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DESIGN AND FINITE ELEMENT ANALYSIS OF DIFFIRENTIAL GEARBOX FOR DIFFERENT MATERIAL WITH VARYING LOADS

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ABSTRACT: — Differential is used when a vehicle takes a turn, the outer wheel on a longer radius than the inner wheel. The outer wheel turns faster than the inner wheel that is when there is a relative movement between the two rear wheels. If the two rear wheels are rigidly fixed to a rear axle the inner wheel will slip which cause rapid tire wear, steering difficulties and poor load holding. Differential is a part of inner axle housing assembly, which includes the differential rear axles, wheels and bearings. The differential consists of a system of gears arranged in such a way that connects the propeller shaft with the rear axles. In the present work all the parts of differential are designed under static condition and modeled. The required data is taken from journal paper. Modeling and assembly is done in CATIA. The detailed drawings of all parts are to be furnished. The main aim of the project is to focus on the mechanical design and contact analysis on assembly of gears in gear box when they transmit power at different speeds at 2500 rpm; Analysis is also conducted by varying the materials for gears, Cast Iron, And Cast Steel. The analysis is conducted 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. The analysis is done in Ansys Workbench.

Keywords— Differential Gear,

I. INTRODUCTION

Gears have wide variety of applications. They form the most important component in a power transmission system. Advances in engineering technology in recent years have brought demands for gear teeth, which can operate at ever increasing load capacities and speeds**[1].**The gears generally fail when tooth stress exceeds the safe limit. Therefore, it is essential to explore the alternate Gear material **[3].**The important considerations while selecting a Gears material are the ability of the Gear material to withstand high frictional temperature and less abrasive wear **[5].**Weight, manufacturability and cost are also important factors those are need to be considered during the design phase **[6].**Moreover, the Gear must have enough thermal storage capacity to prevent distortion or cracking from thermal stress until the heat can be dissipated**[2].**It must have well anti fade characteristics i.e. their effectiveness should not decrease with constant, prolonged application and should have well anti wear properties **[7].**The quest for energy saving with increased performance of mechanical components has risen significantly over the years has resulted in use of composite material in greater percentage. Weight reduction properties with adequate strength have successfully replaced metallic components with composite material. Moreover, the use of composite materials has increased because of their properties such as high specific stiffness, corrosion free, ability to produce complex shapes, high specific strength and high impact energy absorption etc.**[9].** Product development has changed from the traditional serial process of design, followed by prototype testing and manufacturing but to more on computer aids. Developments in Computer aided engineering has taken at a rapid pace have realty influenced the chain of processes between the initial design and the final realization of a product. Ability of CAE software's on product designing, 3D visualization, analysis, simulation has impacted a lot on time and cost saving to the industry. **[10]**.

2.0 LITERATURE SURVEY

In this chapter, literature has been critically reviewed involving various studies carried out by various researchers related to the field of designing and analysis of Differential gearbox. Differential gearbox is an important part of the automobile i.e. used for transmitting different speeds, while for most vehicles supplying equal torque to each of them.

Dirk Wienecke et al. [17]showed the influences of gear oils, characterized by the base oil and the additive package, on the fuel economy of automobiles are discussed. By optimal oil formulations friction losses can be r educed resulting in higher efficiency data. In order to analyze these influences and lo evaluate the effects the transmission gear is considered as; In complex tribo-technical system consisting of different single Tribological systems, e. g. characterized by gear wheels, bearings, seals, etc. 'The tribo-system automobile transmission gear is defined and described in detail resulting in an analysis of the Tribological stresses in the gear. 'The relationship between the structure of the single systems, their technical functions and the function of the lubricant are described. This systematic analysis was the approach for the simultaneous development of gear oils and a new automobile transmission gear.

Jay Belsky [18]had showed the three potential explanations of the transmission gap pertaining to the limited power of heretically important determinants of attachment security to actually predict attachment security have been entertained. The first drew attention to the prospect that measurements of sensitivity may include a great deal of measurement error, but also that the same may be true with respect to Strange Situation assessments of attachment security. The second explanation raised the prospect that while mother's psychological availability to the infant, in the form of observed sensitivity, has appropriately figured centrally in theorizing about the determinants of attachment security, perhaps insufficient attention has been paid to the time that mother is simply physically available to the child.

