Analysis of Crown Pinion of a Differential Gear Box in Rear Axle To Improve Crown Life by Replacing Material with Existing One

Prerana U. Jiwane
Research Scholar, M.E. - CAD/CAM & Robotics,
Datta Meghe COE,
Airoli, Navi Mumbai, India

Prof. (Dr.) Prashant D. Deshmukh
Professor & Project Guide
Dept. of Mechanical Engg.
Datta Meghe COE,
Airoli, Navi Mumbai, India

Dhirajkumar K. More
Asst. Professor
Dept. of Mechanical Engg.
Datta Meghe COE,
Airoli, Navi Mumbai, India

***Abstract:** Differential is an essential component of transmission system. The differential is designed to drive a pair of wheels with equal torque while allowing them to rotate at different speeds. For achieving different speed, gear train of differential must be designed in such a way that gears must transmit different speeds with efficient torque. In this project, regular metal used for gear is replaced by upgraded material to reduce vibrations and increase torque.*

In this project work, bevel pinion and crown gear present in the differential assembly is redesigned. The selected crown gear is the existing part of the SAFARI, the passenger car. In this project the crown and pinion material is replaced by upgraded material with more strength and durability of the gear set. Bevel pinion and crown gear are High efficiency gears (98 and higher) which can transfer power across non-intersecting shafts. Spiral bevel gears transmit loads evenly and are easy going than straight bevel.Here, Crown Gear is designed and analyzed for different speeds at 1200 rpm, 2000 rpm and 4000 rpm. Analysis is also conducted by varying the materials for gears SAE 9310, AISI 4310 and Aluminum Alloy. Further, the analysis is conducted to verify the best material for the crown gear in the gear box at higher speeds by analyzing stress, displacement and also by considering weight reduction. Analysis is done on the differential by applying tangential and static loads. After detailed analysis it is found that the performance of the Crown Gear by using Steel Grade SAE 9310 is appropriate, which results in decreasing vibration and wear.

***KEYWORDS:** Crown Gear, Diffrential.*

Introduction

It is not certain when the gears themselves were used, but one can confirm based on Aristotle's records those they already existed in B.C. used in water wheels. For bevel gears, even though the forms are different from the present versions, through archeological remains in the Middle East, during B.C., one can confirm the use of wooden gears with shaft angles at 90 degrees.

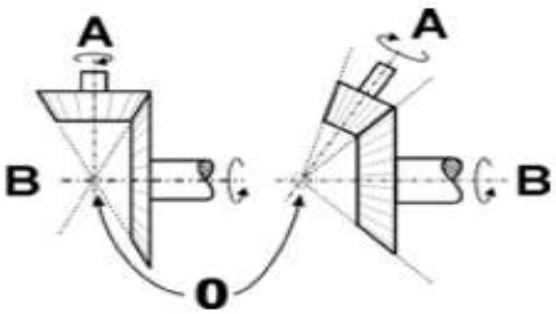


Fig.1 Independently from the operating angle, the gear axes must intersect (at the point O) [1]

In automobiles and other wheeled vehicles, the differential allows the outer drive wheel to rotate faster than the inner drive wheel during a turn. This is necessary when the vehicle turns, making the wheel that is travelling around the outside of the turning curve roll past and faster than the other. The average of the rotational speed of the two driving wheels equals the input rotational speed of the drive shaft [2]. It should be noted that in the interest of sound engineering and good gear design practice, the total number of teeth in a gear pair should, wherever possible, be an odd number which is not divisible by any other number, so that when the gear ratio pair is calculated an unequal number of teeth is called for on the gears.An increase in the speed of one wheel is balanced by a decrease in the speed of the other. When used in this way, a differential (hereafter, diff) couples the input shaft (or prop shaft) to the Pinion, which in turn runs on the Crown wheel of the diff. This also works as reduction gearing to give the ratio [3]. A typical crown wheel and pinion used in heavy vehicles are the most stress liable parts of a vehicle and demands high wear resistance, high contact fatigue strength.

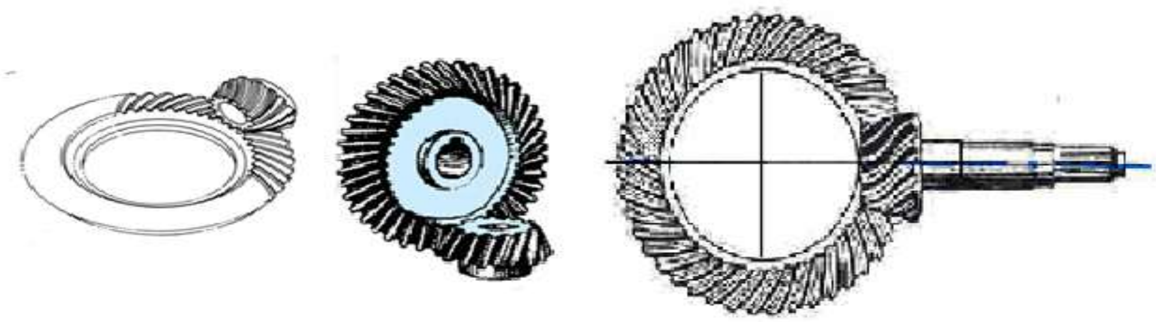


Fig.2 Crown and Pinion of final drive of an automobile [2]

Many years of gear industry experience has led the design community to rely on carburized, case-hardened steel for crown gear. Therefore, Crown gear materials are limited to only those which are easily carburized and case-hardened. In this research work, material AISI 4310 is replaced by SAE9310 which gives more strength while comparison. From the safety point of view Auto-Manufactures must follows certain rules and regulations which were set by public and road safety organisation.

In order to improve the durability of the gears as a pair, the pinion teeth must therefore be strengthened until their load-bearing capacity is equal to that of the crown wheel teeth. This is achieved by increasing the thickness of the pinion teeth and decreasing the thickness of the bevel wheel teeth to compensate. The amount of correction required is dependent on the following:

a. The module or diametral pitch
b. The gear ratio
c. The stress calculations for both the pinion and crown.

Scope and Objectives:

With the demands for smaller gear box transmitting more power at higher speed and to reduce vibration and limit noise without increasing costs. In this work the crown gear and pinion of differential is provided with reduced number of teeth to improve torque transmitted.

By reducing number of teeth on gears to achieve desirable gear ratio with a proper alternative material the failure issue can be resolved.

The major Objectives of this project can be stated in following points:

1. To redesign gears with reduced number of teeth by adjusting pressure angle so as to increase torque with relevant standards in automotive engineering.
2. To Utilize CAD/CAE practises for addressing the design and analysis.
3. To Suggest alternative material and /or process for mass manufacturing.

