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Centrally Suspended Cage-less Differential Gearbox

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Abstract – **In the 21st century, the technological advancements are at its peak. Considering the automobile industry, vehicles with better performance parameters are at our disposal compared to the former vehicles and their technologies. The centrally suspended cage-less differential gearbox is one such innovation which aids the performance of the vehicles like improved cornering efficiency, reduced weight, equal torque distribution and elimination of uneven centre of gravity compared to the conventional differential gearbox. In this study the theoretical designs of the gearbox were obtained using standard design formula. The CAD models were designed on CATIA V5. The models were analysed using ANSYS workbench. After obtaining satisfactory results, manufacturing of the gearbox assembly was carried out.**

Keywords – **Centrally suspended, differential gearbox, cage-less, design and analysis.**

I. INTRODUCTION

A differential gearbox in automobiles allows the wheels (rear) to rotate at different velocities, hence the name. The differential lets the outer wheel of the vehicle to rotate at faster velocity than the inner wheel. This ensures in better stability of the vehicle during turning, cornering in high or low speeds. The vehicle also achieves a better directional stability, i.e. steering geometry will have better results and perfection with less torque generated on the steering wheel.

The conventional types of gearboxes used in automobiles have the bevel gear assembly mounted to the either side of the output gear. This creates different lengths of the output shaft which are connected to the driving wheels and to the differential. The centre of mass and the centre of gravity is also disturbed due to asymmetrical design of the differential gearbox. The cage required for side mounting of the bevel gears adds on weight, hence reducing efficiency of the vehicle.

In the centrally suspended cage-less differential gearbox, the lengths of the output shafts is equal due to central mounting of the bevel gear assembly resulting in even splitting of torque to both the shafts. Elimination of the cage results in reduction of weight of the gearbox.

II. PRE-REQUISITES

The various parameters used for the theoretical design of the differential were selected considering its use for an all terrain vehicle. Furthermore, it is regarded that the differential is coupled to a two stage reduction gearbox with reduction ratio of 12:1.
Engine Power = 7.46 kW
Engine Torque = 14.6 Nm
Max Engine RPM: 3800 rpm
Tyre Size: 23x7x12 inches

Material selection:
Gear and shaft material selected - EN36,
Ultimate tensile strength - 1100 N/mm^2
Hardness - 550 BHN
(Post hardening process)

Table I: Material comparison

Physical properties	EN36	EN24	EN8
Tensile strength N/mm^2	1100	700	700-850
Elongation %	15	13	16
Brinell Hardness HB	341	302	201-255
Yield stress	-	58	465
Density kg/m^3	7800	7833	7850
Young's modulus GPA	200	103	200

III. DESIGN AND CALCULATIONS

Following calculations are for two stage reduction and differential gearbox. (Spur gears)

1. **Stage 1 – Reduction 3:1**
   Input parameters:
   Number of teeth on pinion Z_p= 18
   Number of teeth on gear Z_g= 54
   Factor of safety required N_f= 1.2
   Using Buckingham equation,
   a. Beam Strength:

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□$_b$= $S_{ut}/3$ = 366.67 N/mm^2

b. Bending Strength

F_b = □$_b$xY_px b x m

= 1189.94m^2 N

c. Effective force:

$F_{eff} = K_a$ x K_m x F_t/K_v

F_{eff} = 3814.00 N

F_b= N_f x F_{eff}

1189.94m^2 = 1.2 x 3814

M = 1.79 □2

Therefore, as module = 2,

Dimensions for first stage are

D_p = m x Z_p= 36 mm

D_g = m x Z_g= 108 mm

2. Stage 2 – Reduction 4:1

Input parameters:

Number of teeth on pinion Z_p= 18

Number of teeth on gear Z_g= 72

Factor of safety required N_f= 1.9

Using Buckingham equation,

d. Beam Strength

□$_b$= $S_{ut}/3$ = 366.67 N/mm^2

e. Bending Strength

F_b = □$_b$xY_px b x m

= 1189.94m^2 N

f. Effective force

$F_{eff} = K_a$ x K_m x F_t/K_v

F_{eff} = 3814.00 N

F_b= N_f x F_{eff}

1189.94m^2 = 1.2 x 3804.9

m= 2.46 □2.5

Therefore, as module=2.5,

Dimensions for first stage are:

D_p = m x Z_p= 36 mm

D_g = m x Z_g= 144 mm

3. Bevel gear design

Input parameters:

Pressure angle = 20 degree

Number of teeth on planet = 10

Number of teeth on sun = 18

RPM = 110

a. Tangential load on pinion(N)

P_{tp} = $2Mt_p/D_m$

Pt_p = Tangential load on pinion (N)

M_{tp} = Torque transmitted by pinion (Nmm)

b. Mean radius

R_m = D_p / 2 – b x sin□ / 2

Where,

R_m = Mean radius (mm)

b = Face width (mm)

□ = Pitch angle (degree)

D = p = Pitch circle diameter of bevel pinion

c. Permissible beam strength of the tooth is given by,

S_b = m x b x □bx Y x (1 – b/A_0)

d. Wear strength of tooth is:

S= 0.75 x b x Q x D_p x k/cos □

e. Effective load on tooth is calculated by using Buckingham equation.

P_{eff} = C_s x P_t + P_d

Face width = 10.5 mm

Module = 3.5 mm

CAD Models:

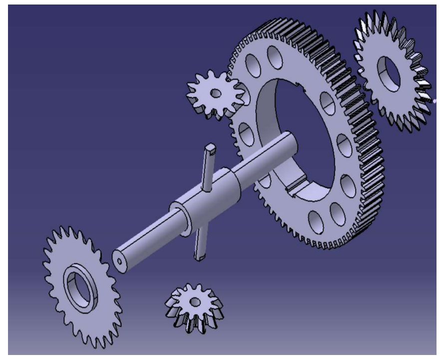

Fig. 1 Bevel assembly exploded view.

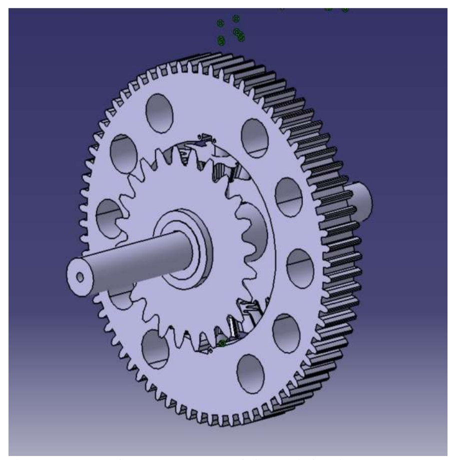

Fig. 2 Bevel gear assembly with ring gear.

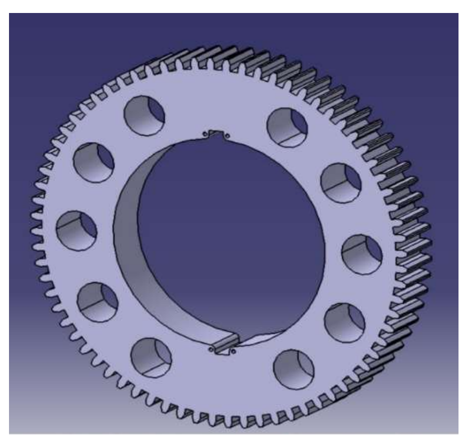

Fig. 3 Ring gear.

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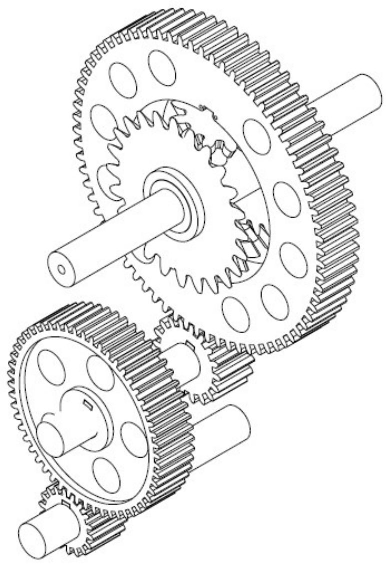