3.0 MODEL PREPARATION

For CAST IRON
CROWN WHEEL& shaft gear

Maximum power = 133 bhp at 2500 rpm
Bevel gearing arrangement = 90°
Diameter of crown wheel = D_G = 267mm
Number of teeth on gear = T_G =132
Number of teeth on pinion = T_P = 33
Module = m = D_G/T_G = 267/132 = 2.02 = 2(according to stds)
Diameter of pinion = D_P = m x T_P = 2 x33 = 66mm
Material used for both pinion and gear is cast iron
Brinell hardness number(BHN) = 300
Pressure angle of teeth is 20° involute system Ø = 20°
P= 133xBHP = 133x745.7w = 99178.1w
* We know that velocity ratio
V.R = $T_G/T_P = D_G/D_P = N_P/N_G$
V.R = T_G/T_P = 132/33 = 4
V.R = N_P/N_G
4 = 2500/ N_G
N_G = 625rpm

*For satisfactory operation of bevel gears the number of teeth in the pinion must not be Less than $\frac{48}{\sqrt[2]{1+(v.r)^2}}$

Where v.r = velocity ratio ==>$\frac{48}{\sqrt{1+(4)^2}}$= 11.64

*Since the shafts are at right angles therefore pitch angle for the pinion
$\theta_{p1} = \tan^{-1}(1/v.r)$
$= \tan^{-1}(1/4)$
= 14.04 °
Pitch angle of gear θ_{p2} = 90° - 14.04° =75.96 °
*We know that formative number of teeth for pinion
$T_{EP} = T_P/\cos\theta_{p1}$ =33/cos14.04 = 34.016
And formative number of teeth for gear
$T_{EG} = T_G/\cos\theta_{p2}$ = 132/cos75.96 = 544.107
*Tooth form factor for the pinion
$y^1{}_P = 0.154-0.912/T_{EP}$, for 20° full depth involute system
= 0.154-0.912/ 34.016
= 0.1272
*And tooth form factor for gear

$y^1_G = 0.154 - 0.912/T_{EG}$
$= 0.154 - 0.912/544.107 = 0.1523$
*since the allowable static stresses(σ_O) for both pinion and gear is same (i.e σ_O=196 Mpa) and y^1_P is less than y^1_G, therefore the pinion is weaker. Thus the design should be based upon the pinion.
*allowable static stress(σ_O) = 196 Mpa

TANGENTIAL TOOTH LOAD(W_T):

$W_T = (\sigma_O \times C_v).b.\Pi.m.\, y^1_{P}((L-b)/L)$
C_v = velocity factor = 3/3+v
for 20° full depth involute
v = peripheral speed in m/s
b = face width
m = module=2
y^1p = tooth form factor
L = slant height of pitch cone

$$= \sqrt{(\frac{D_G}{2})^2 + (\frac{D_P}{2})^2}$$

D_G = pitch diameter of gear = 267
D_p = pitch diameter of gear = 66
$V = \frac{\Pi\, Dp\, NP}{60\times1000}$
= 8.64m/s
$C_v = 3/3+8.64=0.258$
$L=\sqrt{(\frac{267}{2})^2 + (\frac{66}{2})^2}$
= 137.518
* The factor (L-b/L) may be called as bevel factor
* For satisfactory operation of the bevel gears the face width should be from L/4 to L/3
So b is taken as L/3
b = 137.518/3 = 46mm
$W_T = (196 \times 0.258) \times 46 \times \Pi \times 2 \times 0.1279\, (\frac{137.518-46}{137.518}) = 1244.03N$

DYNAMIC LOAD (Bukingham equation):

$W_D = W_T + W_I$
$W_D = W_T + \frac{21V(b.C+WT)}{21v+\sqrt{b.c+Wt}}$
V = pitch line velocity
B = face width
C = deformation/dynamic factor in N/mm
$C = \frac{K.e}{\frac{1}{E_P}+\frac{1}{E_G}}$
K = 0.111 for 20° full depth involute system
E_P = young's modulus for material of pinion in N/mm^2=103000 N/mm^2
E_G = young's modulus for material of gear in N/mm^2=103000 N/mm^2
e = tooth error action in mm
e value for module = 2 used in precision gears is e = 0.045
$c = \frac{0.111\times0.045}{\frac{1}{103000}+\frac{1}{103000}} = 257.242N/mm$
$W_D = W_T + \frac{21V(b.C+WT)}{21v+\sqrt{b.c+Wt}}$
$W_D = 1244.03 + \frac{21\times8.64(46\times257.242+1244.03)}{21\times8.64+\sqrt{46\times257.242+1244.03}}$
$W_D = 9265.52N$