Literature Review

A Bensely Albert et al. (2006) - investigates failure of crown wheel and pinion. A fractured gear was subjected to detailed analysis using standard metallurgical techniques to identify the cause for failure. The study concludes that the failure is due to the negotiation made in raw material composition by the manufacturer, which is evident by the presence of high manganese content and non-existence of nickel and molybdenum. Author pointed out that, Crown and pinion are most stress prone parts of a vehicle and demands high wear resistance, high contact fatigue strength [4].

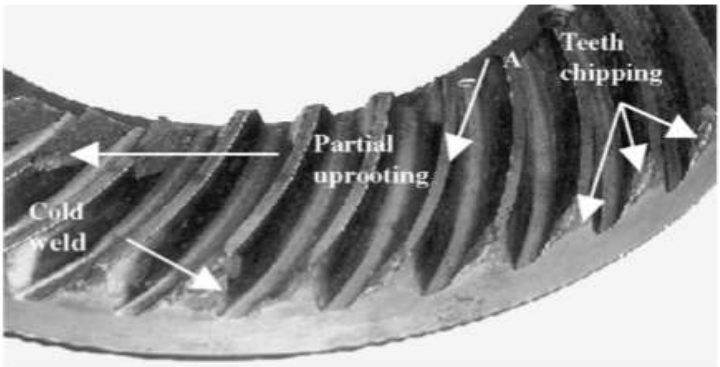


Fig. 3 Crown wheel pinion [4]

Further, an ideal crown wheel and pinion should have uniform and optimum metallurgical quality, excellent heat distortion control, maximum impact strength, stiff wear resistance, optimal transmission efficiency, less noise, vibration free qualities. Author mention here that, above qualities may achieve with Gas carburizing process. He suggested that the manufacturer should make the critical component durable and efficient through accurate and consistent manufacturing standards by selecting appropriate materials and correct heat treatment parameters. En 353(15 Ni Cr 1 Mo 12) And En 207(20 Mn Cr 1) are the two widely used fine-grained steel billet material used in manufacturing of these critical component. This will enable to produce durable components in future.

AnoopLega et al. (2016) [5] – presented his work with objective of the research to develop the composite material gear box using computer aided Engineering. The modelling of gears is done using parametric methodology; 3D family is generated by set of variables which controls other gear dimensions related gear design laws. assembled using CATIA software package. Product Design Specification sheet was developed for the gearbox and simultaneously material selection was carried out through detailed study and past performance of composite materials. Gearbox assembly is imported in Ansys software package and evaluated for equivalent (von-Misses) stress and equivalent (von-Misses) elastic strain for both composite material and existing metallic material. Comparative Results revealed the feasibility of composite material gearbox with approximately 60% weight saving and lower stresses than metallic gearbox, with other composite material advantages.

Zheng Li et. al (2013) investigates regarding the situations of frictional shear stress of gear teeth and the relevant frictional effects on bending stresses and transmission error in gear meshing. Author pointed out that, sliding friction is one of the major reasons causing gear failure and vibration; the adequate consideration at ionic frictional effects is essential for understanding gear contact behaviour accurately. Here, teeth frictional effect on gear performance in spur gear is presented using finite element method is analyses. Author explained, bending stress and transmission error results with static and dynamic boundary conditions. here he indicates that, the significant effects due to the sliding friction between the surfaces of contacted gear teeth, and the friction effect cannot be ignored. The potentially significant contribution of tooth frictional shear stress is presented, particularly in the case of gear tooth contact analysis with both static and dynamic boundary conditions. Bending stress analysis is an important objective of static performance investigation because it is the primary reason of various gear tooth failures [6].

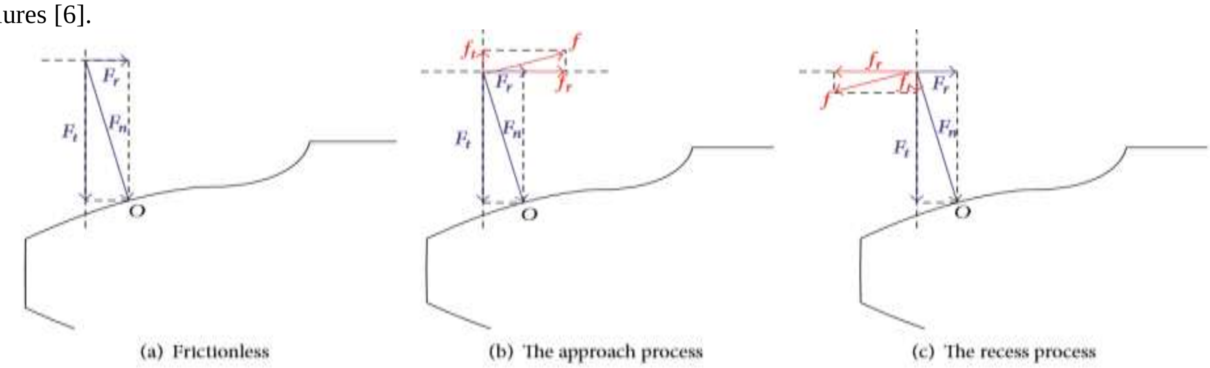


Fig.4 Gear contact force [6]

Fig.4 (a) indicates the relation of the input force and friction force. The input force vertical component FV and horizontal component Fh which cause bending stress of nonfictional model should be

$$FV = Ft$$
$$F\ h = Fr$$

In the first contact phase, the friction force components have been shown as in Fig.4 (b). The current external contact force (including input force and friction force) components which cause bending stress should be

$$f = Fn\times \mu$$
$$FV = Ft - ft$$
$$F\ h = Fr + fr$$

where μ is the coefficient of friction. In the second contact phase, the friction status is shown as in Fig.4 (c), friction force direction is opposite to the first contact phase, and the current external contact force components which cause bending stress should be

$$f = Fn\times \mu,$$
$$FV = Ft + ,$$
$$F\ h = Fr - fr$$

where μ is coefficient of friction. The formulas explain significant phenomena owing to sliding/rolling frictional effect. The friction force component counteracts the input force component in vertical, and the directions of horizontal contact force component and the elastic deformation are different, so the high horizontal contact force components will decrease bending stress. Therefore, the bending tensile stress should be smaller than nonfrictional model in the first phase. In the second contact phase, the friction force direction has become opposite, the friction component and input force component in vertical are in the same direction to amplify resultant contact force, and the horizontal input force component has also been weakening Third, the result indicates that the frictional effect on static transmission error is very limited. The results of author research indicates, that the frictional effect is an important boundary condition in gear design.

Santosh S Bagewadi, et al. (2014) discussed on, Spiral bevel gear teeth are primarily designed for resistance to pitting and for their bending strength capacity. Design for pitting resistance is primarily governed by a failure mode of fatigue on the surface of the gear teeth under the influence of the contact stress between the mating gears. Design for bending strength capacity is based on a failure mode of breakage in the gear teeth caused by bending fatigue. Author advised the following factor for Crown Gear and pinion's number of teeth as shown in Fig. 5 [7].

