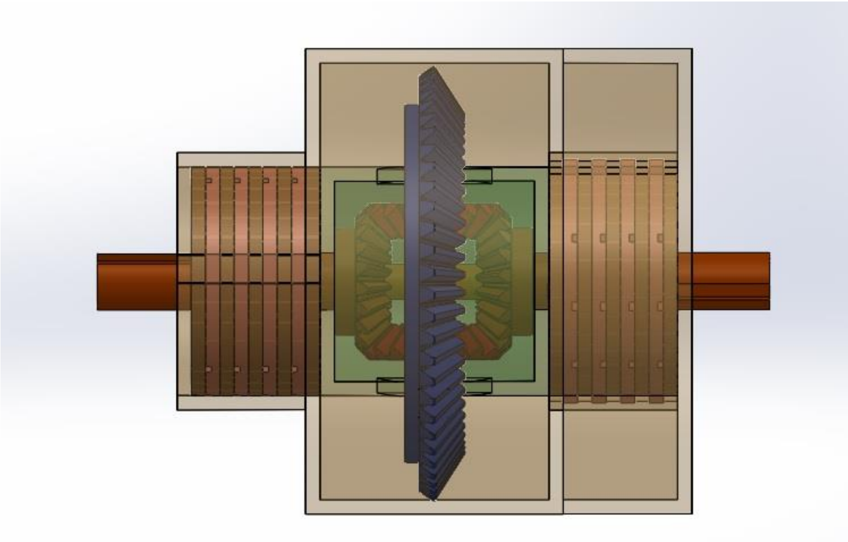
Table 4.1 Properties of alloy used for differential gears

Sl.No	Property	Value
1.	Name of the alloy	20MnCr5
2.	Model type	Linear elastic isotropic
3.	Default Failure Criterion	Maximum von-Mises
4.	Yield Strength	850 Mpa
5.	Tensile Strength	1300 Mpa
6.	Elastic Modulus	2E+5 Mpa
7.	Poisson's Ratio	0.285
8.	Mass Density	8000Kg/m^3
9.	Shear Modulus	7.7821E+4 Mpa
10.	Bulk modulus	1.5504E+5 Mpa

4.3 Exploded View Of The Differential



4.4 Assembled View Of The Differential



4.5 Introduction To FEA

Finite Element Analysis (FEA) was first developed in 1943 by R. Courant, who utilized the Ritz method of numerical analysis and minimization of variational calculus to obtain approximate solutions to vibration systems. Shortly thereafter, a paper published in 1956 by M. J. Turner, R. W. Clough, H. C. Martin, and L. J. Topp established a broader definition of numerical analysis. The paper centered on the "stiffness and deflection of complex structures".

FEA consists of a computer model of a material or design that is stressed and analyzed for specific results. It is used in new product design, and existing product refinement. A company is able to verify a proposed design will be able to perform to the client's specifications prior to manufacturing or construction. Modifying an existing product or structure is utilized to qualify the product or structure for a new service condition. In case of structural failure, FEA may be used to help determine the design modifications to meet the new condition.

There are generally two types of analysis that are used in industry: 2-D modeling, and 3-D modeling. While 2-D modeling conserves simplicity and allows the analysis to be run on a relatively normal computer, it tends to yield less accurate results.

3-D modeling, however, produces more accurate results while sacrificing the ability to run on all but the fastest computers effectively. Within each of these modeling schemes, the programmer can insert numerous algorithms (functions) which may make the system behave linearly or non-linearly. Linear systems are far less complex and generally do not take into account plastic deformation. Non-linear systems do account for plastic deformation, and many also are capable of testing a material all the way to fracture.

FEA uses a complex system of points called nodes which make a grid called a mesh. This mesh is programmed to contain the material and structural properties which define how the structure will react to certain loading conditions. Nodes are assigned at a certain density throughout the material depending on the anticipated stress levels of a particular area. Regions which will receive large amounts of stress usually have a higher node density than those which experience little or no stress. Points of interest may consist of: fracture point of previously tested material, fillets, corners, complex detail, and high stress areas. The mesh acts like a spider web in that from each node, there extends a mesh element to each of the adjacent nodes. This web of vectors is what carries the material properties to the object, creating many elements.

A wide range of objective functions (variables within the system) are available for minimization or maximization:

- Mass, volume, temperature
- Strain energy, stress strain
- Force, displacement, velocity, acceleration
- Synthetic (User defined)

There are multiple loading conditions which may be applied to a system.

