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DESIGN AND ANALYSIS OF DIFFERENTIAL GEAR BOX USED IN HEAVY VEHICLE

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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.

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 Cosmos software.Modeling is done in the Pro/Engineer.

1. INTRODUCTION

A transmission or gearbox provides speed and torque conversions from a rotating power source to another device using gear ratios. In British English the term transmission refers to the whole drive train, including gearbox, clutch, prop shaft (for rear-wheel drive), differential and final drive shafts. In American English, however, the distinction is made that a gearbox is any device which converts speed and torque, whereas a transmission is a type of gearbox that can be "shifted" to dynamically change the speed: torque ratio, such as in a vehicle. The most common use is in motor vehicles, where the transmission adapts the output of the internal combustion engine to the drive wheels. Such engines need to operate at a relatively high rotational speed, which is inappropriate for starting, stopping, and slower travel. The transmission reduces the higher engine speed to the slower wheel speed, increasing torque in the process. Transmissions are also used on pedal bicycles, fixed machines, and anywhere else rotational speed and torque needs to be adapted.

1.1 DIFFERENTIAL GEAR BOX

A differential is a device, usually but not necessarily employing gears, capable of transmitting torque and rotation through three shafts, almost always used in one of two ways: in one way, it receives one input and provides two outputs—this is found in most automobiles—and in the other way, it combines two inputs to create an output that is the sum, difference, or average, of the inputs.

In automobiles and other wheeled vehicles, the differential allows each of the driving roadwheels to rotate at different speeds, while for most vehicles supplying equal torque to each of them.

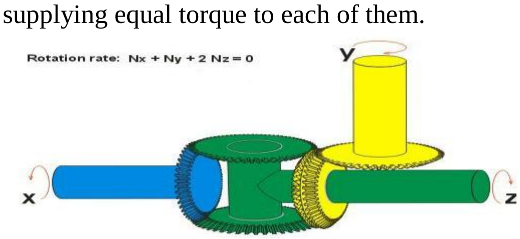

Fig 1 Differential gear box

2. AIM OF THE PROJECT

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 2400 rpm, 5000 rpm and 6400 rpm. Analysis is also conducted by varying the materials for gears, Cast Iron, Nickel Chromium Alloy Steels and Aluminum Alloy.
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.
Design calculations are done on the differential of Ashokleyland 2516M by varying materials and speeds. Differential gear is modeled in Solid works. Analysis is done on the differential by applying tangential and static loads..

3. DESIGN CALCULATIONS OF A DIFFERENTIAL CROWN WHEEL

Specifications of heavy vehicle
Maximum power= 162 bhp at 2400 rpm
Bevel gearing arrangement =90°
Diameter of crown wheel $=D_G= 475$mm
Number of teeth on gear $= T_G=50$
Number of teeth on pinion$= T_P = 8$
Module $= m=D_G/T_G=475/50=9.5=10$ (according to stds)
Diameter of pinion $=m \times T_P=10x8=80$mm
Module $= m=D_G/T_G=475/50=9.5=10$ (according to stds)
Material used for both pinion and gear is nicr steel=30ni4cr1
Brinell hardness number (BHN)=444
Pressure angle of teeth is 20° involute system Ø=20°
P=162BHP = 162x745.7w=120803.4
We know that velocity ratio
$V.R=T_G/T_P= D_G/D_P=N_P/N_G$
$V.R=T_G/T_P=50/8=6.25$
$V.R=N_P/N_G$
$6.25=2400/N_G$
$N_G=384$rpm
For satisfactory operation of bevel gears the number odf teeth in the pinion must not be
Less than $\frac{48}{\sqrt{1+(vr)^2}}$ where v.r=velocity ratio
$=\frac{48}{\sqrt{1+(6.25)^2}}=7.5$
Since the shafts are at right angles therefore pitch angle for the pinion
$\theta_{p1}=\tan^{-1}(1/v.r)$
$=\tan^{-1}(1/6.25)$
=9.0
Pitch angle of gear $\theta_{p2}=90°-9=81$
We know that formative number of teeth for pinion
$T_{EP}=T_P\sec\theta_{p1}=8\sec9 =8$
And formative number of teeth for gear
$T_{EG}=T_G\sec\theta_{p2}=50\sec81 =319.622$
Tooth form factor for the pinion
$y^1{}_P=0.154-0.912/T_{EP}$, for 20° full depth involute system
=0.154-0.912/8
=0.04
And tooth form factor for gear
$y^1{}_G=0.154-0.912/T_{EG}$
=0.154-0.912/319.622
=0.151
since the allowable static stresses(σ_O) for both pinion and gear is same (i.eσ_O=126.66 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) $=\sigma_u/3=380/3=126.66$Mpa
σ_u=ultimate tensile strength=380Mpa
TANGENTIAL TOOTH LOAD(W_T)
$W_T=(\sigma_O x C_v).b.\Pi.m.y^1{}_{P(}(L\text{-}b)/L)$
C_v=velocity factor =3/3+v,
for teeth cut by form cutters
v=peripheral speed in m/s
b=face width
m=module=10
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 =475
D_p= pitch diameter of gear =80
$V=\frac{\Pi\ Dp\ NP}{60\times1000}$
=10.048m/s
C_v==3/3+10.048=0.229
$L=\sqrt{(\frac{475}{2})^2+(\frac{80}{2})^2}$
=240.844
The factor (L-b/L) may be called as bevel factor
For satisfactory operation of the bevel gears the face width should be from 6.3m to 9.5m
So b is taken as 9.5m
b= 9.5x10=95
$W_T=(126.66x0.229)x95x\Pi x10x0.04(\frac{240.844-95}{240.844})$
=2093.840N