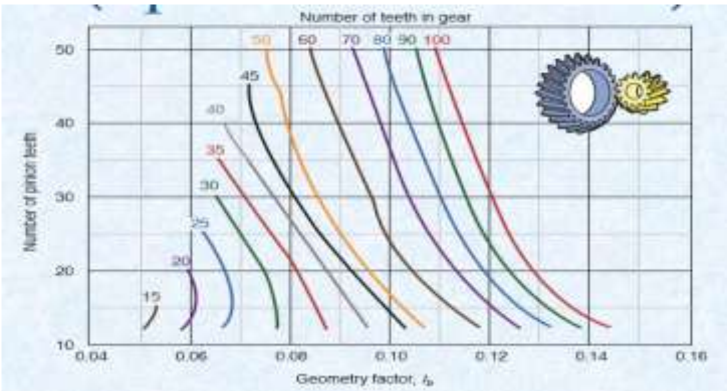


Fig 5. Geometry factors for spiral bevel gears, with pressure angle ϕ=20°, spiral angle Ψ= 25° and shaft angle=90 ° [7]

This study results that, decrease in the number of teeth on pinion leads to the increase in torque at the output. Calculating the design parameters for SAE 4130 steel material gives the margin of safety value of 0.57 for 11 teeth. Similar calculations are prepared for SAE 9310 steel material which gives margin of safety value of 0.68 for 10 teeth. From this, researcher concluded that margin of safety is high even though after reducing 1 tooth on pinion. Reduction of teeth also reduces the weight of the pinion. Hence, it is suggested that, the newly designed pinion can be installed in the existing gear box with the existing bearing, housing and other accessories.In this work, spiral bevel pinion present in the differential assembly is redesigned. The selected pinion is the existing part of the bolero pickup vehicle. The material used to manufacture the existing pinion is SAE 4130 steel. To provide the same or higher margin of safety author picked, SAE 9310 steel material for redesign pinion.

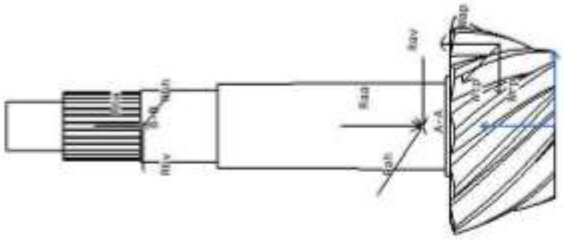


Fig.6 Load acting on pinion.[7]

B. Venkatesh et al. (2014) - obtained Von-Misses stress theoretically with using ANSYS software. He found that for Aluminium alloy, stress values obtained from ANSYS are less than that of the theoretical calculations. The natural frequencies and mode shapes are important parameters in the design of a structure for dynamic loading conditions, which are safe and less than the other materials like steel. Aluminium alloy reduces the weight up to 55.67% compared to the other materials. Aluminium is having unique property like corrosion resistance, good surface finish. Hence it permits excellent silent operation. Weight reduction is a very important criterion, in order to minimize the unbalanced forces setup in the marine gear system, there by improving the system performance [8].

Dong Yang et al. (2011) **-** selected, the appropriate modification size and modification location, tooth deformation would be compensated and the stress distribution would be controlled in the central part of the tooth; the load concentration, agglutination, pitting of the gear could also be avoided effectively. According to the gear geometry theory and the normal meshing motion equation of gear pairs, changes of meshing points and angles were analyzed, and then, the effect of axial modification on gear pairs meshing movement was discussed. The establishment of the relationship between angle changes and modification size provided not only the basis for calculation and the selection of the modification size, but also a reference for the detection of modification effect in the future work [9].

F. K. Choy et al. (2003)**-** provided a comparison and benchmarking of experimental results obtained from a damaged gear transmission system with those generated from a numerical model. Author presented, a study of the dynamic changes in a gear transmission system due to (a) no gear tooth damage, (b) single gear tooth damage, (c) two consecutive gear teeth damage, and (d) three consecutive gear teeth damage. Author found that, the vibration signature analysis using a joint time-frequency procedure, the Wigner-Ville distribution (WVD), seems to be quite effective in identifying single and multiple teeth damage in a gear transmission [10].

Satya Seetharaman et al. (2010) -proposed to predict power losses for gear pairs operating under wind age conditions. The framework of the model included individual formulations for wind age losses on the periphery and faces of the gears as well as a compressible fluid model for power loss due to pocketing taking place in the meshing zone. The wind age conditions simulate jet lubrication operating conditions or very low oil-level dip lubrication conditions. As an example, the wind age power loss model was applied to two unity-ratio gear sets with varying gear geometry parameters to quantify the contributions of each of the components of the total wind age power loss. For both gear pairs, the wind age pocketing loss was shown to dominate the total gear pair wind age loss. Also, the influence of operating conditions, gear geometry parameters, and lubricant properties on wind age power loss was quantified for the gear pairs in consideration [11].

Saket Bhishikaret al. (2014) [12] -researched, in standard 2 Wheel Steering System, the rear set of wheels are always directed forward and do not play an active role in controlling the steering. While in 4 Wheel Steering System, the rear wheels do play an active role for steering, which can be guided at high as well as low speeds. Production cars are designed to under steer and rarely do they over steer. If a car could automatically compensate for an under steer/over steer problem, the driver would enjoy nearly neutral steering under varying operating conditions. Also in situations like low speed cornering, vehicle parking and driving in city conditions with heavy traffic in tight spaces, driving would be very difficult due to a sedan’s larger wheelbase and track width. Hence there is a requirement of a mechanism which result in less turning radius. he has developed an innovative 4 wheel steering design to implement a mechanism that can serve the purpose of changing in-phase and counter-phase steering of rear wheels depending upon the conditions of turning and lane changing with respect to front wheels, thus enhancing the manoeuvrability of a sedan in accordance with its speed. Their 4 Wheel Steering System gives 64.4% reduction in turning circle radius of a sedan which is reduced from 5.394m to 1.92m, considering HONDA CIVIC as a standard car for our calculations, and steering ratio thereby obtained is 8.177:1 which gives much better manoeuvrability and control on the car even while driving at high speeds.