Some examples are shown:

- Point, pressure, thermal, gravity, and centrifugal static loads
- Thermal loads from solution of heat transfer analysis
- Enforced displacements
- Heat flux and convection

Each FEA program may come with an element library, or one is constructed over time. Some sample elements are:

- Rod elements
- Beam elements
- Plate/Shell/Composite elements
- Shear panel
- Solid elements
- Spring elements
- Mass elements
- Rigid elements
- Viscous damping elements

4.5.1 Types of Engineering Analysis

Structural analysis consists of linear and non-linear models. Linear models use simple parameters and assume that the material is not plastically deformed. Non-linear models consist of stressing the material past its elastic capabilities. The stresses in the material then vary with the amount of deformation as in.

Vibrational analysis is used to test a material against random vibrations, shock, and impact. Each of these incidences may act on the natural vibrational frequency of the material which, in turn, may cause resonance and subsequent failure.

Fatigue analysis helps designers to predict the life of a material or structure by showing the effects of cyclic loading on the specimen. Such analysis can show the areas where crack propagation is most likely to occur. Failure due to fatigue may also show the damage tolerance of the material.

Heat Transfer analysis models the conductivity or thermal fluid dynamics of the material or structure. This may consist of a steady-state or transient transfer.

Steady-state transfer refers to constant thermo properties in the material that yield linear heat diffusion.

4.5.2 Results of Finite Element Analysis

FEA has become a solution to the task of predicting failure due to unknown stresses by showing problem areas in a material and allowing designers to see all of the theoretical stresses within. This method of product design and testing is far superior to the manufacturing costs which would accrue if each sample was actually built and tested.

In practice, a finite element analysis usually consists of three principal steps:

a. Preprocessing: The user constructs a model of the part to be analyzed in which the geometry is divided into a number of discrete sub regions, or elements," connected at discrete points called nodes." Certain of these nodes will have fixed displacements, and others will have prescribed loads. These models can be extremely time consuming to prepare, and commercial codes vie with one another to have the most user-friendly graphical "preprocessor" to assist in this rather tedious chore. Some of these preprocessors can overlay a mesh on a preexisting CAD file, so that finite element analysis can be done conveniently as part of the computerized drafting-and-design process.

b. Analysis: The dataset prepared by the preprocessor is used as input to the finite element
code itself, which constructs and solves a system of linear or nonlinear algebraic equations

$$K_{ij}u_j = f_i$$

where u and f are the displacements and externally applied forces at the nodal points. The formation of the K matrix is dependent on the type of problem being attacked, and this module will outline the approach for truss and linear elastic stress analyses. Commercial codes may have very large element libraries, with elements appropriate to a wide range of problem types.

One of FEA's principal advantages is that many problem types can be addressed with the same code, merely by specifying the appropriate element types from the library.

c. **Postprocessing**: In the earlier days of finite element analysis, the user would pore through reams of numbers generated by the code, listing displacements and stresses at discrete positions within the model. It is easy to miss important trends and hot spots this way, and modern codes use graphical displays to assist in visualizing the results. A typical postprocessor display overlays colored contours representing stress levels on the model, showing a full field picture similar to that of photo elastic or moiré experimental results.

4.5.3 Introduction to ANSYS

ANSYS is general-purpose finite element analysis (FEA) software package. Finite Element Analysis is a numerical method of deconstructing a complex system into very small pieces (of user-designated size) called elements. The software implements equations that govern the behaviour of these elements and solves them all; creating a comprehensive explanation of how the system acts as a whole. These results then can be presented in tabulated, or graphical forms. This type of analysis is typically used for the design and optimization of a system far too complex to analyze by hand. Systems that may fit into this category are too complex due to their geometry, scale, or governing equations.

ANSYS is the standard FEA teaching tool within the Mechanical Engineering Department at many colleges. ANSYS is also used in Civil and Electrical Engineering, as well as the Physics and Chemistry departments.

ANSYS provides a cost-effective way to explore the performance of products or processes in a virtual environment. This type of product development is termed virtual prototyping.

With virtual prototyping techniques, users can iterate various scenarios to optimize the product long before the manufacturing is started. This enables a reduction in the level of risk, and in the cost of ineffective designs.

The multifaceted nature of ANSYS also provides a means to ensure that users are able to see the effect of a design on the whole behavior of the product, be it electromagnetic, thermal, mechanical etc.

