IOSR Journal of Mechanical and Civil Engineering (IOSR-JMCE)
ISSN(e) : 2278-1684, *ISSN(p) :* 2320–334X, *PP : 59-66*
www.iosrjournals.org

Design, Manufacturing & Analysis of Differential Crown Gear and Pinion for MFWD Axle

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ABSTRACT : *The main objective of this paper is to perform mechanical design of crown wheel and pinion in differential gear box of MFWD (FWA) Axle (of TAFE MF 455). Detailed modeling, assembly and analysis of tooth of crown gear and pinion is explained which is performed in Pro-E.*

Keywords: - *MFWD, ANSYS, Crown wheel & pinion gear, Bevel gear, FEA.*

I. INTRODUCTION

Front wheel assist (FWA) is commonly known as 4WD or unequal 4-wheel drive Tractor is capable of delivering between 50-55% of the rated power at the drawbar typically between 75% and 85% of the rated engine HP is delivered to the rear PTO (Power Take Off) on any diesel tractor. On a FWA tractor the front drive tires are smaller than the rear tires. These tractors require 40% of the weight distributed over the front axle and 60% over the rear axle. The major advantage in using this type of tractor is that it can deliver 10% more power to the ground at all four tires for the same fuel consumption. 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 road wheels to rotate at different speeds, while for most vehicles supplying equal torque to each of them. A vehicle's wheels rotate at different speeds, mainly when turning corners. The differential is designed to drive a pair of wheels with equal torque while allowing them to rotate at different speeds. In vehicles without a differential, such as karts, both driving wheels are forced to rotate at the same speed, usually on a common axle driven by a simple chain-drive mechanism. When cornering, the inner wheel needs to travel a shorter distance than the outer wheel, so with no differential, the result is the inner wheel spinning and the outer wheel dragging, and this results in difficult and unpredictable handling, damage to tires and roads, and strain on (or possible failure of) the entire drive train.

II. LITERATURE SURVEY

1. Differentiation

The two Primary function of a differential system is to allow the driven wheels to rotate at different angular velocities relative to each other and also to transfer power to them from transmission. The first function is important to allow smooth turning as can be seen in the following Fig.1 which illustrates a left hand run.

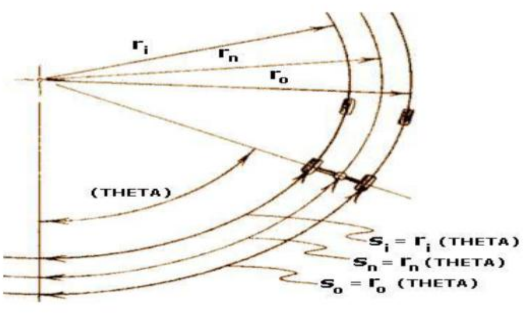

Fig.1 Turning radius diagram

Fig.2 An Open Differential

From Fig.1 it is clear that r_o is larger than r_i , therefore the outer wheel travelling along r_o has to cover more distance than the inner wheel r_i since both wheels are of the same diameter ,the outer wheel has to complete more revolution than the inner wheel to accomplish this and hence spin faster. If Differentiation in speeds was not allowed and the wheels were locked together along a single drive shaft, then during turning the outer wheel would lose traction and will slip. This would result in reduced turning performance and increased tire wear.

2. Types of differential

2.1 Open Differential

Open differentials (Fig.2) make use of a planetary gear set mechanism which distributes torque equally between the drive axles while allowing the wheels to rotate at different rates [6]. The input shaft transmits torque from the driveline to a large ring gear outside the differential carrier. When the vehicle is travelling in straight line the mechanism remains disengaged and the differential casing rotates at same rate as the drive axles. As the vehicle enters the turn the gear set engages and the meshing of the pinion gears allow the drive axle to rotate at different speeds [5].

a) Dynamic Model of Cornering

A model of the forces on the tires of the driving wheel has been presented by [6]. The forces transmitted by a tire to the ground are separated into longitudinal and lateral components. The longitudinal force is responsible for forward motion. When torque applied to the wheel exceeds the maximum longitudinal force the tire can transmit, slippage results, the maximum force transmittable by the tire is a function of the normal force acting against the bottom surface of the tire and thus also a function of the amount of load above the tire. During high speed cornering, the weight of a vehicle shifts away from the turning direction due to inertia creating a higher load on the outer wheel and a smaller load on the inner. The maximum transmittable longitudinal force is therefore reduced for the inner tire and increased for the outer tire, thus the outer tire becomes capable of handling more torque and the inner tire, less (Fig.3). As torque is divided equally between both drive axles in an open differential, the inner wheel soon experiences more torque than its tire can transmit causing slippage and loss of control. In high performance conditions, this instability can pose a significant risk to the occupants safety.