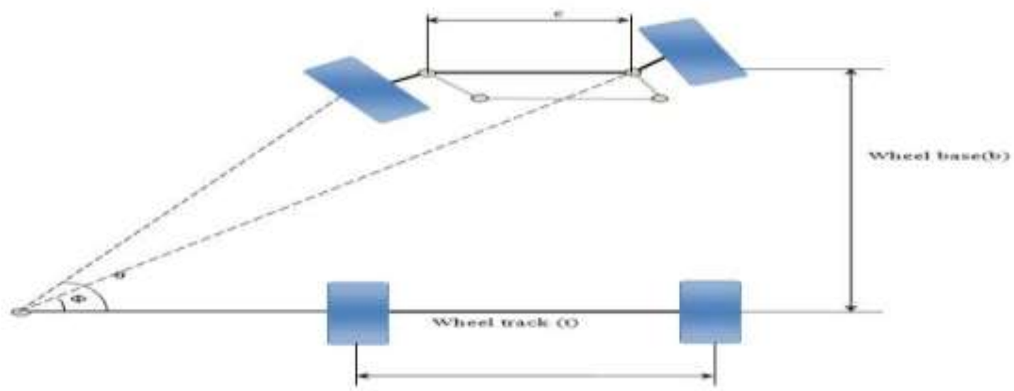


Fig.7 True Rolling Condition [12]

While tackling a turn, the condition of perfect rolling motion will be satisfied if all the four wheel axes when projected at one point called the instantaneous center, and they suggested that thefollowing equation for the satisfaction ofinstantaneous center.

$$\cot\phi - \cot\theta = c/b$$

Here, authorfocused to creates, an innovative 4 wheel active steering mechanism which is feasible to manufacture, easy to install and highly efficient in achieving in-phase and counter-phase rear steering with respect to the front wheels using DRRC. Due to this project they assists in high speed lane changing and better cornering. It combats the problems faced in sharp turning. They reduces the turning circle radius of the car and gives better maneuverability and control while driving at high speeds, thus attaining neutral steering.

C.Veeranjaneyulu1 et al, (2012)- focused on the mechanical design and analysis on assembly of gears in gear box when they transmit power at different speeds i.e-2500 rpm, 5000 rpm and 7500 rpm. Analysis is also conducted by varying the materials for gears, Cast Iron, Cast Steels and Aluminium Alloy. Presently used materials for gears and gear shafts is Cast Iron, Cast steel. In this paper to replace the materials with Aluminium material for reducing weight of the product. Stress, displacement is analyzed by considering weight reduction in the gear box at higher speed. The analysis is done in Cosmos software. It's a product of Solid works. In the present work all the parts of differential are designed under static condition and modelled. The required data is taken from journal paper. Modelling and assembly is done in Solid Works. The detailed drawings of all parts are to be furnished [13].

Name	Type	Min	Max
Stress1	VON: von Mises Stress	5.37613e-008 N/mm^2 (MPa) Node: 38657	2.29414 N/mm^2 (MPa) Node: 20365

Fig. 8 Aluminium_staticload , Under speed 2400 rpm. [13]

By observing the structural analysis results using Aluminium alloy the stress values are within the permissible stress value. So using Aluminium Alloy is safe for differential gear. When comparing the stress values of the three materials for all speeds 2400rpm, 5000rpm and 6400 rpm, the values are less for Aluminium alloy than Alloy Steel and Cast Iron. By observing the frequency analysis, the vibrations are less for Aluminium Alloy than other two materials since its natural frequency is less. And also weight of the Aluminium alloy reduces almost 3 times when compared with Alloy Steel and Cast Iron since its density is very less. Thereby mechanical efficiency will be increased. By observing analysis results, Aluminium Alloy is best material for Differential.

Methodology

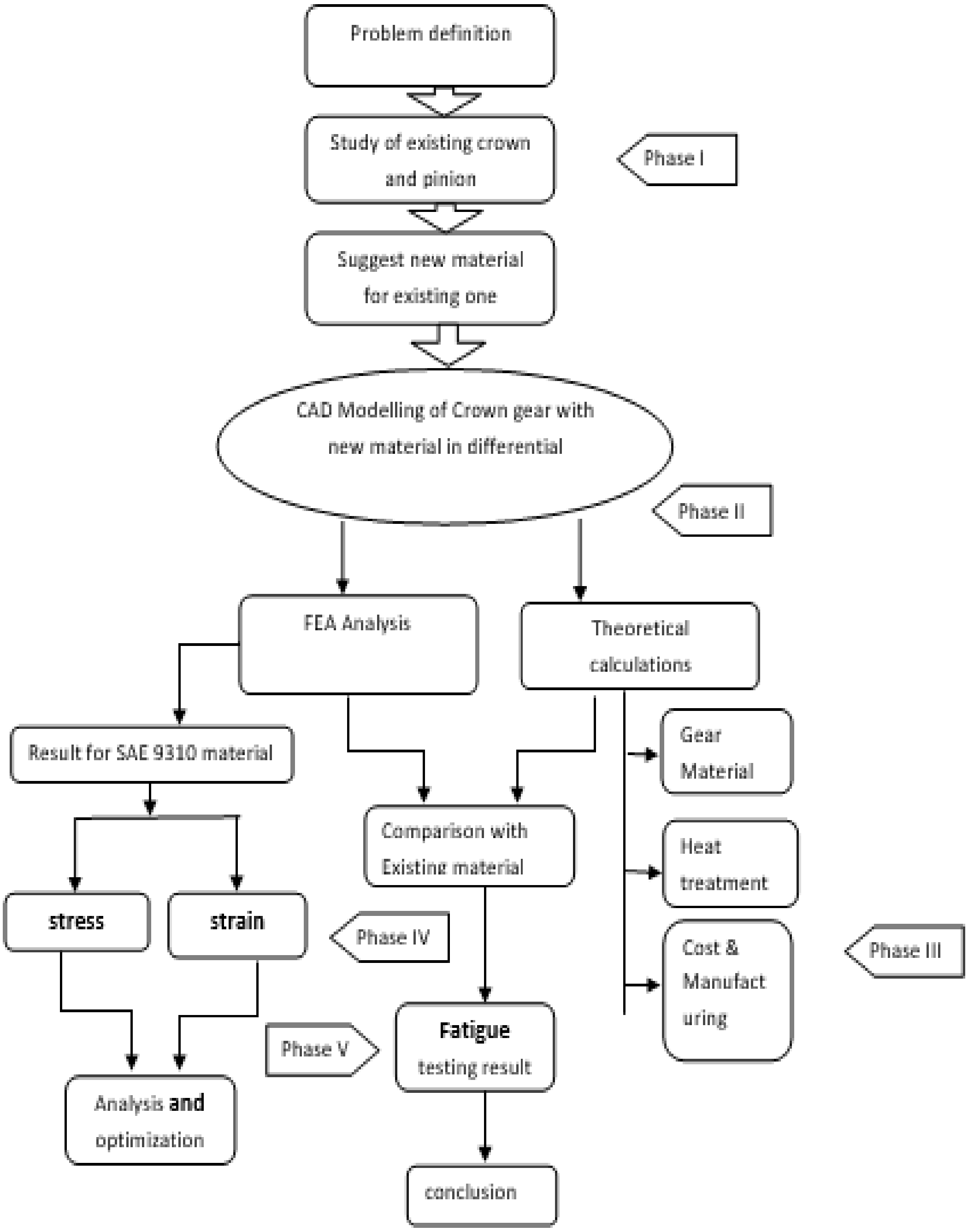


Fig.9. The design methodology for Crown Gear

On basic of chassis, there are two type of Crown gear found in SAFARI CAR, one is Telco and second Spicer. Each car model has their own kind of crown in differential differ from another. Teeth with sets **41/9, 47/11, 43/8** are generally found in crown and pinion respectively of differential. Following Fig. shows real crown and dimensions of crown.