4.5.4 Generic Steps to Solving any Problem in ANSYS

Like solving any problem analytically, you need to define (1) your solution domain, (2) the physical model, (3) boundary conditions and (4) the physical properties. You then solve the problem and present the results. In numerical methods, the main difference is an extra step called mesh generation. This is the step that divides the complex model into small elements that become solvable in an otherwise too complex situation. Below describes the processes in terminology slightly more attune to the software.

a. Build Geometry

Construct a two or three dimensional representation of the object to be modeled and tested using the work plane coordinate system within ANSYS.

b. Define Material Properties

Now that the part exists, define a library of the necessary materials that compose the object (or project) being modeled. This includes thermal and mechanical properties.

c. Generate Mesh

At this point ANSYS understands the makeup of the part. Now define how the modeled system should be broken down into finite pieces

d. Apply Loads

Once the system is fully designed, the last task is to burden the system with constraints, such as physical loadings or boundary conditions.

e. Obtain Solution

This is actually a step, because ANSYS needs to understand within what state (steady state, transient… etc.) the problem must be solved.

f. Present the Results

After the solution has been obtained, there are many ways to present ANSYS' results, choose from many options such as tables, graphs, and contour plots.

4.5.5 Specific Capabilities of ANSYS

a. Structural

Structural analysis is probably the most common application of the finite element method as it implies bridges and buildings, naval, aeronautical, and mechanical structures such as ship hulls, aircraft bodies, and machine housings, as well as mechanical components such as pistons, machine parts, and tools.

Static Analysis - Used to determine displacements, stresses, etc. under static loading conditions. ANSYS can compute both linear and nonlinear static analyses. Nonlinearities can include plasticity, stress stiffening, large deflection, large strain, hyper elasticity, contact surfaces, and creep.

Transient Dynamic Analysis - Used to determine the response of a structure to arbitrarily time-varying loads. All nonlinearities mentioned under Static Analysis above are allowed.

Buckling Analysis - Used to calculate the buckling loads and determine the buckling mode shape. Both linear (eigenvalue) buckling and nonlinear buckling analyses are possible.

In addition to the above analysis types, several special-purpose features are available such as **Fracture mechanics**, **Composite material analysis**, **Fatigue**, and both **p-Method and Beam analyses.**

4.6 Static Structural Analysis Of Differential Gears

A simple structural analysis is performed as the first step to see if components were structurally strong. If component fails due to loadings, then there is no need to continue any further analysis since the component isn't strong enough to be used.

The analysis of the various components of the differential was done in ANSYS 14.5 WORKBENCH for meshing as well as solving. Meshing of all the parts was done in ANSYS. The mesh is generated by using tetrahedron elements of 1 mm size. Mesh quality is further improved by using proximity and curvature function. This improves mesh density where curvature is small or edges are closed in proximity.

4.6.1 Crown gear

When designing a pair of bevel gears there are 3 forces which come into action. These are radial force (Pr), tangential force (Pt) and axial force (Pa). To analyse the bevel gears these three forces have been applied on gear's teeth. It is considered that there are 2 teeth in mesh with the bevel pinion.

After analysis in ANSYS it is found out that the maximum stress zone is at the root of the gear near the small filleted areas. The maximum stress in the component is 158.01MPa.

The displacement of the gear is near the tops of the loaded teeth. The maximum displacement in this component is 0.0126 mm. From the values of stress and displacement it is shown that the given component is safe.