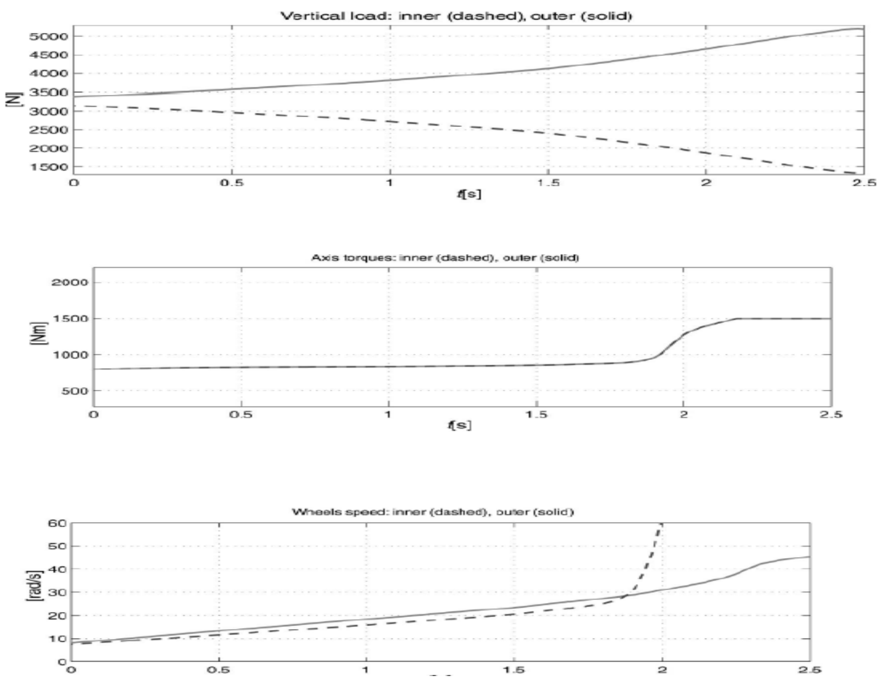

Fig.3 Graphical representation of the vertical loads, axial torques, and rotation speed of the inner & outer wheel

b) Redistributing Torque Appropriately

Ideally the outer wheel ought to receive a progressively larger amount of torque over time as the load shifts. Several disparate solutions have been developed to mitigate the problems experienced by open differentials during the negotiation of turns at high speed.

2.2 Limited Slip Differentials:

a) Clutch Plate Limited Slip Differentials (Fig.4)

One of the earliest solutions to the problem of inappropriate torque distribution between axles is the clutch plate differential. This unit functions by making use of the difference in speed of the inner and outer wheels during a turn to improve traction. They employ a set of friction plates on either side of the differential case. Half of these plates are meshed with the profile of the case whilst the other half is meshed with each of the drive axles. In straight line driving the carrier and the axles turn at the same rate as the differential gearing remains disengaged. At the beginning of a turn, the outer wheel begins to spin faster than the inner wheel resulting in the engagement of the differential and initially the driveshaft torque is still equally distributed between the two axles. The differentiation causes the friction plates to turn in opposite directions and produce a restraining force on the mechanism to keep each of the axles spinning at the same rate. As mentioned above, the speed of the inner wheel begins to increase due to the initial excess torque transfer and once it has become equal to that of the outer wheel, the friction plates ensure that it does not rise any further. As a result, more torque is gradually transmitted to the outer wheel as its vertical load increases. The torque and axle speed characteristics are outlined below Fig. 6 [6].