STATIC TOOTH LOAD (W_S):

*The static tooth load or endurance strength of the tooth for bevel gear is given by

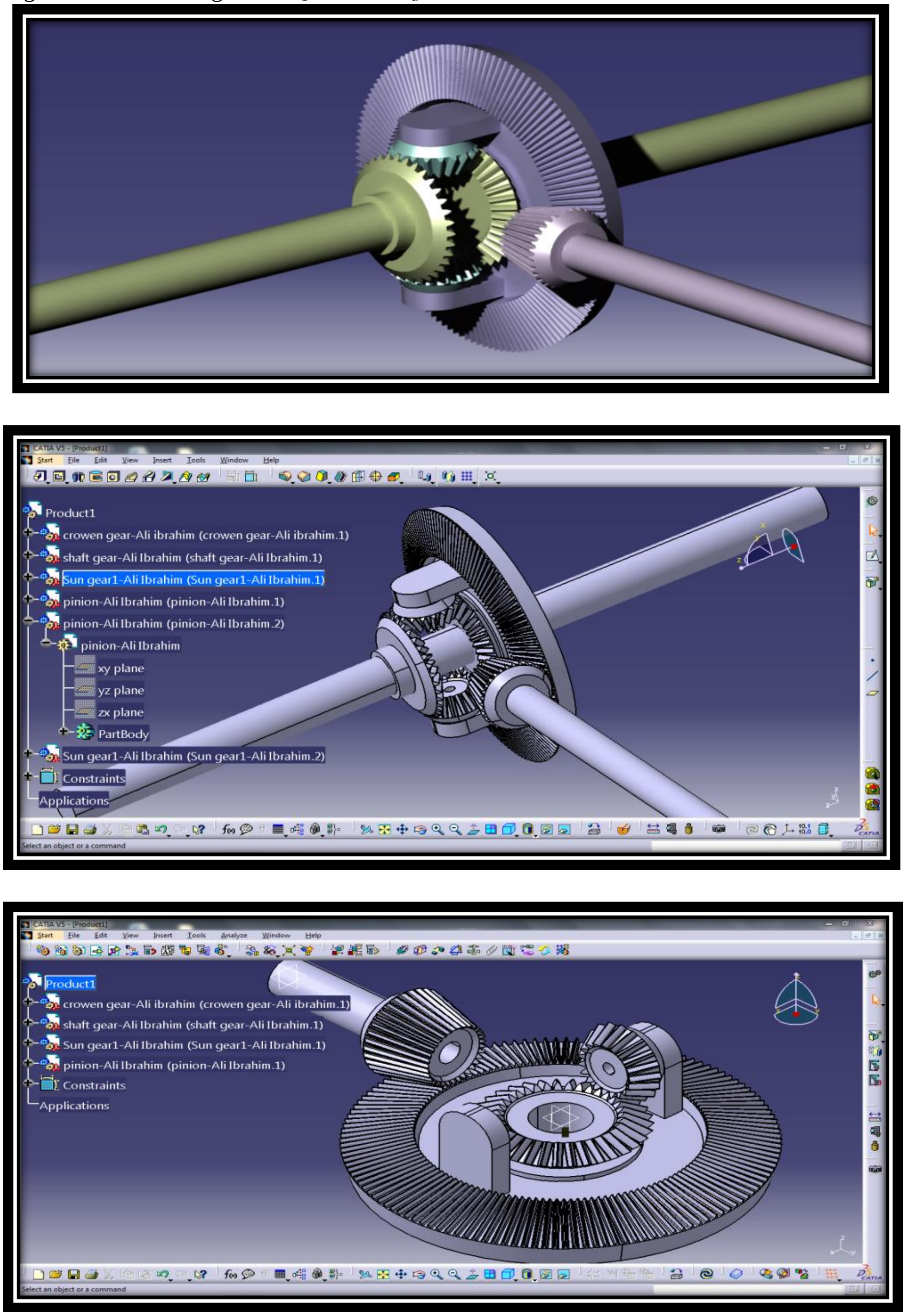
$W_S = \sigma_e.b.\Pi.m.\ y^{1}{}_{P}\,(\frac{L-b}{L})$