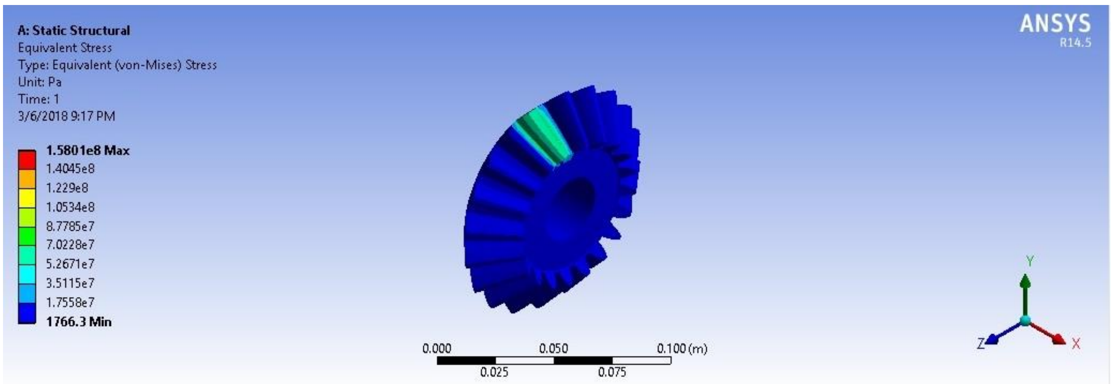


Fig.4.37 Stress analysis in ANSYS

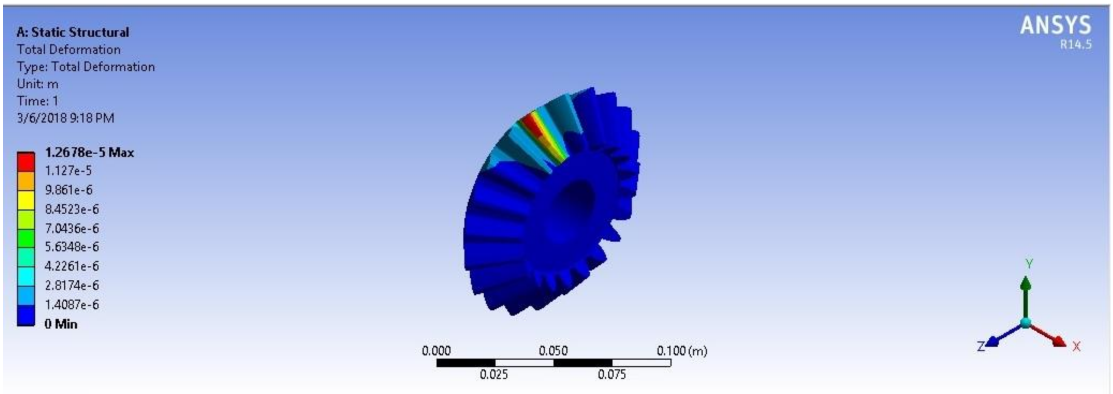


Fig.4.38 Maximum deformation in ANSYS

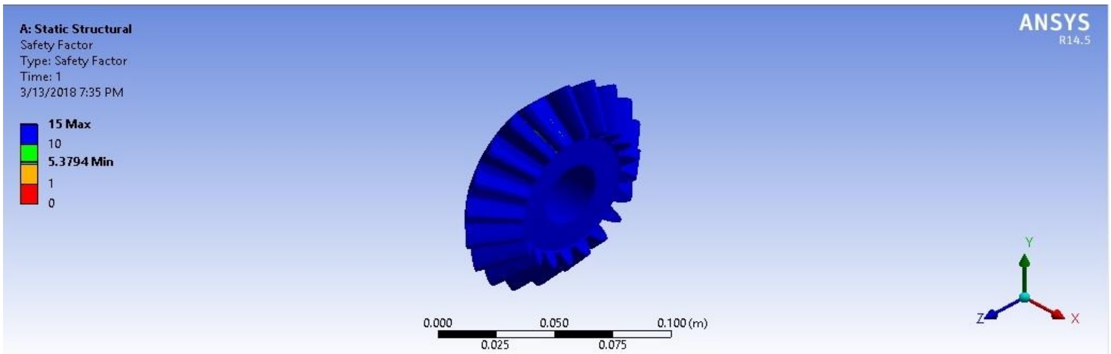


Fig.4.39 Safety factor in ANSYS

4.6.2 Side gear

Design of Side gears is similar to that of Crown gears. After analysis in ANSYS it is found out that the maximum stress zone is at the root of the gear near the small filleted areas and is 125.46MPa. The displacement of the gear is near the tops of the loaded teeth and is 0.012499mm. From the values of stress and displacement it is shown that the given component is safe.

