ANALYSIS OF GEAR PAIR CONTACT TO PRIDICT EFFICIENCY

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Abstract—**Gear pair is higher pair point contact is under the stress. Due to that point contact get converted into line contact. Deformation due to contact losses some energy during meshing of gear is generally neglected, but so many researchers calculate efficiency of gear drive by practically and analytically is upto 98% for metal only.if the error due to deformation in transmission in gear box is small no one proposed this kind work is workout in this paper**.

Keywords— **Gear pair, Contact, Deformation, Efficiency.**

I. INTRODUCTION

Efficiency of spur and helical gear systems has become an increasingly important research topic as the fuel economy requirements for today's passenger vehicles and rotorcraft are more stringent, not only due to fuel cost, but also environmental concerns associated with energy utilization and air pollution. Improved gear system efficiency also results in less frictional heat generation within the gearbox, resulting in improvements in gear failure modes such as scoring and pitting, and lower-capacity lubrication systems. Sliding friction contributes about 98% of the total power loss for gear trains operating at relatively low speeds (less than 2000 rpm input speed). Rolling frictional losses decrease with increased load while windage losses are only significant for gears running at very high speeds (greater than 3000 rpm). The results also showed that the overall efficiency varies over the path of contact of the gear meshes ranging between 94% to 99.5%.

There are other benefits to improving gear efficiency as well. Since the efficiency losses amount to additional heat generation within the gearbox, several gear failure modes including scoring and contact fatigue failures are directly impacted by the efficiency of the gear pair. A more efficient gear pair generates less heat, and therefore, it is likely to perform better in terms of such failures. In the process, demands on the capacity and the size of the lubrication system and the amount and quality of the gearbox lubricant are also eased with improved efficiency. This also reduces the overall weight of the unit contributing to further efficiency improvements.

The total efficiency loss of the gearbox is attributable to sliding and rolling frictional losses between the gear teeth, windage losses due to complex interactions with the air surrounding the gears, and oil splashing and churning losses inside the gearbox, as well as the losses associated with the bearing and seals. While churning and windage losses are mostly geometry and speed related, friction losses are mainly associated with sliding velocities and load. Friction losses of hypoid gears are of primary interest here since they are a major source of losses in a drive train. As a common automotive application, the drive train of a rear-wheel-drive (or all-wheel-drive) car or truck contains one (or two, one rear and one front) axle

gearbox that is formed by a hypoid final drive reduction unit and a differential. Unlike parallel axis gears that may have a mechanical efficiency well over 99 percent, the efficiency of hypoid gears usually falls into the range of 86 to 97 percent depending on the amount of relative sliding induced the gear geometry. This is mainly because sliding velocities in a hypoid gear contact are significantly larger than parallel-axis gears. As a result of this, an axle unit that is formed by a single hypoid gear set has the same levels of (and in May applications more) power loss as a manual or automatic transmission that contains many spur or helical gears. This presents the hypoid gear drive as a prime candidate for any efficiency improvement efforts.

Until now many researchers have worked on static and dynamic contact of contact with

Gosselin etal [1] applied a new methodology based on FEM to analyze the contact stress and displacement of the line contact condition in spur gears.

Braucr Zanzi and Pedrero and Guing and etal [2] on Tooth contact analysis and stress analysis of gear drives.

Umezawa and Ishikawa [3] presented experimental studies for the deflections and moments of a gear tooth.

Chabert et al also presented valuable results using finite element method (FEM) on the stresses and deflection of spur gear teeth induced by a static load.

Litvin etal [1] developed computer programs that integrate computerized design, tooth contact analysis and automatic modeling and finite element simulation of a new type of helical gear drives.

Vedmar and Henriksson introduced a method to use the deformations of the gears to determine the variation of the gear mesh stiffness and the boundary of the action between the gears.Little research has been reported on finite element based methods to investigate the dynamic contact/impact problems of gear drives.

GEARS are the most common power transmission systems in industrial machinery, automobiles, aircrafts, marine vessels et cetera. The efficiency of these power transmission systems is an important design factor due to the following

reasons:

i. Efficient power transmission systems ensure fuel economy of automobiles, marine vessels and aircrafts.

ii. With less fuel consumption, less pollutant gases and particulate are emitted to the environment

As the recent work is on the gear contact, when gear pair is rotated due tensional effect due to that causes the small deformation in the profile causes minor loss in energy that error affects in transmission efficiency.

1.   Gear pair is the higher pair with line contact under the stress , line contact get converted into surface contact i.e. lower pair due to that degree of freedom is zero hence motion will not possible, law of gearing is not possible at all point due to series of deformation.

2.   The motion will takes place because of the additional deformation with satisfaction of law of gearing.

3.   Resistance to the motion is due to :
a. Friction   b. Additional deformation

4.   Tractive forces causes the deformation at point of contact, hence some Energy is utilized for Local Deformation. Practically, complete Strain Energy is not regained under elastic deformation.

NOMENCLATURE:

Z1	No. of teeth on pinion
Z2	No. of teeth on gear
α	Pressure Angle
m	Module
dp	Pitch circle Diameter of Gear1
dg	Pitch circle Diameter of Gear2
A	Addendum of gear
D	Dedendum of gear
t	Tooth thickness
w	Face width
P	Circular pitch
F	Applied load vector
Pmax	Maximum contact stress
Fi	Load per unit width
σH	Maximum Hertz stress
a	Contact width
h	Working depth
H	Whole depth
Ac	addendum circle diameter
Dc	dedendum circle diameter
CR	Contact Ratio
CL	Contact Length
R1	Radius of Gear1
R2	Radius of Gear2

L	length of cylinder
Ft	Tangential Force
Fr	Radial Force
Fn	Normal Force
Ei	Young's Modulus of elasticity
μ	Poisson's ratio
∂x	in x-direction

∂y	Deformation in y-direction
∂z	Deformation in z-direction
σ	Normal Stress
τ	Shear Stresses
σ_{xx}, σ_{yy}, σ_{zz}	Normal Stress in X, Y and Z Direction
ε_{xx}, ε_{yy}, ε_{zz}	Normal Elastic Strain in X, Y and Z Direction
τ_{xy}, τ_{yz}, τ_{xz}	Shear stress component in XY, YZ and XZ Plane
γ_{xy}, γ_{yz}, γ_{xz}	Shear strain component in XY, YZ and XZ Plane

II. CONTACT ANALYSIS OF TWO CYLINDRICAL

1. Why is the Contact Problem Significantly Difficult

Despite the importance of contact in the mechanics of solids and its engineering applications, contact effects are rarely seriously taken into account in conventional engineering analysis, because of the extreme complexity involved. Mechanical problems

involving contacts are inherently linear elastic behavior.