Fig. 4 Gearbox drafting isometric view

IV. ANALYSIS

The Analysis of the Gearbox and Differential assembly was performed on ANSYS workbench. A simple static structural analysis was carried out and four parameters were tested to check the feasibility of the design. The four parameters tested were:

- Total Deformation
- Equivalent Von-Misses stress
- Life Cycle
- Factor of safety

Ansys parameters:

- Mesh Type – Quads/Triads
- Element Size – 1mm
- Number of elements – 30436
- Number of nodes – 176968
- Mesh Transition – Fast
- Mesh Smoothing – Medium
- Mesh Algorithm – Patch Conforming

Quality Checks:

- **Tetra Collapse**
  Ideal Value = 1.0 (Acceptable > 0.1)
  Tetra collapse = h * 1.24 / A
  (Defined as the distance of a node from the opposite face divided by the area of the face multiplied by 1.24)
- **Volumetric Skew**
  Create a sphere passing through the corner nodes of the ideal and actual tetra elements.
  Ideal value = 0 (Acceptable < 0.7)
  Volumetric Skew = $(V_{ideal} – V_{actual}) / V_{ideal}$
- **Stretch**
  Ideal value = 1.0 (Acceptable > 0.2)
  Stretch = R * □24 / L_{max}
  R = Radius of largest possible sphere inside given tetra element.
- **Distortion**
  Ideal value = 1.0 (Acceptable > 0.5)
  Distortion = | J | * V_{olm} / V_{olm}
  LCS GCS
  LCS – Local Coordinate System
  GCS – Global Coordinate System
- **Jacobean**
  Ideal value = 1.0 (Acceptable > 0.5)
  In simple terms, the Jacobean is a scale factor arising because
  Transformed from global coordinates to local Coordinates

Element Quality Check:
Tetra Collapse 1.25
Volumetric Skew 0.5
Stretch 0.75
Distortion 0.75
Jacobean 0.75

Fig. 5 Ring gear meshing for ANSYS.

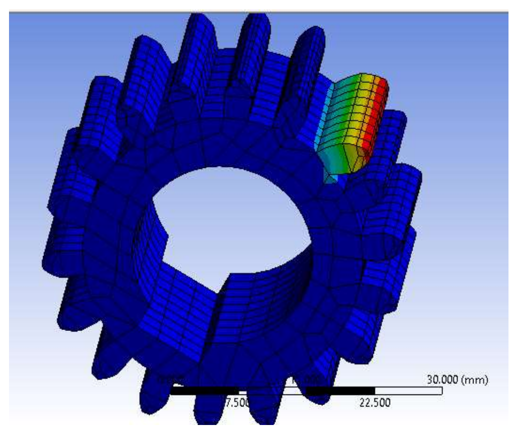

Fig. 6 Total deformation of Pinion 1.

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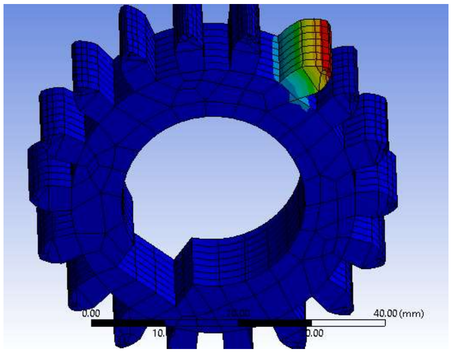

Fig. 7 Total deformation of Pinion 2.

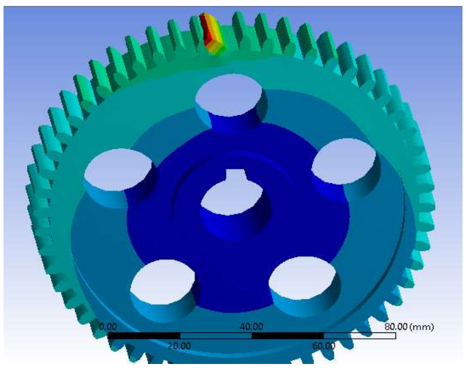

Fig. 8 Total deformation of Gear 1.