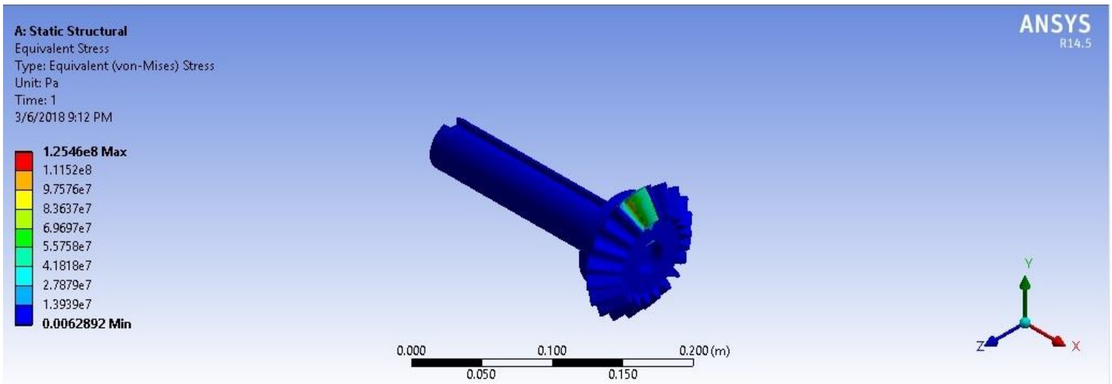


Fig.4.40 Stress analysis in ANSYS

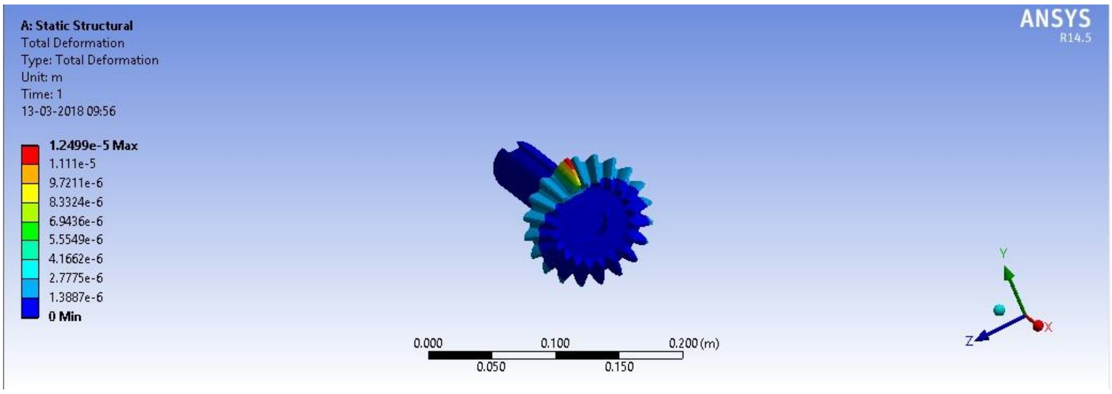


Fig.4.41 Maximum deformation in ANSYS

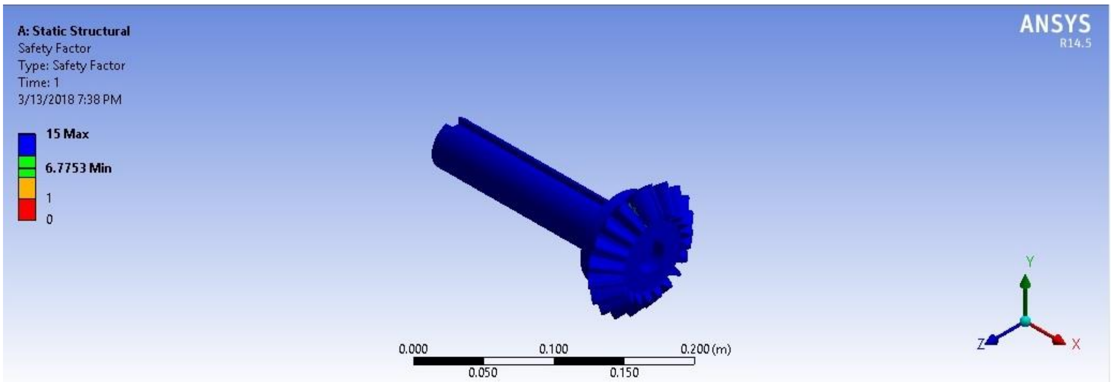


Fig.4.42 Safety factor in ANSYS

4.6.3 Ring gear

Gear teeth are subjected to both bending and wear. The section where it experiences the maximum stress is the root of the tooth. It is considered that, there are 3 teeth in contact with the mating gear.

Tangential force (Pt) because of the torque which the gear is transmitting is applied on these three teeth. Another force acting on the gear is Radial forces (Pr) which is generated due to separating force between the two meshing gears. This force acts towards the centre of the gear. Magnitude of force (Pt) and (Pr) acting on gear are taken from above designed values.

The constraints which were given are only physical constraints. The inner surface of the gear is fixed, where shaft is mounted and also the diagonally opposite surfaces of the slots which are made for fixing the pin.

The maximum stress produced in the gear is 32.477MPa and the maximum deformation is 0.0038695mm which shows that the gears are safe.