(Flexible endurance limit) $\sigma_e = 1.75$ X B.H.N = 1.75 X 300 = 525N/mm^2

$W_S = 525x\ 46\ x\ \pi\ x\ 2\ x\ 0.12118\ (\frac{137.518-46}{137.518})$

$W_S = 12236.999$N

*For safety against tooth breakage the $W_S \geq 1.25\ W_d = 11581.9$N



Figures showing the assemblies of gears

4.0 FINITE ELEMENT ANALYSIS

In this section the analysis work of different parts of differential gears was done using ansys work bench. The different parts of gears includes shafts, crown etc. Firstly analysis of gears is carried out.

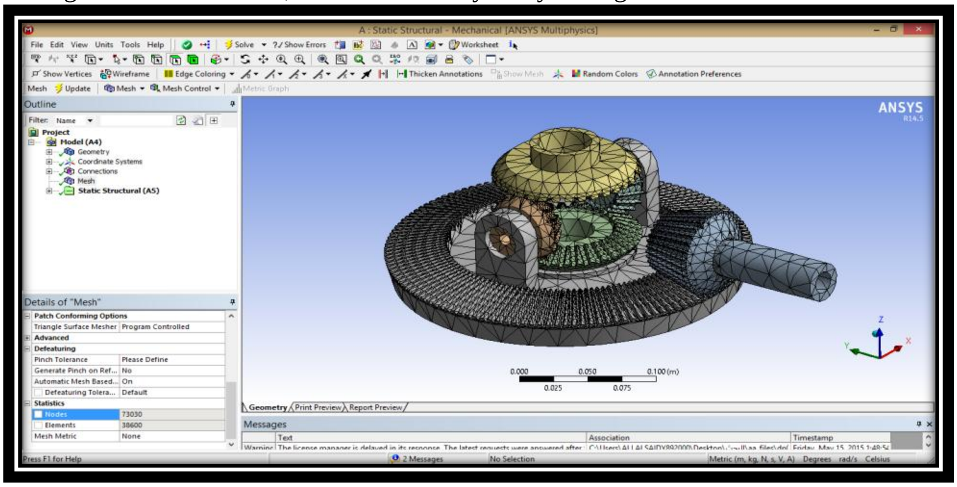
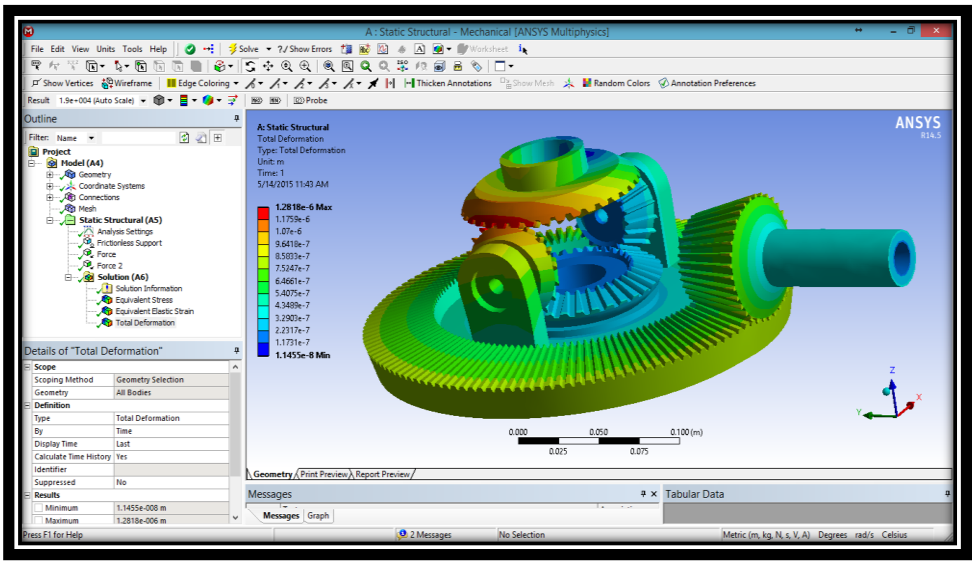