Fig. 10 Real Crown Gear in differential

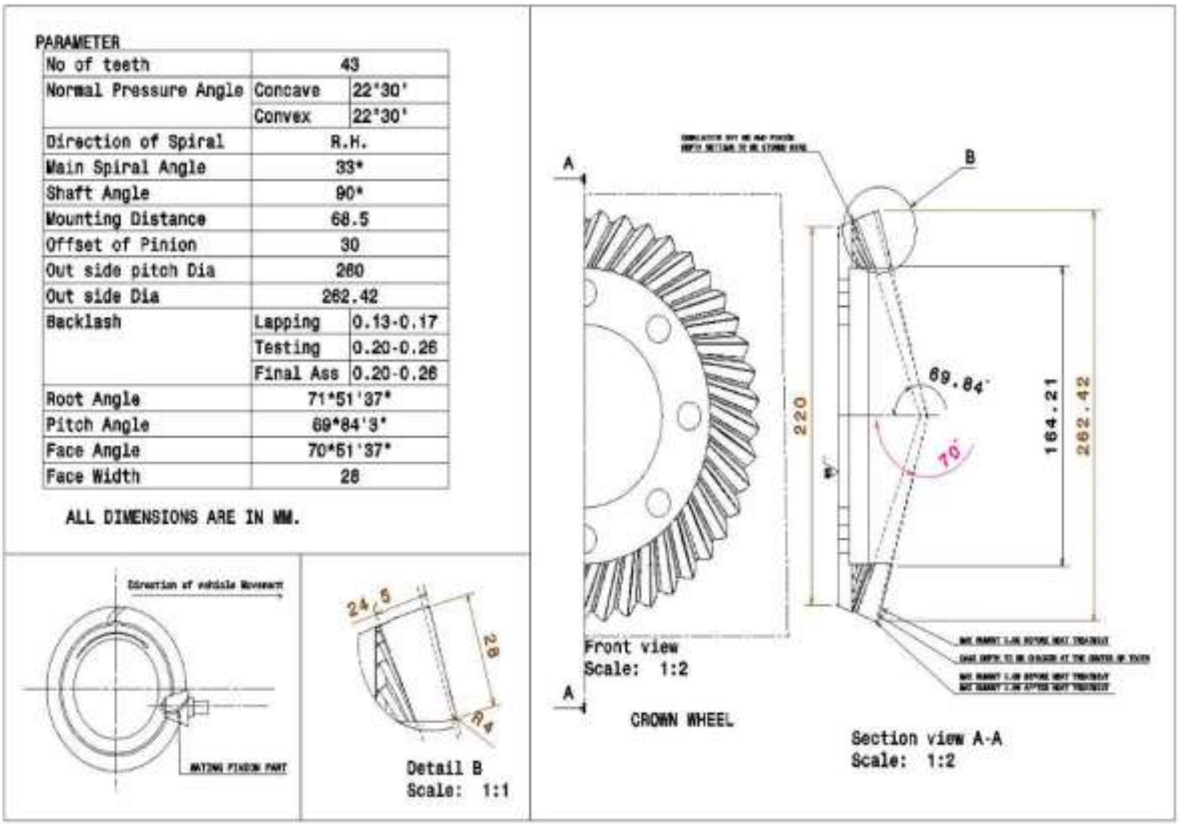


Fig.11 Dimension of Crown Gear

Loading

Torque application to a spiral bevel gear mesh induces tangential, radial, and separating loads on the gear teeth. For simplicity, these loads are assumed to act as point loads applied at the mid-point of the face width of the gear tooth. The radial and separating loads are dependent upon the direction of rotation and hand of spiral, in addition to pressure angle, spiral angle and pitch angle. The tangential loads are defined as

for pinion ,

$$Wtp = \frac{2Tp}{dp - F\sin\gamma}$$

For the gear,

$$Wtg = \frac{2Tg}{dp - F\sin r}$$

$$= \frac{2x1400}{220 - 62.35\sin 30}$$

$$= 15.173\ N/mm^2$$

Where, T= the torque ´ϒ = pitch angle of the pinion

dp = pitch diameter r = pitch angle of the gear F= Face width

The radial and separating loads are calculated as a percentage of the tangential loads calculated above. For a right hand of spiral rotating counter clockwise, the axial thrust load for a driving member (pinion) is defined as

$$Wa = \frac{Wt}{\cos\psi}(\tan\theta\sin\gamma + \sin\psi\cos\gamma)$$

And the separating load is defined as,

$$Wr = \frac{Wt}{\cos\psi}(\tan\theta\cos\gamma - \sin\psi\sin\gamma)$$

$$= \frac{15.173}{\cos 15}(\tan 20\cos 30 - \sin 15\sin 30)$$

$$= 2.91\ x\ 10^3\ N/mm^2$$

Where,

ϕ = pressure angle Ψ = spiral angle

The basic equation for compressive stress in a bevel gear tooth is given by [10],

$$f_c = C_p\sqrt{\frac{W_t \times C_o}{C_v}\frac{1}{F \times dp}\frac{K_m}{I}}$$

= 110.05 Mpa

where, Cp = elastic coefficient = 2805.40 , Wt = tangential tooth load =15.173 Nmm , Co = overload factor =1, Cv = dynamic factor =1, F = face width =62.354mm, dp = pitch diameter =220mm,Km = load distribution factor =1.6, I =geometry factor =1.15.

The elastic coefficient is defined as

$$C_p = \sqrt{\frac{3}{4\pi}\frac{E}{(1-\mu^2)}}$$

=2805.40s

Where

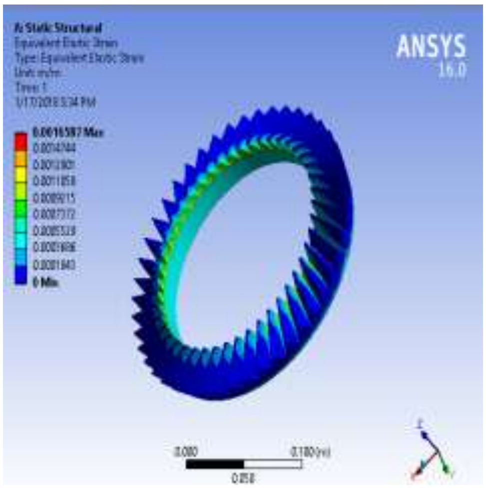
E = Young’s modulus of the material

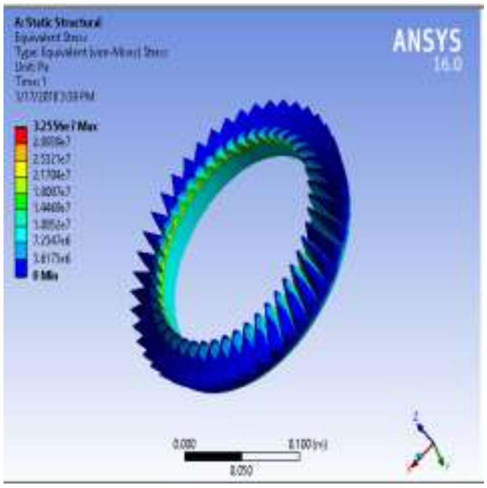
μ = Poisson’s ratio.