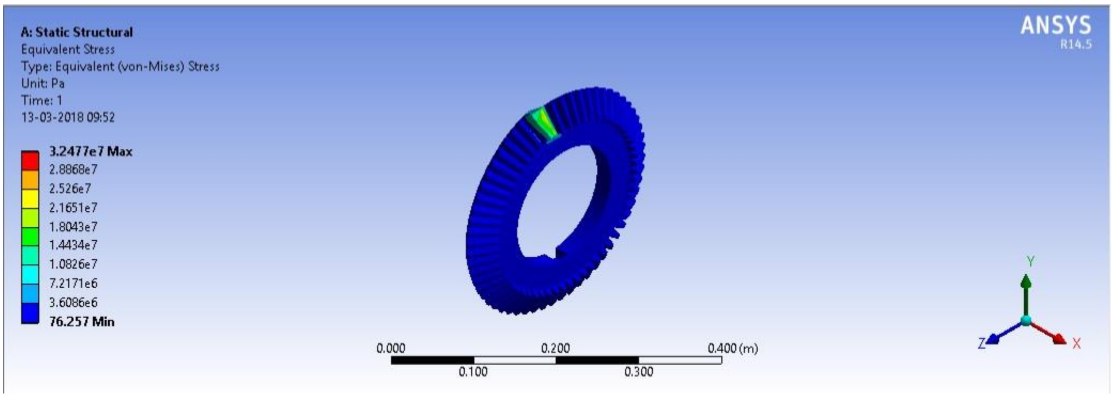


Fig.4.43 Stress analysis in ANSYS

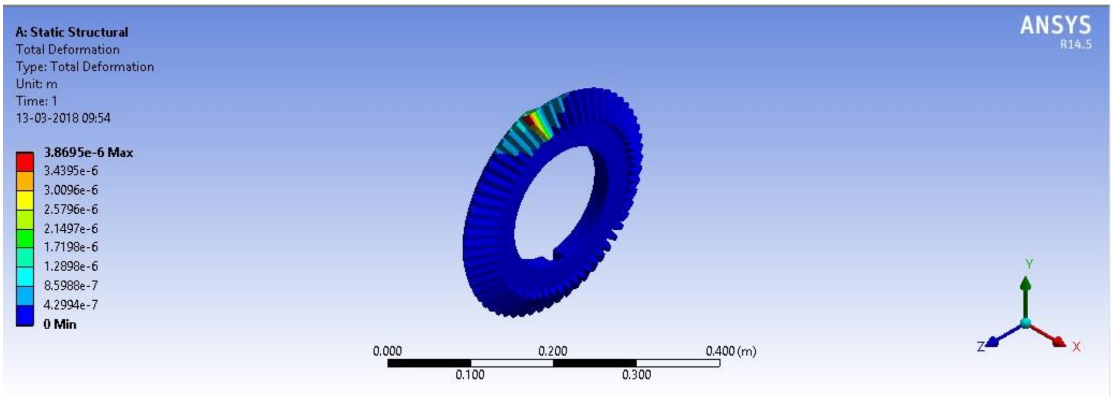


Fig.4.44 Maximum deformation in ANSYS

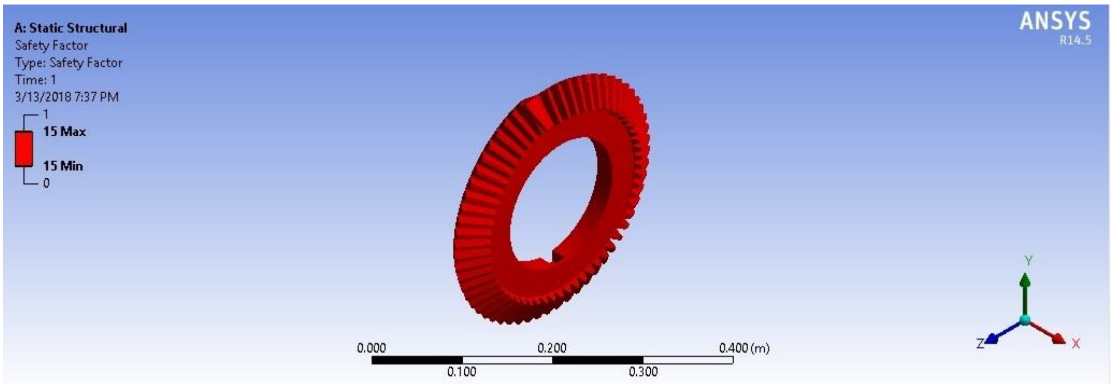


Fig.4.45 Safety factor in ANSYS

4.6.4 Final drive gear

Designing of Side gears is similar to that of Crown gears and is found out that the maximum stress zone is at the root of the gear near the small filleted areas.The maximum stress in the component is 22.066MPa.