First, the actual region of contact between deformable bodies in contact is not known until the solution has been obtained. Depending on the loads, materials, and boundary conditions, along with other factors, surfaces can come into and go out of contact with each other in a largely unpredictable manner. Secondly, most contact problems need to account for friction. The modelling of friction is very difficult as the friction depends on the surface smoothness, the physical and chemical properties of the material, the properties of any lubricant that might be present in the motion, and the temperature of the contacting surfaces. There are several friction laws and models to choose from, and all are nonlinear. Frictional response can be chaotic, making solution convergence difficult (ANSYS). In addition to those difficulties, many contact problems must also address multi-field effects, such as the conductance of heat and electrical currents in the areas of contact. Bodies in contact

may have complicated geometries and material properties and may deform in a seemingly arbitrary way. With the rapid development of computational mechanics, however, great progress has been made in numerical analysis of the problem. Using the finite element method, many contact problems, ranging from relatively simple ones to quite complicated ones, can be solved with high accuracy. The Finite Element Method can be considered the favourite method to treat contact problems, because of its proven success in treating a wide range of engineering problem in areas of solid mechanics, fluid flow, heat transfer, and for electromagnetic field and coupled field problems. An easy way to comply with the conference paper formatting requirements is to use this document as a template and simply type your text into it.

2 How to Solve the Contact Problem

There are many types of contact problems that may be encountered, including

contact stress, dynamic impacts, metal forming, bolted joints, crash dynamics, and assemblies of components with interference fits, etc. All of these contact problems, as well as other types of contact analysis, can be split into two general classes (ANSYS),

Rigid - to - flexible bodies in contact,

Flexible - to - flexible bodies in contact.

In rigid - to - flexible contact problems, one or more of the contacting surfaces are treated as being rigid material, which has a much higher stiffness relative to the deformable body it contacts. Many metal forming problems fall into this category.

Flexible-to-flexible is where both contacting bodies are deformable.

Examples of a flexible-to-flexible analysis include gears in mesh, bolted joints, and interference fits.

3. Types of Contact Models

In general, there are three basic types of contact modelling application as far as

ANSYS use is concerned. Point-to-point contact: the exact location of contact should be known beforehand. These types of contact problems usually only allow small amounts of relative sliding deformation between contact surfaces. Point-to-surface contact: the exact location of the contacting area may not be known beforehand. These types of contact problems allow large amounts of deformation and relative sliding. Also, opposing meshes do not have to have the same discretisation or a compatible

mesh. Point to surface contact was used in this chapter. Surface-to-surface contact is typically used to model surface-to-surface contact applications of the rigid-to-flexible classification.

3.2.3 Why contact between two cylindrical Surfaces

Contact of two curved surface is represent the gear teeth so two cylindrical surfaces in contact cuases the contact stress and deformation, in machine design,problem frequently occurs when two members with curved surfaces are deformed when press against one another giving rise to an area contact under the compressive stresses. Of particular interest to the gear designer is the case where the curved surfaces are of cylindrical shape because they closely resemble gear tooth surface

Consider two circular cylinder, B1 and B2, with a radius of R1= 5 mm. and R2= 5mm. as shown in Figure 3.2
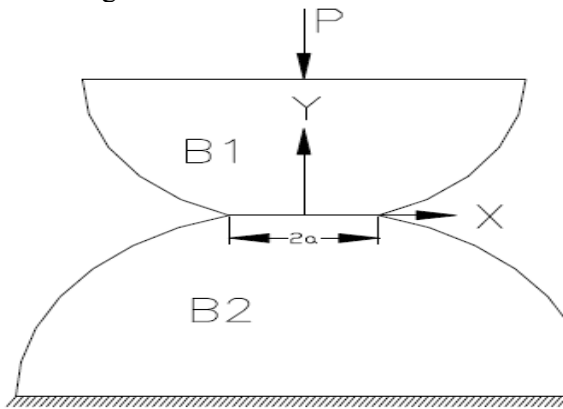
3.2.3.1 Hertz Contact Stress Equations:

Usually, the current methods of calculating contact stresses use Hertz's equations, which were originally derived for contact between two cylinders.

The maximum surface (Hertz) stress:

$$Pmax = \sigma H = 0.564 \frac{\sqrt{F\left(\frac{1}{R1}+\frac{1}{R2}\right)}}{\sqrt{\left(\frac{1-\mu1^2}{E1}+\frac{1-\mu2^2}{E2}\right)}}\ldots\ldots\ldots\ldots\ldots(1)$$

Where,

F -is load per unit length,
Ri- is radius of the cylinder,
μi- is Poisson ratio,
Ei- is Young's modus of cylinder.

Material properties of the two cylindrical bodies and load data of the contact model are shown in Table 4.1.