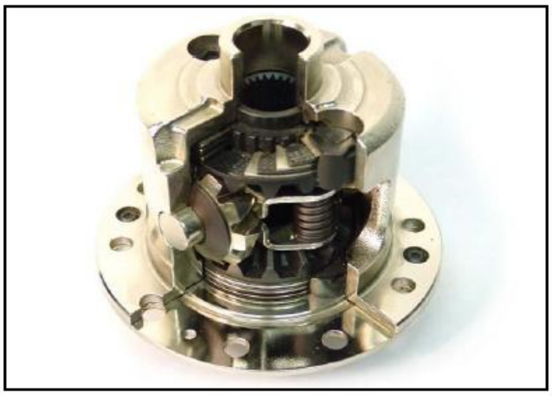

Fig.4 Clutch plate limited slip differential

Fig.5 Viscous limited slip differential

b) Viscous Limited Slip Differential

Similar in many ways to the clutch plate design, these mechanisms employ viscous couplings joined to an open differential. A set of friction surfaces is surrounded by one of several viscous fluids which have the property of solidifying at high temperatures. As the two axles begin to spin at different rates, the friction surfaces rotate in the fluid performing work and thus generating heat. The increasing viscosity of the fluid provides the resistance to differentiation. Engagement is slightly delayed as the fluid cannot heat up instantaneously. If the situation becomes so extreme that one wheel has almost no traction at all (either due to a very high speed while cornering or as a result of being on a nearly frictionless surface like ice), the resistance provided by the fluid may not be applied quickly enough. One of the shortcomings of this type of differential is that the work generated in the fluid can eventually exceed a critical limit that it can tolerate [8]. The result is a partial or complete loss of its mechanical properties. These differentials are difficult and expensive to repair since a unique fluid is used. (Fig. 5).

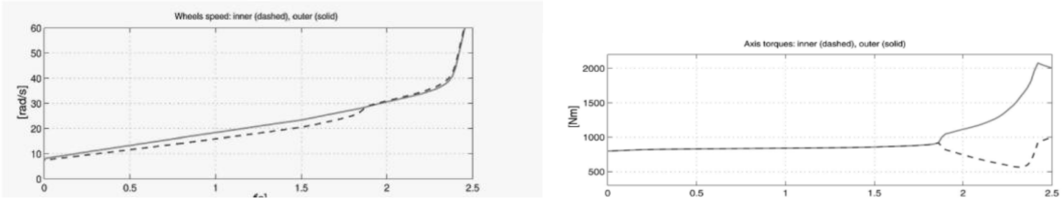

Fig.6 Torque and speed characteristic of inner and outer wheel

III. BEVEL GEAR TERMINOLOGY

Fig.7 shows the bevel gear terminology used in this work.

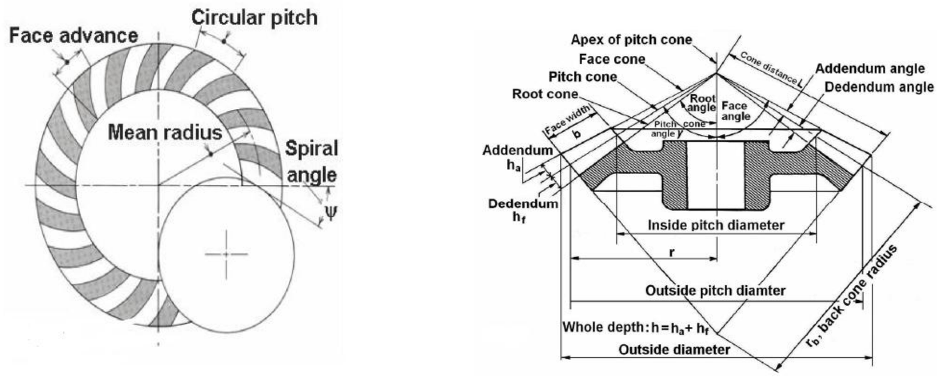

Fig.7 Bevel gear terminology

IV. PROBLEM FORMULATION AND OBJECTIVE

Problem statement: Design manufacturing and FE analysis of Differential Crown Gear and Pinion for MFWD Axle's for TAFE MF 455 with specifications as shown in Table I.