The displacement of the gear is near the tops of the loaded teeth. The maximum displacement in this component is 0.0182mm. From the values of stress and displacement it is shown that the given component is safe.

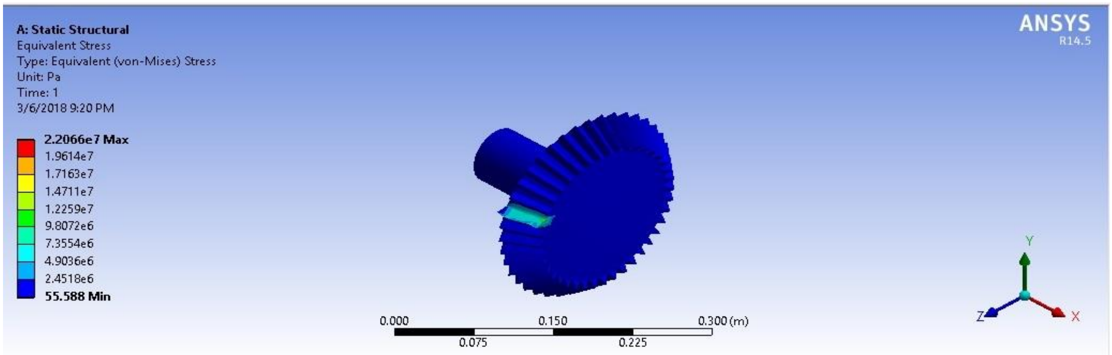


Fig.4.46 Stress analysis in ANSYS

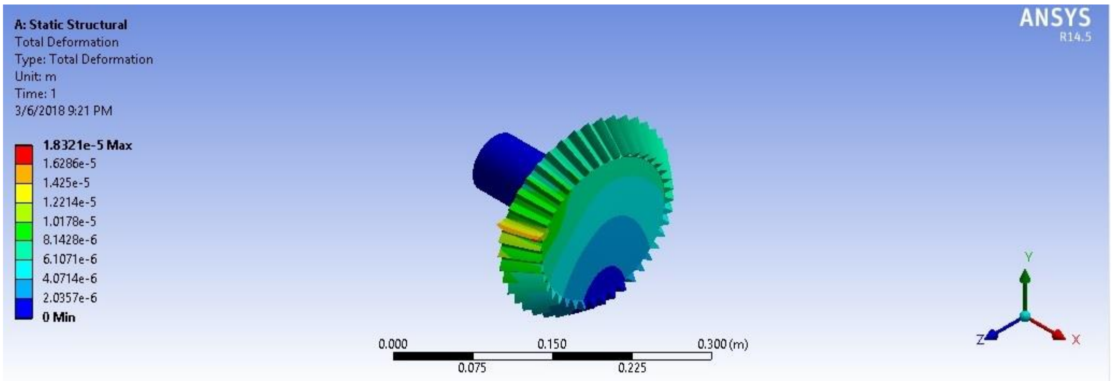


Fig.4.47 Maximum deformation in ANSYS

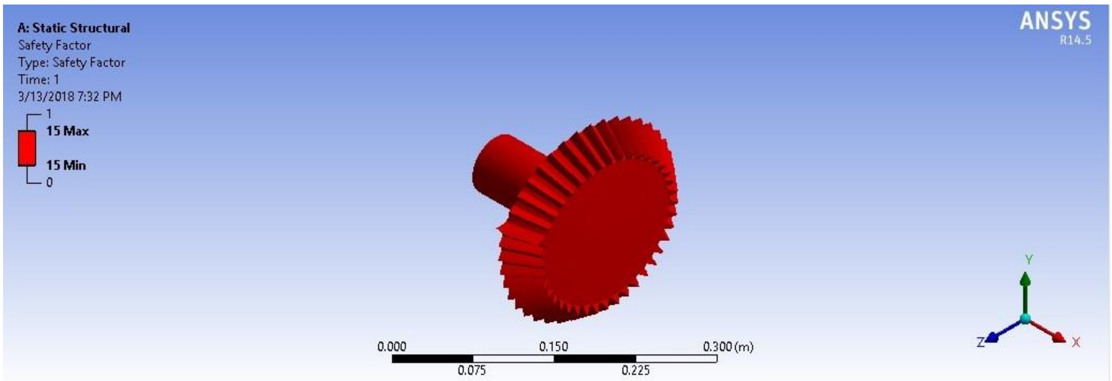


Fig.4.48 Safety factor in ANSYS

CHAPTER V

RESULTS & DISCUSSION

In chapter III, design calculations for finding out tangential, radial, axial forces and factor of safety are calculated for differential gears of centrally suspended cage-less limited slip differential from the input torque given to the final drive gear. The results are tabulated in table 5.1