Sr. No	Material Property and Parameter		
	Surface Body	MaterialProperty	Dimension
1	Cylinder 1 (B1)	1.Modulus of Elasticity (E1)=200X10^5Pa 2. Poisson's ratio (μ¹)=0.3	Radius (R1)=5mm, Length (L) = 60mm
2	Cylinder 2 (B2)	1.Modulus of Elasticity (E 2)=200X 10^5Pa 2. Poisson's ratio (μ²)=0.3	Radius (R2)= 5mm, Length (L) = 60mm
3	Load Data:	Load (F) = 2000 N	

Following fig 3.3 (a) and fig 3.3(b) shows the deformation and von misses stress contact zone is near contact surface.
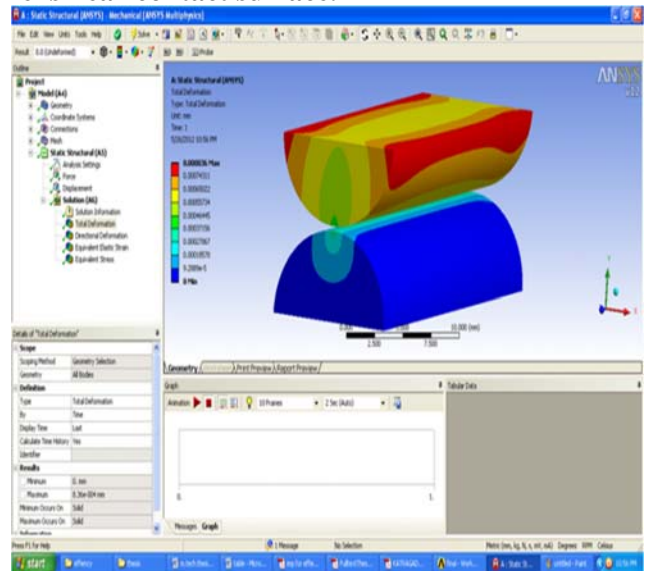

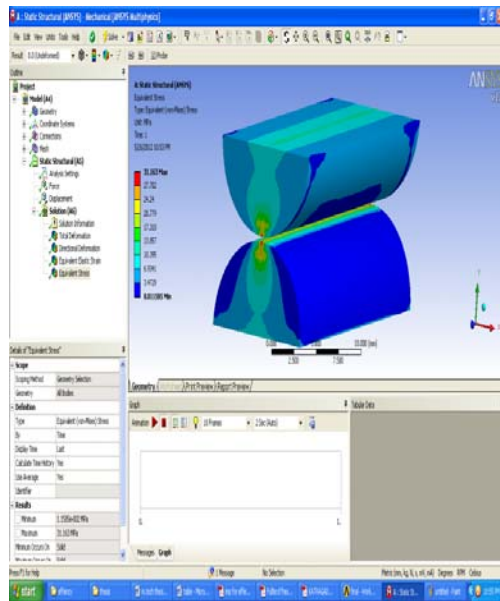**Figure a): Deformed Model two half cylinder surfaces**

Figure b): Von Mises Stress Model of two half cylinder surfaces

Table 3.1: Comparison of finite element analysis results with analytical results for contact model

REGION	ANSYS	THEORO TICAL	% ERROR
CONTACT DEFORMATI ON(mm)	0.00 0836	0.00092 8	1.10
CONTACT DEFORMATI ON(mm)	31.6 4	25.467	0.5

Conclusion: Finite element modeling of the contact between two cylinders was examined in detail. The finite element method with special techniques, such as the incremental technique of applying the external load in the input file, the deformation and contact stress and the introduction of the contact element were used. It was found that initial loading using displacements as inputs was helpful in reducing numerical instabilities.

III.Force Analysis for Spur Gear

One of the fundamental parameters to be considered, analyzed and checked for designing a gear system is the load transmitting capability of gear teeth. For this the circumferential force effective on the tooth at the pitch circle of the gear when in mesh, must be known. Irrespective of the value of the contact ratio, for the calculation the total gear forces are taken to be effective on a single pair of teeth in mesh.

Rated Power P_R = 30 KN,

Speed on Pinion = N_P = 1000 RPM

Speed on Gear = N_G = 500 RPM

Z_1 = No. Of Teeth on the Pinion = 18

Z_2 = No. Of Teeth on the Gear = 18

Pressure Angle α = 20 deg for Full Depth

Torque on Pinion & Gear T_P & T_G = $\frac{P \times 60}{2\pi N_P \text{ or } N_G}$ = $\frac{30\,000 \times 60}{2\pi \times 1000}$

$= 286.4788$ N-mm

Material for Gear and Pinion = Structural Steel

Brinell hardness number (B.H.N) =200

Assuming the steady state condition& 8-10 hrs. service per day

4.1. Design Power (P_d) = $(P_R) \times K_1$ = $30 \times 10^3 \times 1 = 30 \times 10^3$ W

Where, P_R = Rated Power, K_1 = Load Factor = 1

4.2. Tooth Load $F_{t, N}$

$F_t = \frac{(P)d}{V_p}$

Where V_p = Pitch Line Velocity, m/sec

$V_p = \frac{\pi D_p N_p}{60 \times 1000} = \frac{\pi \times m \times 18 \times 1000}{60 \times 1000} = 0.9424$ m/sec

$$F_t = \frac{30\times10^3}{0.9424m} = \frac{31830.9886}{m} \ldots\ldots (1)$$

4.3 Bending Strength by Lewis equation, F_B ,N

$$F_B = S_o \times C_v \times b \times Y \times m$$

Where,

S_o= Basic Strength, MPa= 140MPa

C_v = Velocity Factor = 3 / 3 +Vp = 0.3 (trial value)

b= Face width of gears =10mm (trial value)

Y =Modified Lewis form factor = 0.485 − 2.87 / Tp

=0.485 − 2.87/18

= 0.3256

m= Module, mm

F_B=140×0.3×10m×0.3256×m=138.74 ×m²....... (2)

Equating Equation (1) & (2), we get

F_B = Ft

138.74 ×m² = 31830.9886/m

m³ = 229.4290

m=6.1218 ≈ 8mm

Therefore,

The standard value of module m =8mm

Face width b= 10m = 10×8=80 mm

Diametral Pitch for pinion dp = 8 ×18 =144 mm

Diametral Pitch for gear dg = 8 ×18 = 144 mm

Pitch Line Velocity, Vp= 0.9424 ×m =0.9424 ×8 =7.5392 m/sec

Ft = 31830.9886/m =31830.9886/8 =3,978.8735 N

F_B=138.74×m² = 138.74 ×8² =8,879.36 N

4.4 Dynamic Load Fd, N Based on Buckingham's Equation

The dynamic force for the gear is given by

$$F_d = F_t + \frac{21Vp\,(Ceb+Ft)}{21Vp+\sqrt{Ceb+Ft}}$$

The permissible error at 7.5392 m/sec i.e. 452.352 m/min is 0.045.