TABLE I TECHNICAL DETAILS

Vehicle Data	Front Axle	Rear Axle
Laden Weight Distribution		
(Road Transfer)	2500kg	3500kg
Wheel Tread (Min-Max)	1470-1870(mm)	-
Max.Load (Breakout) :	4500kg	-
Engine Data		
Power gross/Engine Speed	71.6kw/2200 rpm	-
Max.torque/Engine speed	402Nm/1400rpm	
Tyres Data		
Index Radius	0.65m	
Front Axle Specification		
Bevel Set ratio	2.9	
Final Drive Ratio(epicyclic)	6	Total Ratio=22.44

V. MANUAL DESIGN CALCULATIONS FOR CROWN GEAR AND PINION

Slip torque = laden wt. on front axle $\times$ index radius

= 2500 $\times$ 9.81 $\times$ .65

= 15941.25 Nm

Peak torque (M_p) = 1.5$\times$ Slip torque

=1.5$\times$ 15941.25 = 23911.875 Nm

Torque transmitted by pinion (M_t)

$$M_t = \frac{M_p}{i_s \times i_b}$$

Where, i_s =epicycle gear ratio,

i_b =Conic set ratio

$$= \frac{23911.875}{6 \times 2.9}$$

$$= 1374.245 \times 10^3 \text{Nmm}$$

Now, tangential load on pinion tooth (P_t)

$$P_t = \frac{2M_t}{D_m}$$

But $r_m = \frac{D_p}{2} - \frac{b \sin\gamma_g}{2} = \frac{m \times Z_p}{2} - \frac{b \sin\gamma_g}{2}$ where r_m=mean radius(mm)

taking , $\frac{b}{A_0} = 0.3$ where , A_o=cone distance

$$A_o = \sqrt{\left(\frac{D_p}{2}\right)^2 + \left(\frac{D_g}{2}\right)^2}$$

$$= m\sqrt{\left(\frac{11}{2}\right)^2 + \left(\frac{32}{2}\right)^2} \qquad \text{as, } Z_p = 11 \text{ and } Z_g = 32$$

= 16.9189 m

b = 0.3 $\times$ A_o = 5.0757m

$$r_m = m \times \frac{11}{2} - \frac{5.0757m \times \sin 18.97}{2} \qquad \ldots\left(\gamma_p = \tan^{-1}\left(\frac{11}{32}\right)\right)$$

=4.675 m …...pinion

D_m = 9.350 m

$$P_t = \frac{2M_t}{D_m} = \frac{2 \times 1374.245 \times 10^3}{9.35m}$$

$$P_t = \frac{293.956 \times 10^3}{m}$$

Material selected = 18_NiCrMo5

$$P_t = \sigma_{b_{all}} \times C_v \times b \times \pi \times m \times y' \times \left(\frac{A_0 - b}{A_0}\right)$$

Where, $\sigma_{b_{all}}$ = 1/3× S_{ut} =1/3× 2069=698.667 N/mm^2 (S_{ut} =2069 N/mm^2)

$$V_m = \frac{\pi D_m N}{60} = \frac{\pi \times 9.35m \times 409.53 \times 10^{-3}}{60} = .2005m$$

$$\text{(Velocity factor)}\ C_v = \frac{3.5 + \sqrt{V_m}}{3.5} = \frac{3.5 + \sqrt{0.2005m}}{3.5} = 1 + 0.128\sqrt{m}$$

Where, $V_m = \frac{2\pi \times 402 \times 1400}{60} = \frac{2\pi \times N \times 1400}{60}$

N=409.53 rpm

$T_{EP} = \frac{Z_p}{\cos\gamma_p \times \cos^3\beta}$ ……. $(\gamma_p = 18.97^\circ, \beta = 34.88^\circ)$, (T_{EP}=equivalent teeth on pinion)

$$T_{EP} = \frac{11}{\cos(18.97) \times \cos^3(34.88)} = 21$$

$$y' = 0.154 - \frac{0.912}{T_{EP}} = .111$$

Putting all value in (*) we get

$$\frac{293.956 \times 10^3}{m} = 698.667 \times (1 + 0.128\sqrt{m}) \times 5.0757m \times \pi \times m \times 0.111 \times .7$$