DESIGN CALCULATION OF SUN GEAR

Diameter of sungear $=D_G=150$mm
Diameter of pinion $=D_P=70$mm
Number of teeth on gear $= T_G =18$
Number of teeth on pinion $= T_P = 15$
$D=D_G+D_P=220$
$T=T_G+T_P = 33$
Module = m=D/T=220/33=6.66=7(according to stds)
Brinell hardness number(BHN)=444
Pressure angle of teeth is 20° involute system Ø=20°
P=162BHP = 162x745.7w=120803.4w

We know that velocity ratio
$V.R=T_G/T_P= D_G/D_P=N_P/N_G$
$V.R=D_G/D_P =150/70=2.142$
$V.R=N_P/N_G$
$2.142=2400/N_G$
$N_G=1120.448$rpm
Since the shafts are at right angles therefore pitch angle for the pinion
$\theta_{p1}=\tan^{-1}(1/v.r)$
$=\tan^{-1}(1/2.142)$
$=25.025$
Pitch angle of gear $\theta_{p2}=90°-25.025=64.974$
We know that formative number of teeth for pinion
$T_{EP}=T_P\sec\theta_{p1}=15\sec 25.025 =16.554$
And formative number of teeth for gear
$T_{EG}=T_G\sec\theta_{p2}=18\sec 64.974 =42.55$
Tooth form factor for the pinion
$y^1{}_P=0.154-0.912/T_{EP}$, for 20° full depth involute system
$=0.154-0.912/16.554$
$=0.099$
And tooth form factor for gear
$y^1{}_G=0.154-0.912/T_{EG}$
$=0.154-0.912/42.55$
$=0.132$
since the allowable static stresses(σ_O) for both pinion and gear is same (i.eσ_O=126.66 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) $=\sigma_u/3=380/3=126.66$Mpa
σ_u=ultimate tensile strength=380Mpa

4 MODEL OF DIFFERENTIAL GEAR

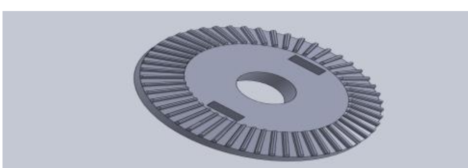

Fig 2:CROWN

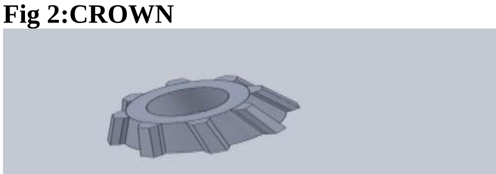

Fig 3:PINION

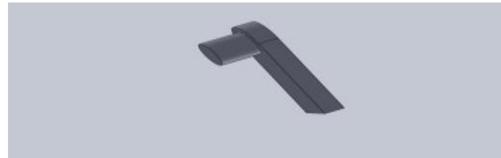

Fig 4: PLANET

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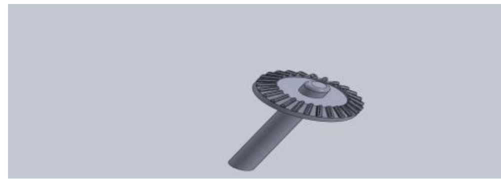

Fig 5: SUNGEAR

4.1STRUCTURAL ANALYSIS OF DIFFERENTIAL GEAR

Fig 6: ASSEMBLY OF DIFFERENTIAL GEAR BOX

Table 1 **Material Properties**

Model Reference	Properties

	Name:	**Nickel Chrome Steel**
	Model type:	**Linear Elastic Isotropic**
	Default failure criterion:	**Max von Mises Stress**
	Yield strength:	**1.72339e+008 N/m^2**
	Tensile strength:	**4.13613e+008 N/m^2**
	Elastic modulus:	**2e+011 N/m^2**
	Poisson's ratio:	**0.28**
	Mass density:	**7800 kg/m^3**
	Shear modulus:	**7.7e+010 N/m^2**
	Thermal expansion coefficient:	**1.1e-005 /Kelvin**