For almost all steels, including SAE 9310 the type used for this gear application, the values for E and μ are 3 x 107 psi and .30 respectively.

Results and Discussions:

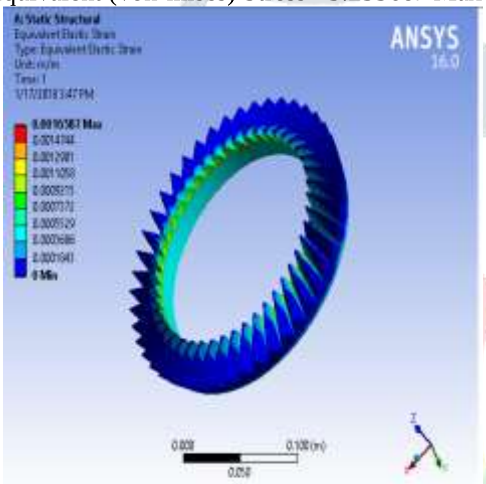
Ansys Analysis

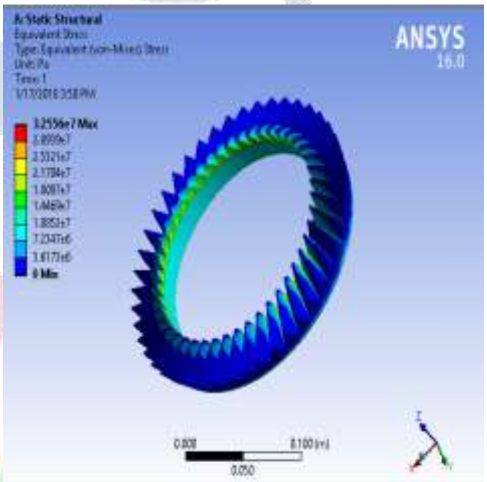


v 

(a) (b)

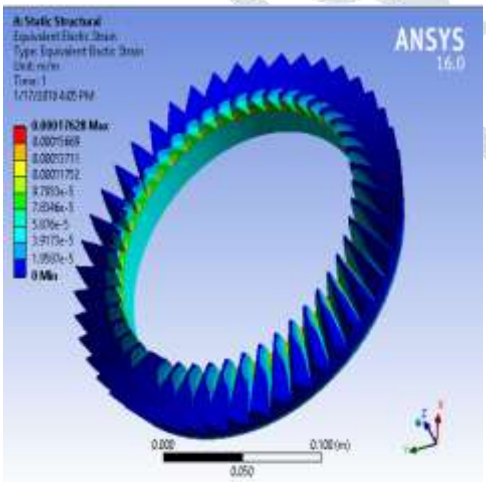
a. Alu_4000rpm_Equivalent Elastic strain =0.0016587Max
b. Alu_4000rpm_Equivalent (von-mises) Stress =3.2556e7 Max

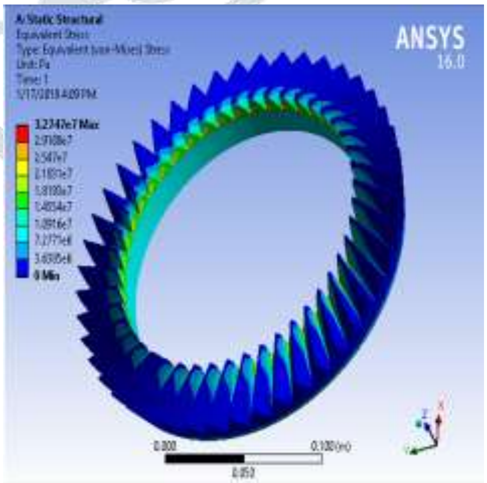




(c) (d)

c.SAE9310_4000rpm_ Equivalent Elastic strain =0.0017658Max
d. SAE9310_4000rpm_ Equivalent (von-mises) Stress =3.2555e7Max



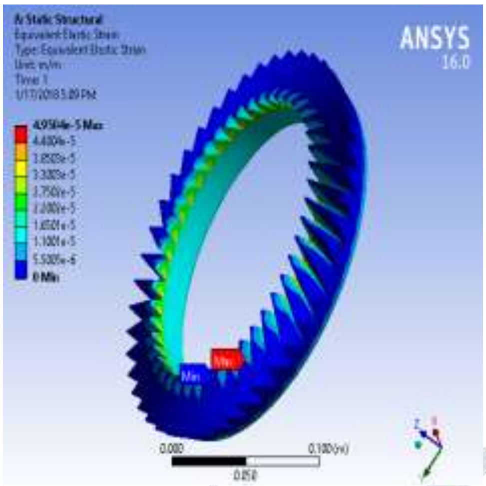


(e) (f)

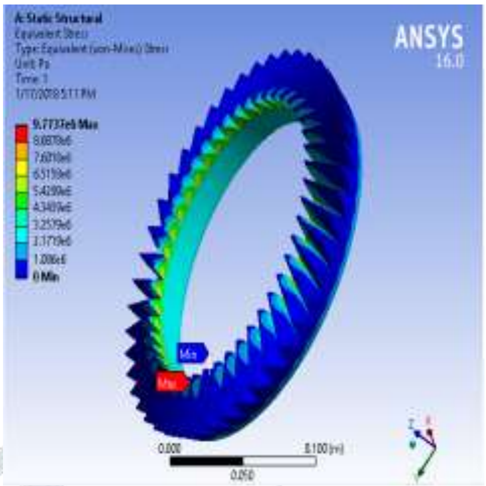
e. AISI4130_4000rpm_ Equivalent Elastic strain = 0.00017628 Max
f .AISI4130_4000rpm_ Equivalent (von-mises) Stress =3.2747e7Max

Load=4000 rpm				
Crown	**Equivalent stress(pa)**		**Equivalent Elastic Strain(m/m)**	
Material	**Maximum**	**Minimum**	**Maximum**	**Minimum**
Aluminium	3.2556e7	0	0.0016587	0