We must use class-2 chain gear for which the permissible error for which the probable error

For 8mm module will be 0.4mm.For deformation factor C for steel with 20⁰ full depth teeth

C= 11800

Ceb = 11800×0.04×80 = 37,760

Ceb + Ft = 37,760 + 3,978.36= 41,738.36

21Vp = 158.34

$$F_d = 3,978.36 + \frac{158.34\times(41,738.36)}{158.34+\sqrt{41,738.36}}$$

= 22,203.33 N

4.5. Limiting Wear Strength Fw, N

$$F_W = D_P \times b \times K \times Q$$

Where, K= Load Stress Factor

Dp= 144mm

b= 80 mm

tg & tp =No. of teeth on gear & pinion respectively = 18

Q = Size Factor = 2tg / tg+tp =1….for internal gear

Fw = 144 ×80×1×k

= 11520 k……….……….(3)

Equating Fw to Fd, We get kmin;

Kmin = 22203.33/11520=1.9273

Hardness for pinion & gear= 350 B.H.N

4.6. Endurance strength, Fen, N

Fen = Seb×b×Y×m

Seb = Endurance Strength =596 MPa for Steel with 350B.H.N

b= 80mm

Y= 0.1034

Fen= 596×80×0.1034×8

   = 39440.8960N

The greater than Fd   so O.K.

IV. 3D CONTACT ANALYSIS OF GEAR

5.1 SPUR GEAR PAIR MODEL

This model of spur gear is made in catia V5R20 then this model is import in though IGES format in ANSYS 12.1 Software. It can be seen in fig 2.11 that during the course of action of the teeth engagement along the path of action in a meshing pair of gear teeth, i.e. from beginning of contact to the end of the contact comprising the length of contact ,the load during rest of the time.
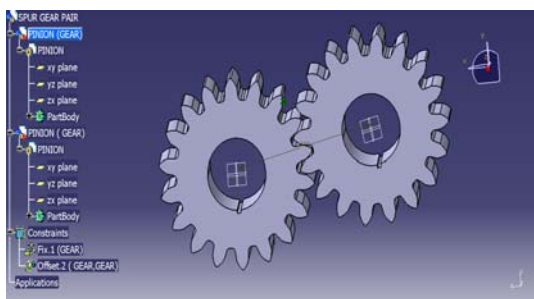
The following parameters are used to design the pair of spur gear.

5.2 Contact Ratio

It can be seen from fig that during the course of action of teeth engagement along the path of action in a meshing pair of gear teeth, i.e. from beginning of contact to the end of contact comprising the length of contact, the load is transmitted by a single tooth of the driving gear for part of the time and by two teeth during rest of the time. For continuous contact, the angle of action must be greater than the angle subtended at the centre by the arc representing the circular pitch (called "pitch angle").the relation between these two angles is termed as the contact ratio.

The physical significance of the contact ratio lies in the fact that it is a measure of average number of teeth in contact during the period in which a tooth comes and goes out of contact with the mating gear. Contact ratio 1 means that only one pair (one tooth from each gear) is engaged at all times during course of action. This is the case when the angle

The zone of action of meshing gear teeth is shown in Figure 4.4. Tooth contact begins and ends at the intersections of the two addendum circles with the pressure line.

In figure, initial contact occurs at a and final contact at b. Tooth profiles drawn through these points intersect the pitch circle at A and B, respectively. As shown, the distance AP is called the arc of approach ua, and the distance P B, the arc of recess ur. The sum of these is the arc of action AB= La.

Now, consider a situation in which the arc of action is exactly equal to the circular pitch= p.This means that one tooth and its space will occupy the entire arc AB. In other words,

When a tooth is just beginning contact at a, the previous tooth is simultaneously ending its contact at b. Therefore, during the tooth action from a to b, there will be exactly one pair of teeth in contact.

According to Machinery's Handbook [8-9], the contact ratio is the ratio of the arc of action in the plane of rotation to the circular pitch.
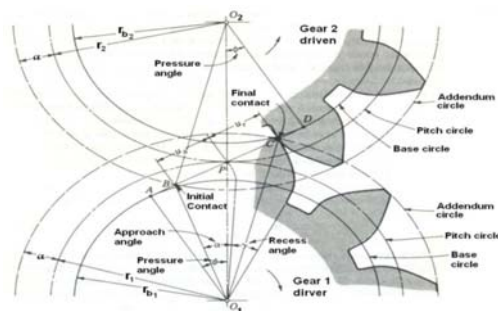
Referring to Fig,

The path of contact La = ua+ ur =Arc of approach (AP) + Arc of recess (PB)

La =√ (〚 (r1+a) 〛 ^2- 〚r1〛 ^2×cos^2 α) + √ (〚 (〚r2〛 ^2+a) 〛 ^2- 〚r2〛 ^2×cos^2 α) - (r1 + r2) sin α