m = 6.38 mm

Selecting standard m =6.50 mm

$\therefore D_m = 9.35m = 9.35 \times 6.5 = 60.775\text{mm}$,

b=5.0757m =33mm,

A_0 =16.9189m=109.973mm

V_m =0.2005m=1.30 mm/s , $C_v = \frac{3.5 + \sqrt{V_m}}{3.5}$ =1.3257

$$\frac{293.956 \times 10^3}{6.5} = \sigma_{b_{actual}} \times (1 + 0.128\sqrt{6.5}) \times 5.0757 \times 6.5 \times \pi \times 6.5 \times 0.111 \times .7$$

$\sigma_{b_{actual}}$ = 651.517 N/mm^2

$\sigma_{b_{actual}} < \sigma_{b_{all}}$

So design is safe at peak load

Now for 90% duty cycle

$P_{t(working)}$= 0.5 × P_t

$=0.5\times 45.224$ KN

$=22.612$KN

Checking for continuous working

$$P_t = \sigma_{b_{actual}} \times C_v \times b \times \pi \times m \times y' \times \left(\frac{A_0 - b}{A_0}\right) \quad .....*$$

$$22.612 = \sigma_{b_{actual}} \times 1.3257 \times 33 \times \pi \times 6.5 \times .111 \times \left(\frac{109.973 - 33}{109.973}\right)$$

$$\sigma_{b_{actual}} = 325.79 N/mm^2$$

$$\sigma_{b_{actual}} < \sigma_{b_{all}}$$

$\therefore$ Design is safe.

VI. MODELLING AND FINITE ELEMENT ANALYSIS

After completing design of crown gear and pinion theoritically by using given data (Table.1) and to perform FE analysis of any component, the solid model (Fig. 8) of the same is essential, so we modeled crown gear (Fig.8) and pinion in PRO-E (wildfire4.0) which is excellent CAD software , which makes modelling so easy and user friendly.The model is then transferred in IGES format and exported into the Analysis software ANSYS 13.0 .We analysed gear tooth (Fig. 9) in ANSYS by importing IGES file of tooth. First is pre-processing which involves modeling, geometric clean up, element property definition and meshing. Next comes, solution which involves imposing boundary conditions and applying loads on the model and then solution runs. Next in sequence comes post processing, which involves analyzing the results plotting different parameters like stress, strain.