SAE9310	3.2555e7	0	0.0017658	0
AISI4130	3.2747e7	0	0.00017628	0

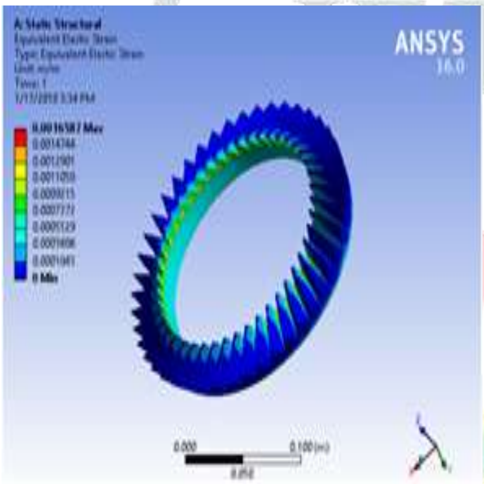


(g)

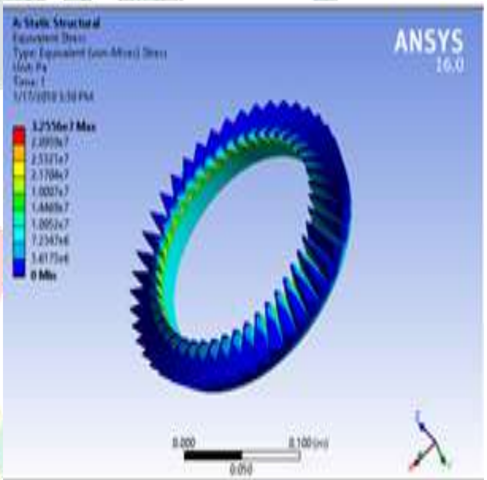


(h)

g. 9310_1200_Equivalent strain = 4.9504e-5 Max
h. 9310_1200_ Equivalent stress = 9.7737e6 Max

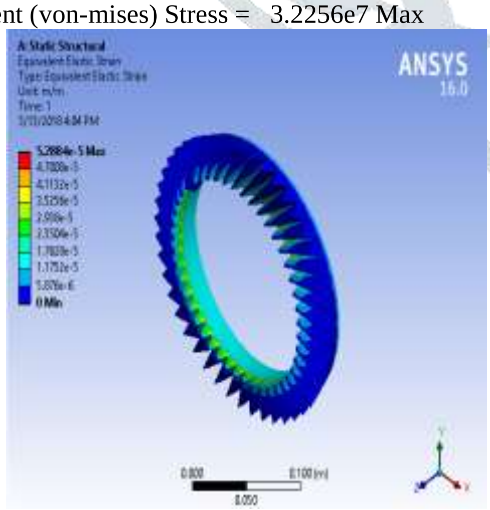


(i)

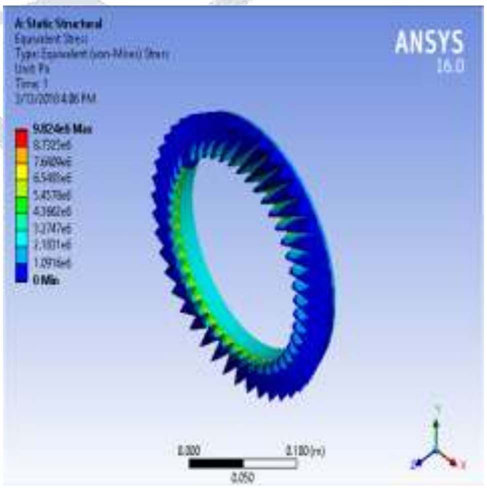


(j)

i.Alu_1200rpm_Equivalent Elastic strain = 0.000165e-6Max
j.Alu_1200rpm_Equivalent (von-mises) Stress = 3.2256e7 Max



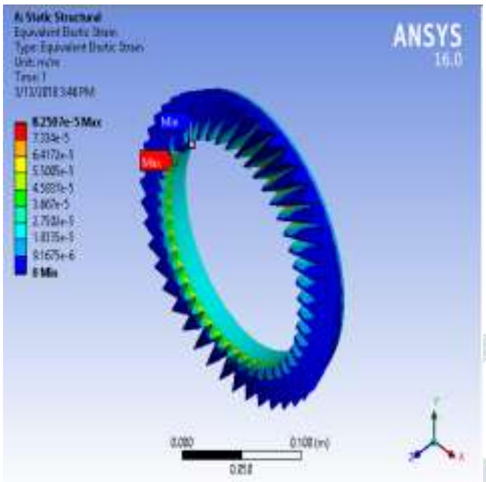
(k)

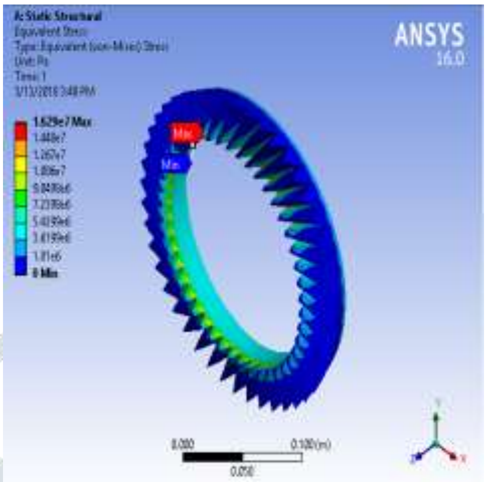


(l)

k. AISI4130_1200rpm_ Equivalent Elastic strain = 5.2884e-5 Max
i.AISI413s0_1200rpm_ Equivalent (von-mises) Stress = 9.824e6 Max

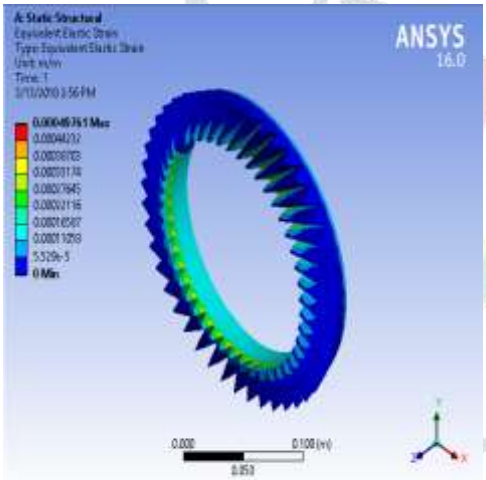
Load=1200 rpm				
Crown	**Equivalent stress(pa)**		**Equivalent Elastic Strain(m/m)**	
Material	**Maximum**	**Minimum**	**Maximum**	**Minimum**
Aluminium	3.2256e7	0	0.000165e-6	0
SAE9310	9.7737e6	0	4.9504e-5	0
AISI4130	9.824e6	0	5.2884e-5	0

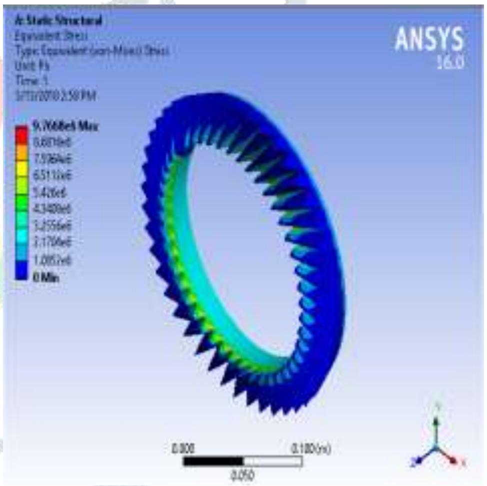




(m) (n)