Table 5.1 Theoretically calculated forces

Sl.No	Gear name	Tangential Forces (N)	Radial Forces (N)	Axial/Thrust Forces (N)	Factor of safety
1	Final drive gear	1310.89	396.99	264.66	>10
2	Crown gears	2126.127	547.1929	547.1929	6.9
3	Side gears	2126.127	547.1929	547.1929	6.9
4	Ring gear	1310.89	264.66	396.99	>10

Beam and Wear strength of differential gears, safety factors against bending failure and pitting failure are calculated from Tangential forces acting on the differential gears and are tabulated in table 5.2

Table 5.2 Beam and Wear strength of differential gears

Sl.No	Gear name	Beam strength (N)	Wear strength (N)	FS_b	FS_w	Effective load (N)
1	Final drive gear	20,178.32	35,954.53	3.46	6.1714	5825.962
2	Crown gears	11,204.9296	10,800	1.6048	1.5468	4855.786
3	Side gears	11,204.9296	10,800	1.6048	1.5468	4855.786
4	Ring gear	20,178.32	35,954.53	3.46	6.1714	5825.962

The beam strengths obtained above are greater than actual working Tangential load acting on the differential gears.

The Beam and Wear strength of tooth of the differential gears is more than the effective load between the meshing teeth of differential gears. Hence, the design is safe.

Safety factors against bending and pitting failures are more than 1 and less than 7 to avoid unnecessary weight of Components which increases weight of the differential resulting in increase of the total weight of vehicle.

By conducting static structural analysis on the following differential gears, solutions obtained are maximum and minimum von-Mises stresses and factor of safety. The results obtained are tabulated in table 5.3

Table 5.3 von-Mises stresses

Sl.No	Gear name	Minimum stress (Pa)	Maximum stress (Pa)	Allowable stress (Pa)	Factor of safety
1	Final drive gear	55.588	2.2066e7	5.5e8	>10
2	Crown gears	1766.3	1.5801e8	5.5e8	5.3794
3	Side gears	0.0062892	1.2546e8	5.5e8	6.7753
4	Ring gear	76.257	3.2477e7	5.5e8	>10

Safety factors obtained here are close to those obtained in theoretical calculations .It is observed that Maximum stress on all differential gears is less than the allowable stress. So, the design is safe.

The displacements obtained after deformation in static structural analysis are tabulated in table 5.4

Table 5.4 Deformation of differential gears

Sl.No	Gear name	Minimum (mm)	Maximum (mm)
1	Final drive gear	0	1.8321e-5
2	Crown gears	0	1.2678e-5
3	Side gears	0	1.2499e-5
4	Ring gear	0	3.8695e-6

The maximum displacement occurred during above deformations in differential gears are in permissible limits which can be determined by the properties of material.

This differential is easy to assemble and disassemble. For an auto racing application, where the driveline is routinely disassembled, inspected and repaired before and after race.

CHAPTER VI

CONCLUSIONS

From the theoretical calculations and FEA analysis the following conclusions are drawn:

- Maximum tangential force of 2126.127N, axial and radial force of 547.1929N obtained for Crown gears and Side gears. This is because of small size of the gear, but receives more torque.

- Maximum Effective load of 5825.962N acts on the Final and Ring gears that has Beam and Wear strength of 20,178.32N and 35,954.53N respectively. So, Factor of safety is required to keep the gears safe.

- The Maximum von-Misses obtained in Static structural analysis of Final, Crown, Side and Ring gears are far less than the Allowable stress of 5.5e8Pa.

- Factor of safety obtained in theoretical calculations for Crown gears (6.9) and Side gears (6.9) are close to the Factor of Safety obtained from FEA for Crown gears (5.3794) and Side gears (6.7753). It is same in case of Final drive and Ring gears.

- The above results are valid only for a torque of 122N-m and speed of 4000rpm.

- In the present work, we considered single torque, speed and gear material 20MnCr5. However, the work can be extended to other materials at different torque and speeds.

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