= √ (〚80〛 ^2- 〚72〛 ^2×cos^2 20) + √ (〚80〛 ^2- 〚72〛 ^2×cos^2 20) - (72+72) sin 20

= 42.6897 + 42.6897 - 49.2509

= 36.1284mm

The contact ratio is given by following formula

Circular pitch = pitch angle = Base pitch

 = π×m Cos 20

= π×8 cos20

=23.6170

C.R = (the path of contact)/ (circular pitch)

= La/(π×m Cos 20)

= (36.128)/ (23.6170)

= 1.530................................... (1)

Table 5.1 Design Parameter of Spur Gear

Parameter	Formulas	Driving Gear	Driven Gear
Profile		Involute	involute
No. of teeth	Z1/Z2=1	Z1 =18	Z2=18
Pressure angle	α	20 deg	20 deg
Module	m	8 mm	8mm
Circular pitch	P= π m	25.1328 mm	25.1328 mm
Pitch circle diameter	d= m*z	144 mm	144 mm
addendum	a = 1m	8mm	8mm
Dedendum	b = 1.25m	10mm	10mm
Tooth thickness	t =1.57m	12.56 mm	12.56 mm
Face width	b	80 mm	80 mm
Working depth	H =2m	16 mm	16mm
Whole depth	h=2.25m	18mm	18mm
addendum circle diameter	Ac = d+2m	160mm	160mm
dedendum circle diameter	Dc = d- 2m	128 mm	128 mm
Fillet radius	R_f= 0.4m	3.2 mm	3.2 mm
Top land	tp= 0.25m	2mm	2mm
Centre distance	C = (z1+z2)m|2	144mm	144mm
Contact ratio	CR = L_a/Pcosα	1.5297	1.5297
Power (P)	P= 2¶NT/60	30 KN	30 KN
Speed (N)	N	1000 rpm	500rpm
Torque(T)	T	286.32 N-mm	572.95N-mm
Tangential Force	Ft	3978.8748 N	
Normal Force	Fn=Ft/cosα	4234.2301 N	
Radial force	Fr=Ft *tanα	1448.1920 N	

4 Meshing of spur gear

In this session gear teeth in mesh contact causes the deformation in micron over the gear teeth and contact stress at point of contact is maximum. This gear tooth mesh model there is total nodes no 59790 to 61549 and element number 38643 to 39908 on this node in contact, minimum elemental edge length is 0.00984mm.
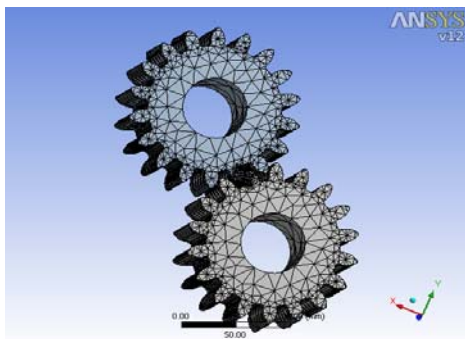

Fig 5.2 Fine Mesh of Spur Gear near Contact

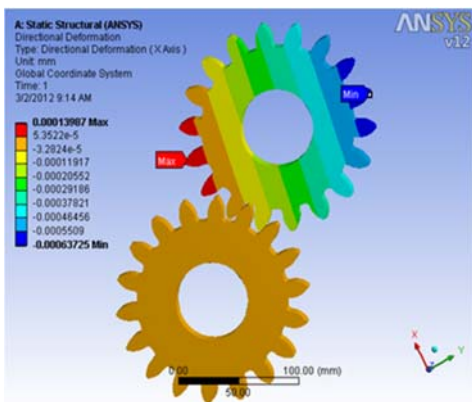## 5.2 DEFORMATION, ENERGY AND EFFICIENCY OF GEAR TEETH
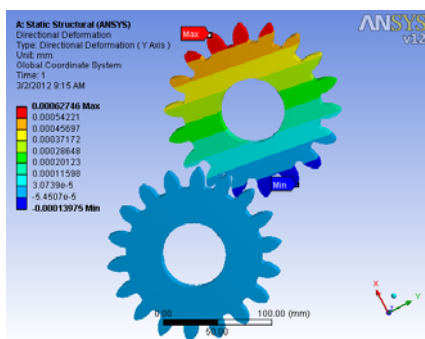

Fig: a) Deformation Model in X- Direction

Fig: b) Deformation Model in Y- Direction
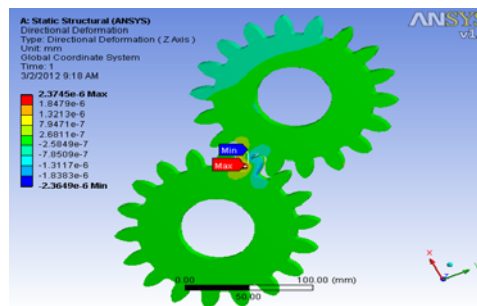
Fig: c) Deformation Model in Z- Direction

Fig 5.3: 3D Deformed Model for gear under contact.

5.2.2 Methods for Energy Estimation under Strain

The following types of energy method used to calculate the gear pair losses

5.2.2.1 Strain Energy Method

The energy stored in body due to deformation is called strain energy. The strain energy stored in an elastic material upon deformation is calculated below for a number of different geometries and loading conditions.