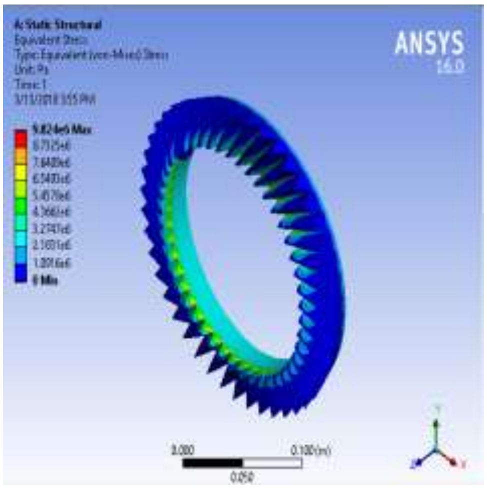
m. 9310_2000_Equivalent strain = 8.2507e-5 Max
n.9310_2000_ Equivalent stress = 1.629e7Max

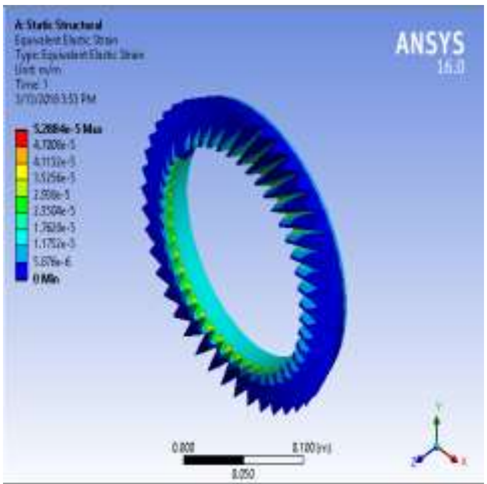




(o) (p)

o. Alu_2000rpm_Equivalent Elastic strain = 0.00049761m/m Max
p. Alu_2000rpm_Equivalent (von-mises) Stress = 9.7668e6 pa Max





(q) (r)

q . AISI4130_2000rpm_ Equivalent Elastic strain = 5.2884e-5 Max
r. AISI4130_2000rpm_ Equivalent (von-mises) Stress = 9.8245e6 Max

Load=2000 rpm				
Crown	Equivalent stress(pa)		Equivalent Elastic Strain(m/m)	
Material	Maximum	Minimum	Maximum	Minimum
Aluminium	9.7668e6	0	7.0004e-5	0
SAE9310	1.629e7	0	8.2507e-5	0
AISI4130	9.8245e6	0	5.2884e-5	0

Conclusion

The existing Crown gear is manufactured by AISI4130 steel material is redesigned by altering SAE9310 Steel as the Crown material. Further it is found that;

- Decrease in the no. of teeth the crown, not only the increase in torque at the output but also.
- The maximum Elastic strain induced is 5.2884e-5 m/m in AISI 4130, which is higher than 4.9544e-5 m/m obtained for SAE 9310.
- The maximum stress induced is 1.629e7 pa in case of AISI4130, which is lower than the maximum Stress induced as 3.256e7 pa obtained for SAE9310.
- SAE 9310 Material is used for Crown and are analysed using ANSYS for equivalent(von-Misses) stress, Displacement (total deformation) and maximum shear elastic strain for different revolutions (1200,2000,4000rpm) under static condition. Comparisons of various stress and strain of SAE9310 material with Aluminium alloy and AISI 4310 being performed and found that among all, SAE9310 is suitably better material for Crown.
- Material SAE9310 also enhances the efficiency of torque. Hence, SAE9310 material is selected as alternative material for crown gear in differential.In this research study, it is noticed that, with the material SAE9310 crown gear gives better result as compared to other materials. The material AISI 4130 gives smooth machining however, SAE 9310 achieve the good meshing properties and meet the analysis factor.

Future scope

In this research study, it is noticed that, with the material SAE9310 crown gear gives better result as compared to other materials. The material AISI 4130 gives smooth machining however, SAE 9310 achieve the good meshing properties and meet the analysis factor.

REFERENCE

[1] ”Development of stainless corrosion Resistant Carburizing Bearing Steel”, Author Thomas J. MCCaffery And David E.Wert .

[2] *Machine Design-Gear Failures*, Eugene E. Shipley, 1967,

[3] ANSI/AGMA 2003-A86, American National Standard Institute/American Gear Manufacturers Association, “Rating the Pitting Resistance and Bending Strength of Generated Straight Bevel”.

[4] A Bensely, S Stephen Jayakumar, D Mohan Lal, G Nagarajan and A Rajadurai (2006), “Failure Investigation of Crown Wheel and Pinion”, Engineering Failure Analysis, Vol. 13, pp. 1285–1292.

[5] Anoop lega, Puneet katyal, vishal Gulati,2013, “Computer Aided Design and analysis of composite gearbox material”.vol.-1,pp.15-25.

[6] Zheng li, Ken Mao, 2013,Frictional Effects on gear tooth Contact analysis, vol-3,pp. 10-18

[7] Santosh Bagewadi, I. G. Bhavi, and S. N. Kurbet,2014,Design And Analysis of Crown Pinion of A Differential Gear Box For Reduced Number Of Teeth To Improve Torque Trasmitted, vol.3,No.4.

[8] B. Venkatesh ,V.Kamala, A.M.K. Prasad ,2010, “Modelling ana Analysis of Aluminium a360 Alloy Helical Gear for Marine Appications”vol-1,pp.24-34.

[9] Dong Guo, Yawen Wang, Teik Lim, Peng Yi, 2015, turning axle while characteristics with emphasis on gear dynamic and psychoacoustics.,Vol.10, pp.280-301.

[10] F. K. Choy, H.Chen ,J Zhou,1997, Modelling And Analysis Of differential Gear box for Stress,vol.6, pp. 56-70.

[11] Satya seetharaman , Ahmet Kahraman,2010, “A Wing age Power loss Model For Spur Gear Pairs”,Vol.53,No.4, pp.473-484

[12] Saket Bhishikar, Vatsal Gudhka, Neel Dalal, Paarth Mehta, Sunil Bhil, A.C. Mehta, 2014, **Design and Simulation of 4 Wheel Steering System**,Volume 3, Issue 12.

[13] C.Veeranjaneyulu1, U. Hari Babu2, march2012, DESIGN AND STRUCTURAL ANALYSIS OF DIFFERENTIAL GEAR BOX AT DIFFERENTLOADS ,201,2IJAERS,Vol. I/,Issue II, pp.65-69