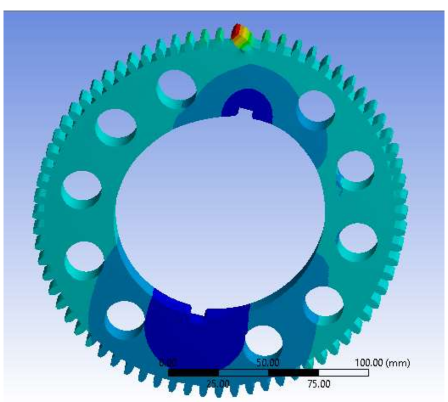

Fig. 9 Total deformation of Gear 2.

Table II: Ansys result-Total Deformation

Gear	Total Deformation (mm)
Pinion 1	0.01217 mm
Pinion 2	0.01204 mm
Gear 1	0.02096 mm
Gear 2	0.02046 mm

The results obtained from ANSYS static structural analysis are more than satisfactory as the effective force was assumes to be incident on a single tooth which does not actually happen. The value of total deformation is very low meaning that a single tooth can withstand the entire effective force transmitted.
Hence the analysis performed obtained satisfactorily good results and was successful.

V. EXPERIMENTAL SET-UP

Actual working model is manufactured and assembled according to the design and calculations obtained. The gearbox assembly is tested to check its reduction ratio. The gearbox is coupled to the engine with the help of V-belt and pulley system. Different input rpm is given and output rpm is noted down. To measure the torque difference, a rope-belt dynamometer is used.

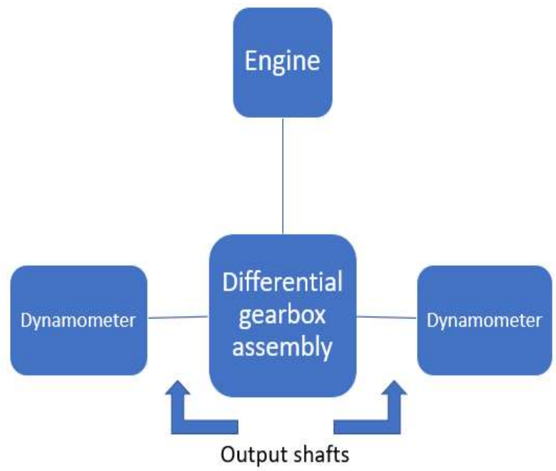

Fig. 10 Experimental set-up.

Fig. 11 Actual manufactured gearbox.

VI. RESULTS AND DISCUSSION

1. **Torque distribution:**
   The following table shows the difference in right and left torque vs. the input rpm taken. According to the experimental observations, torque splitting among the two wheels was equal due to the equal lengths of the output shafts achieved due to mounting the differential assembly centrally.

Table III: Dynamometer readings.

2. Steering stability:

Input RPM	Left side dynamometer Reading(Kg)	Right side dynamomete Reading (Kg)	Left side RPM	Right side RPM
1350	0.557	0.565	110.2	110
1350	0.630	0.615	109.6	109.5
1350	0.682	0.680	109.1	109.3
1350	0.710	0.706	107.8	107.9
1350	0.753	0.762	107.5	107.4

The steering characteristics also depend on the length of the output shafts of rear wheels of the vehicle. In conventional open differential, unequal shaft lengths cause uneven torque distribution (more torque is supplied to the shorter length shaft) which leads to poor steering performance. In centrally suspended cage-less differential, the shaft lengths are equal which eliminates the problem discussed above.

3. Weight Reduction:

The weight of this differential is comparatively lesser than the conventional differential due to elimination of the cage required to support the bevel gear assembly. 35% weight reduction is achieved.

VII. CONCLUSION

The entire methodology of designing and analyzing of this differential gearbox is demonstrated in this paper. All the observations and results emphasize the advantages of this differential gearbox over the conventional gearbox. Theoretical designs and calculations were validated using finite element analysis simulated on ANSYS workbench. The results show that the differential gearbox possessed an appreciable factor of safety and hence validating the designs.

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