The strain energy stored due to deformation in gear pair contact is calculated as follows:

Strain energy per unit volume.

$$Ue = \frac{1}{2} \times Stress \times Strain = \frac{1}{2} \times \sigma \times \varepsilon$$

$$= \frac{1}{2} \times [\sigma_{xx} \times \varepsilon_{xx} + \sigma_{yy} \times \varepsilon_{yy} + \sigma_{zz} \times \varepsilon_{zz} + \tau_{xy} \times \gamma_{xy} + \tau_{yz} \times \gamma_{yz} + \tau_{xz} \times \gamma_{xz}]$$

$$= 1/2 \times [0.46881 \times 1.82E\text{-}06 + 0.2662 \times 7.83E\text{-}07 + 0.19354 \times 4.92E\text{-}07$$

$$+ 1.59E\text{-}06 \times 1.22E\text{-}01 + 4.62E\text{-}02 \times 6.00E\text{-}07 + 9.28E\text{-}02 \times 1.21E\text{-}06]$$

$$= 7.45E\text{-}07 \text{ Joules}$$

Assumptions for Sample Calculations

Material under consideration is - Cast Steel 0.20 % Carbon, Untreated

Throughout the length of contact, average deformation is assumed to be constant

Assumptions for Energy Estimation

Energy Calculation by considering,

1. Sliding friction and rolling friction is absent No Power losses under friction.

2. Zero Bearing and Churning loss for lubricating oil.

3. The windage losses due to complex interactions with the air surrounding the gears is Zero.

5.2.2.2 Efficiency Calculation of Gear

In gear drive, the efficiency of system is given by

Efficiency (η) = (output power)/(input power)

= (Toutput×ωoutput)/(Tinput ×ωinput)..... (5) There is a power loss in the system due to the sliding action of toothing.

Besides this, there are power losses at the bearing, loss due to the churning of

Lubricating oil and other losses, so that the overall efficiency becomes still lesser

Than the efficiency calculated on the basis of loss due to sliding only.

The following values are typical for the spur gear drive system

η = 92 to 94 % for unmachined teeth

= 96 % for smooth and lubricated teeth

= 98 to 99 % for very carefully machined teeth with hydrodynamic

Lubrication between teeth surfaces With the ideal kind of design, construction and bearing systems coupled with very good Lubrication and gears of high quality, it is not difficult to attain an efficiency as high as 98% to 99%.for automotive drives in general, it is around 97%.

If the sliding velocity is high, it leads to considerable power loss due to friction and the Efficiency may fall to around 85%. According to Machinery's Handbook [8-9],

Hence efficiency is calculated as follows:

Input Energy = Torque acting on the pinion = 286.4788 Joules

Total loss = Input Energy –Output Energy

Efficiency Estimation is

Efficiency (η) = (Output Energy)/(Input Energy) ×100

= (Input Energy –Total Loss)/(Input energy) ×100

= 286.4788- 7.4544E-7/286.4788×100

= 99. 9997 %

Estimated loss due to strain energy method in transmission is 0.03 %.loss in transmission is minor in due to sliding of the contact of spur gear, hence from that loss we predict to efficiency by reducing the loss.

5.2.2 POSSIBILITIES TO CALCULATE THE AVERAGE DEFORMATION:

The deformation over the gear tooth Profile considering the 100 point on contact surface then each point contact is converted into line contact hence deformation over teeth in nearly equal as the calculated by direct contact in triaxial direction. For that divide gear tooth profile having total length 23.77mm and having dividend distance between points is 0.2377mm is shown in fig 5.1

The first model used is a 2D model. With this model we will find the real points on the tooth profile where the single pair in contact change into a double pair of teeth in contact this way we find with a good approximation the loading diagram for the 100 points on the involute active profile.
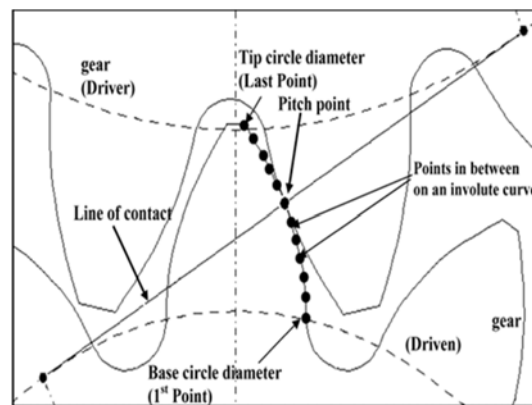

Fig 5-7 2D Meshing point contact of gear teeth model

To keep track of the surface elements the single point observation method was used. Thesingle point observation method was introduced by Andersson and Eriksson [17] and has been successfully used by Hugnell [18], Björklund [19] and others. Andersson and Eriksson studied how a point on a gear tooth moved in relation to

another point on the opposing tooth, see figure 5.2. By regarding single points on the teeth, interesting information can be deduced, such as the difference in sliding distance for a local point on the pinion- and gear surfaces, which is discussed below. Local wear and pressure on the flanks can be investigated.

a) Deformed Model in X- Direction
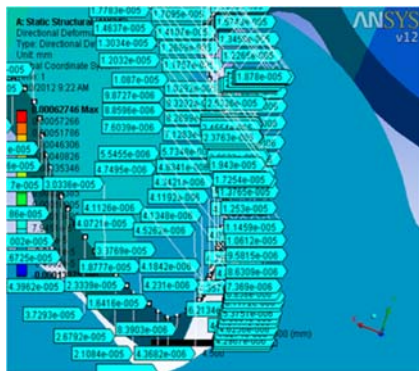
b) Deformed Model in Y- Direction
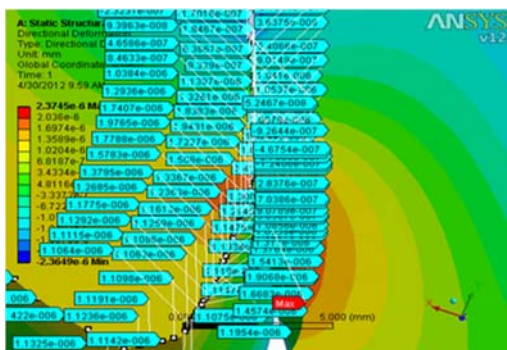
c) Deformed Model in Z- Direction

Figure 5.5 Contact Point along Tooth Face Width

The single point observation method naturally relates the wear, pressure and other relevant quantities, directly to the surface thus facilitating the study of local phenomenon on a tooth face. The surface geometry is also easily changed which is important when researching wear. The change of the surface curvature in contact affects the pressures and contact width as well as the load distribution during the meshing. This is important but often neglected and some works treat the wear as a curve fitting procedure [20], making the wear calculation very specific and not general

Table 5.3 shows the Average Deformation in X, Y & Z Direction over the Profile considering 100 point on face width compare with actual deformation calculated from the ansys.