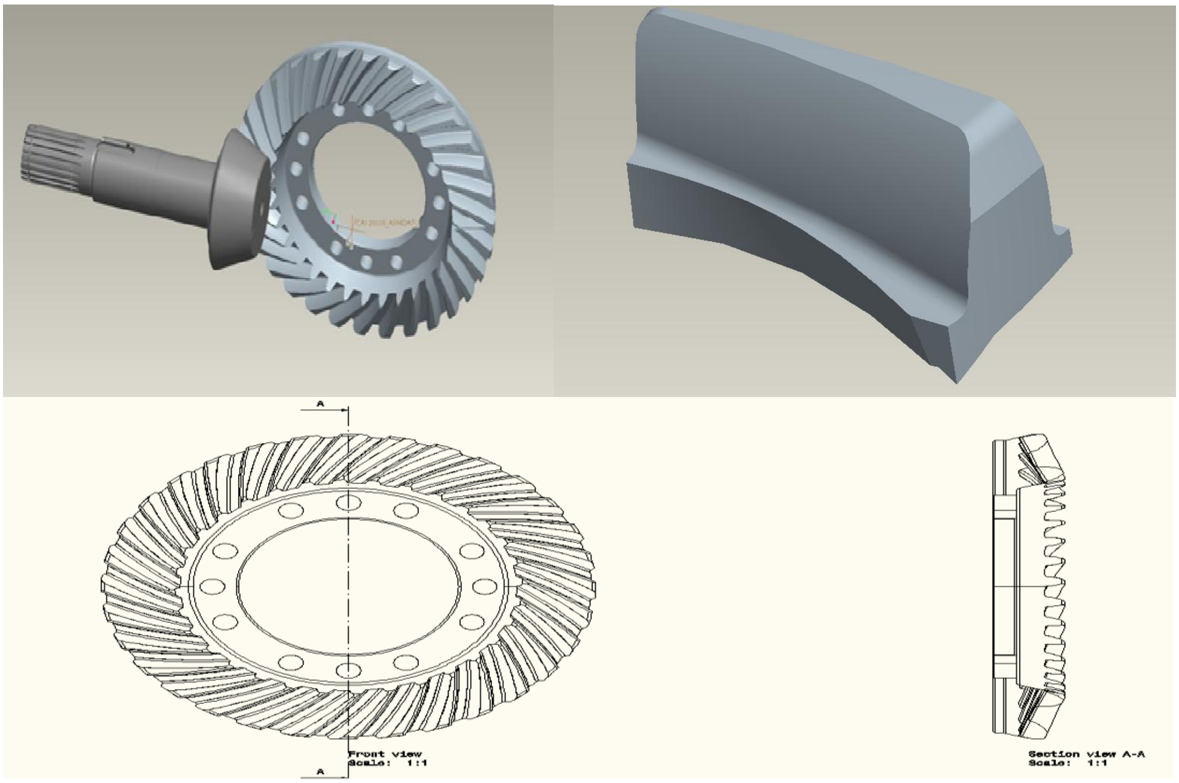

Fig. 8 3-D model of assembly and tooth 2-D view of Crown gear

VII. FE ANALYSIS RESULTS

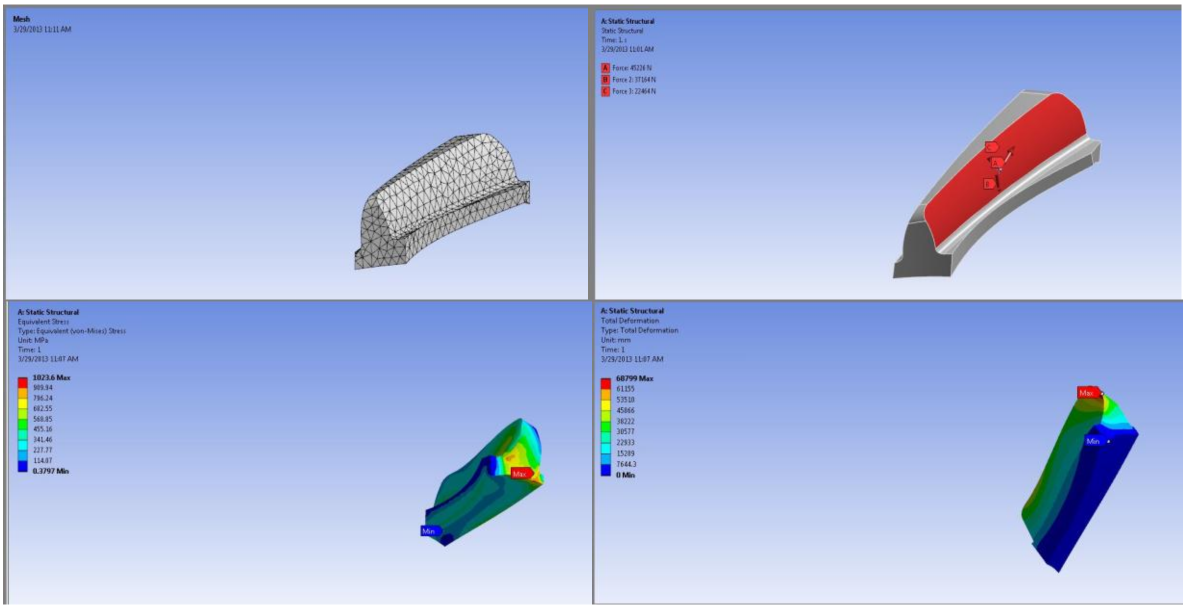

Fig.9 Finite element analysis of crown gear tooth

VIII. CONCLUSION

In this work detailed manual and computer aided designing of crown gear and pinion is carried out. Finite element analysis is performed to analyse the crown gear tooth for working load and from Fig. 9 it is seen that equivalent stress on tooth is approximately 682 N/mm^2, which is less than $\sigma_{b_{all}}$, i.e. 698.667N/mm^2. Therefore, from it is concluded that design is safe.

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