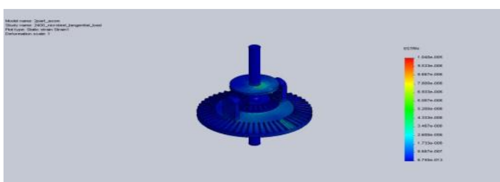

Fig7:2part_assm-2400_nicrsteel_tangential_load-Strain-Strain1

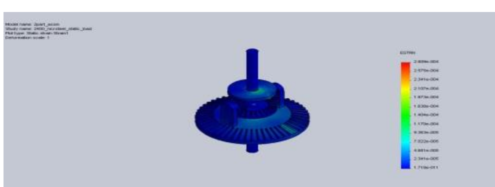

Fig 8 2part_assm-2400_nicrsteel_static_load-Stress-Stress1

Model Reference	Properties	
	Name:	**al_alloy7475-t761**
	Model type:	**Linear Elastic Isotropic**
	Default failure criterion:	**Max von Mises Stress**
	Yield strength:	**1.65e+008 N/m^2**
	Tensile strength:	**3e+007 N/m^2**
	Elastic modulus:	**7e+010 N/m^2**
	Poisson's ratio:	**0.33**
	Mass density:	**2600 kg/m^3**
	Shear modulus:	**3.189e+008 N/m^2**

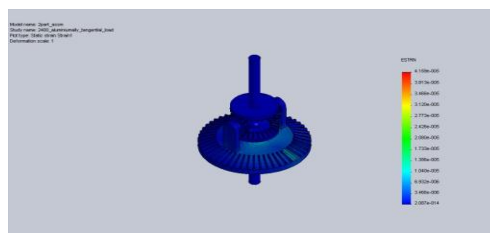

Fig9:2part_assm-2400_aluminiumally_tangential_load-Strain-Strain1

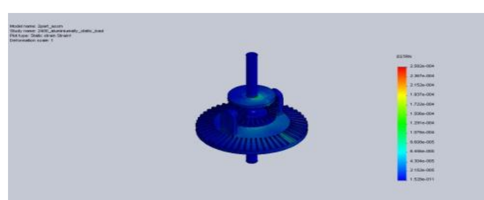

Fig10:2part_assm-2400_aluminiumally_static_load-Strain-Strain1

4.5 ALUMINUM ALLOY

Table 2 Material Properties

4.7 CAST IRON

Table 3 Material Properties

Model Reference	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 coefficient:	**1.2e-005 /Kelvin**

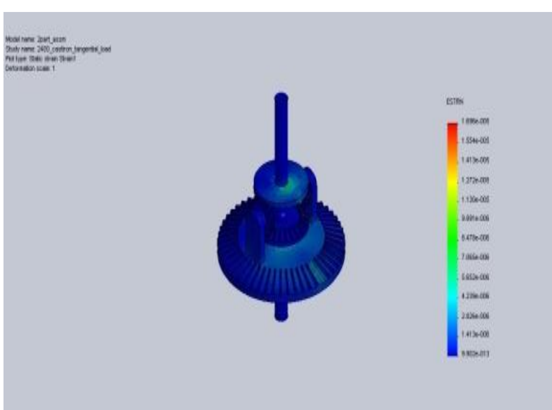

Fig11:2part_assm-2400_castiron_tangential_load-Strain-Strain1

Fig 12: 2part_assm-2400_castiron_static_load-Strain-Strain1

5. RESULTS TABLE :

ANALYSIS OF THREE DIFFERENT MATERIALS BY USING ON DIFFERENT SPEEDS

Table 4: ANALYSIS OF 2400 RPM

TANGENTIAL	Cast Steel	Aluminum Alloy	Cast Iron
LOAD (N)	2093.8	2922.51	3243.08
DISPLACEMENT (mm)	0.00615413	0.0241696	0.0100566
STRESS (N/mm^2)	2.29414	3.19018	3.57544
STRAIN	$1.0400e^{-5}$	$4.1593e^{-5}$	$1.69558e^{-5}$
STATIC LOAD (N)	56141.9	18143.3	37933.7
DISPLACEMENT (mm)	0.164988	0.150063	0.11763
STRESS (N/mm^2)	63.5052	19.8068	41.8212
STRAIN	0.000280882	0.000258239	0.000198329

Table 5:ANALYSIS OF 5000RPM

TANGENTIAL	Cast Steel	Aluminum Alloy	Cast Iron
LOAD (N)	1818.54	1595.22	1770.24
DISPLACEMENT (mm)	0.0054118	0.0131944	0.00548866
STRESS (N/mm^2)	2.584	1.70369	2.01579
STRAIN	1.04958e^{-5}	2.2558e^{-5}	9.32532 e^{-6}
STATIC			
LOAD (N)	56141.9	18143.3	37933.7
DISPLACEMENT (mm)	0.164853	0.150036	0.117614
STRESS (N/mm^2)	74.4963	22.6949	43.1949
STRAIN	0.000309415	0.000274774	0.000199826

6. CONCLUSION

In our project we have designed a differential gear box for Ashok Leyland 2516M. Loads are calculated when the gears are transmitting different speeds 2400rpm, 5000rpm
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 Aluminum alloy the stress values are within the permissible stress value. So using Aluminum Alloy is safe for differential gear. When comparing the stress values of the three materials for all speeds 2400rpm, 5000rpm the values are less for Aluminum alloy than Alloy Steel and Cast Iron.

By observing analysis results, Aluminum Alloy is best material for Differential.

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