Figure 2 gear analysis

Table 1material properties

Name:	Malleable Cast Iron
Model type	**Linear Elastic Isotropic**
Default failure criterion	**Max von Mises Stress**
Yield strength	**2.75742e+008 N/m^2**
Tensile strength	**4.13613e+008 N/m^2**
Elastic modulus	**1.9e+011 N/m^2**
Poisson's ratio	**0.27**
Mass density	**7300 kg/m^3**
Shear modulus	**8.6e+010 N/m^2**
Thermal expansion	**1.2e-005 /Kelvin**



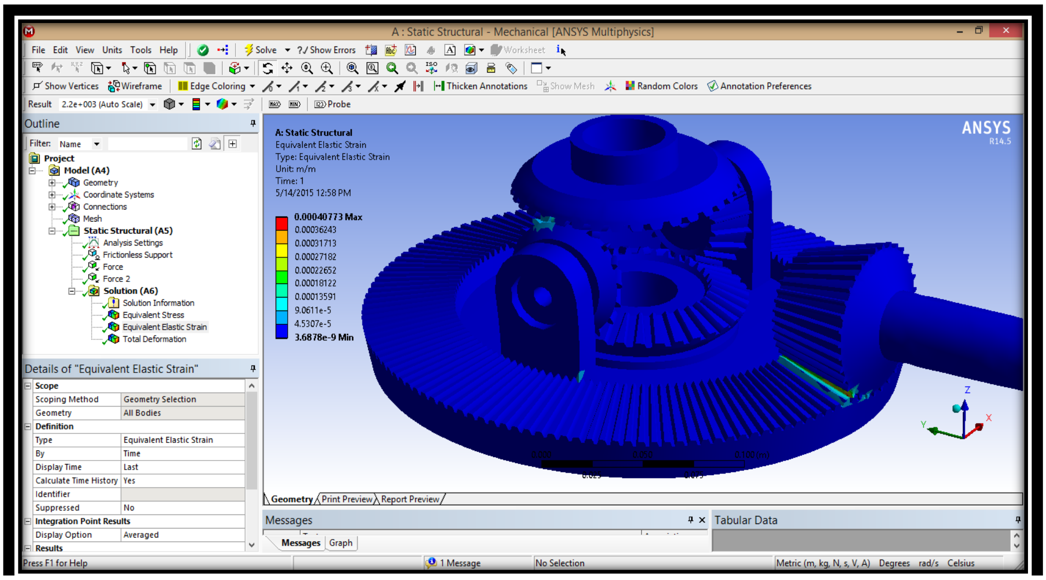


Figure 3 Von misses stress

5.0 RESULTS AND DISCUSSION

In the present paper analysis work has been carried out, where there was a continuous effort to find out the boundary conditions and solving techniques..The below table shows comparison of results of two materials.

material	Ni Cr Alloy Steel	Cast Iron
TANGENTIAL LOAD (N)	**803.93N**	**1244.03N**
Deformation	**1.0755e-5**	**0.00010909**
STRESS (N/mm^2)	**8.076e7**	**9.8967e6**
STRAIN	**0.00040773**	**3.2281e-8**
STATIC LOAD (N)	**18110.76N**	**12236.999N**
Deformation	**0.00016146**	**0.00010909**
STRESS (N/mm^2)	**1.6044e9**	**9.8967e6**
STRAIN	**0.0080997**	**3.2281e-8**

In this project we have designed a differential gear box. Loads are calculated when the gears are transmitting the speed 2500rpm and for different materials Alloy Steel, and Cast Iron, Structural analyses are done on the differential gear box to verify the best material by taking in to account stresses, displacements, weight etc. By observing the structural analysis results using Alloy steel and the cast iron stress values they are both within the permissible stress value. So using Alloy steel and the cast iron is safe for differential gear. When comparing the stress values of the two materials at the speed of 2500rpm the values are less for cast iron than the Alloy steel.

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