From Table 5.4 which is shown in Appendix calculate value of normal stress, strain and shear stress and shear strain that value of all around 100 point on tooth edge face along tooth width used to calculated for strain energy.

Strain Energy per Unit Volume

$$U_e = \frac{1}{2} \times Stress \times Strain = \frac{1}{2} \times \sigma \times \varepsilon$$

$$= \frac{1}{2} \times [\sigma_{xx} \times \varepsilon_{xx} + \sigma_{yy} \times \varepsilon_{yy} + \sigma_{zz} \times \varepsilon_{zz} + \tau_{xy} \times \gamma_{xy} + \tau_{yz} \times \gamma_{yz} + \tau_{xz} \times \gamma_{xz}]$$

$$= 2.087e\text{-}3 \text{ joules}$$

Efficiency Estimation is.................

Efficiency (η) = (Output Energy)/ (Input Energy) ×100

= (Input Energy–Total Loss)/(Input energy)×100

= (286.4788- 2.087e-3)/286.4788 ×100

= 99. 9992 %

Estimated loss is

Using Strain Energy Method ignoring cantilever Effect Loss in transmission is estimated to be 0.08%.

Figure 5.5 shows variation of the deformation over point number along the tooth width contact region when complete load acted on contact region it will reached the pick level during contact by considering 50 points on the gear teeth profile.

Figure 5.5 shows variation of the deformation over point number along the tooth width
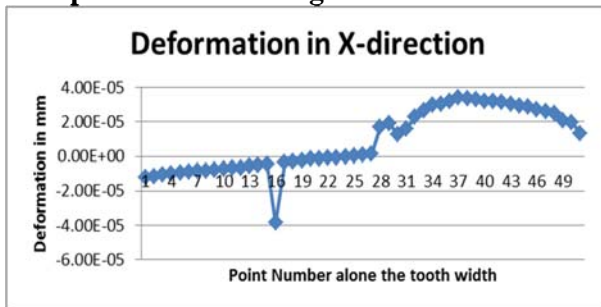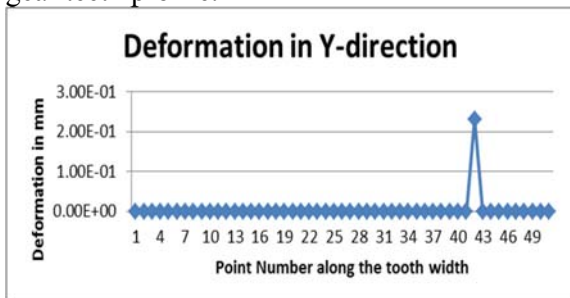

Figure 5.6 also shows variation of the deformation over point number along the tooth width contact region when complete load acted on contact region it will reached the pick level during contact by considering 50 points on the gear teeth profile.

Figure 5.6 shows variation of the deformation over point number along the tooth width

As seen in figure 5.7 shows the deformation is constantly increases in tooth entering the contact and decreases in tooth contact is leaving.
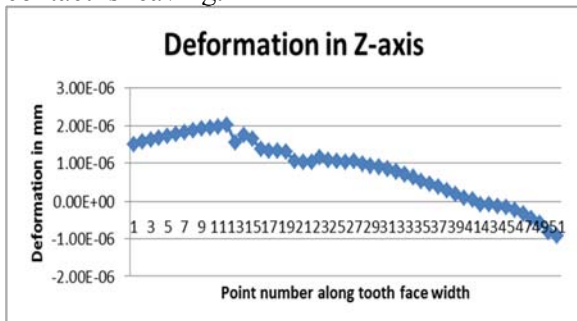

Figure 5.7 shows variation of the deformation over point number along tooth face width

5.4 Energy Required For the Cantilever Effect

Essentially, gear tooth in operation can be considered as a cantilever beam under load. For this reason, the ability of the tooth to resist tooth breakage of the tooth at the root is often reffed to as its beam strength .the basic tooth stresses caused by bending were first investigated by wildfred Lewis and the bending stress equation is still known as the Lewis formula although much modification of the original formula has taken place during the period of nearly 10 years.

Referring to fig5.5 and 5.6, the tooth load F is supposed to act at the tip corner as shown .under this assumption, calculation becomes simplified. Load F acts along the line of action at a grip angle of α1at the tip corner. When referred to pitch point P, angle α is the pressure angle of the system .As explained in earlier section, force F is normal to the tooth profile at pitch point the line of action

If at the intersection of the line of action and the present Centre line of the tooth this force F is resolved, then

We get,

The radial component Fr=F Sin α=1448.1920 N
The tangential component Ft=F Cos α =3978.8748 N
Normal Force Component F= 4234.2301 N

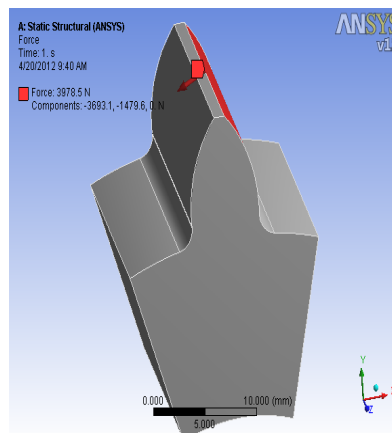**Fig 5.6: Loading and Boundary**

Bending Moment at the profile is maximum is given follows:

M = Ft*D = 3978.4788* 18 = 71612.6184N-mm

M1=Ft*D1= 3978.4788*16 = 63655.6608 N-mm

M2= Ft*D2=3978.4788*10 = 39784.7880N-mm

M3= Ft*D3= 3978.4788*8 =31827.8304 N-mm

Bending Moment is Maximum at the top tooth end hence force is applied at the end of gear tip corner. Single tooth is imported into ansysis software and the apply the Boundary

Condition the we got Model is discredited into small element by meshing then bottom end of the tooth is fixed and Tangential force is applied at end.
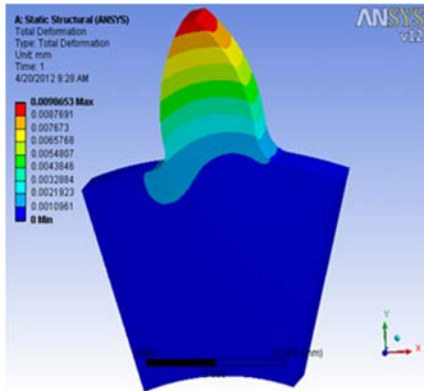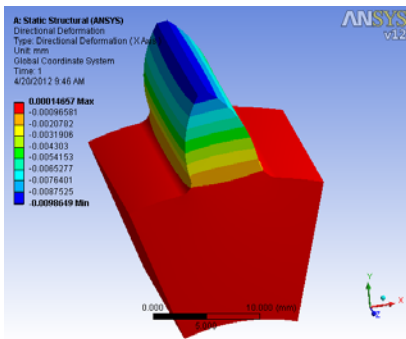

Fig: 5.1 Total Deformations
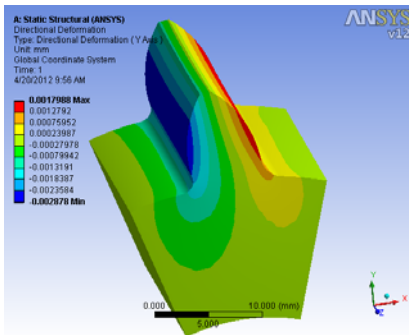

Fig: 5.2 Deformation in x –direction
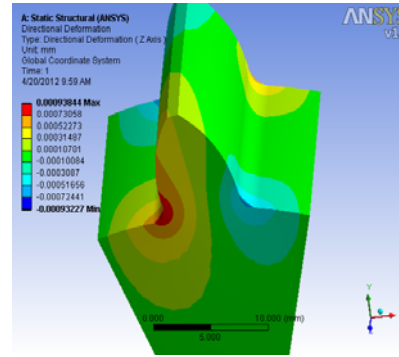

Fig: 5. 3 Deformation in y –direction

Fig: 5.4 Deformation in z –direction

Strain Energy per Unit Volume is......

$U_e = \frac{1}{2} \times$ Stress $\times$ Strain $= \frac{1}{2} \times \sigma \times \varepsilon$

$= \frac{1}{2} \times [\sigma_{xx} \times \varepsilon_{xx} + \sigma_{yy} \times \varepsilon_{yy} + \sigma_{zz} \times \varepsilon_{zz} + \tau_{xy} \times \gamma_{xy} + \tau_{yz} \times \gamma_{yz} + \tau_{xz} \times \gamma_{xz}]$

$= \frac{1}{2} \times [105.99 \times 0.0004445 + 147.15 \times 0.00066001 + 49.212 \times 0.000233$

$12.731 \times 0.00016 + 9.8418 \times 0.0001297 + 17.28 \times 0.00022465]$

$= 0.08147$ Joules

Efficiency Estimation is................

Efficiency $(\eta) = $ (Output Energy)/(Input Energy) $\times 100$

$= $ (Input Energy –Total Loss)/(Input energy) $\times 100$

$= (286.4788 - 0.08147)/286.4788 \times 100$

$= 99.9715\%$

Estimated loss is..........

Using Strain Energy Method Considering Cantilever Effect, Loss in transmission is estimated to be 2.85%

Energy required for the cantilever effect is large as compared to energy required for the ignoring the cantilever effect so that calculation made by ignoring cantilever effect is correct and also the loss in transmission more than ignoring the cantilever effect

CONCLUSION

A spur gear model was developed to predict the overall efficiency of a gearbox system.The analysis showed that the strain energy losses are highly dependent on the rotational speed of the gear wheels and is insignificant for speeds below 1000 rpm. The rolling friction loss was found to contribute a small percentage of the total power

loss in the system. The rolling friction losses depend on the position of contact of the mating gear teeth and reduce with increased load.

The efficiency of gearbox system is varies from 92%,95%,98%,99% by considering contact losses there is 1% loss in transmission contact deformation this big problem in transmission .

The additional energy is required to overcome the deformation of gear pair contact and Force required overcoming the friction.Loss of energy to predict efficiency

Softer gear the material larger is loss in transmission and harder the material Gear material lesser losses in transmission. The Young's modulus of elasticity is more percent loss in energy is less and Youngs modulus of elasticity is less percent loss in energy is more. The speed of rotation of gear is more therefore percent loss of energy is less and Speed is less percent in loss of energy is more.

FUTURE SCOPE

After the successful completion of the current project, the recommendations to improve

the current results and scope for the future work are as follows.

1. Use the full version software to mesh the contact model. Mesh near the contact region using very fine elements, to avoid the high magnitude errors at smaller stick zone sizes.
2. Mesh the contact model as symmetric near the contact region, and solve the model to observe the symmetry in stick zone results.
3. Resistant to motion is due to
4. a. Friction   b. Additional deformation
5. Strain energy loss affects efficiency of a gear unit means that it produces the highest possible output power for a given input power. The energy lost in the process manifests itself in the form of heat, for example in bearings, O-ring seals or gear wheels. As gear efficiency increases, its temperature will go down. This has a number of positive effects: a decreasing temperature not only extends the oil life, but the service life of seals as well. This in turn reduces the risk of

leakage. Another benefit is that fans or air conditioners in production facilities might be switched off, which is another contributor to lower energy costs and a better CO_2 balance.

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