Effect of Shaft Misalignment on The Stresses Distribution of Spur Gears

Dr. Hani Aziz Ameen*

Received on :17/8/2009
Accepted on:7/1/2010

Abstract

Shaft misalignment is considered as one of the common repeated problems in most rotating machineries , which leads to generate vibrations and extra dynamic loads on transmitting gears teeth, also leads to non- uniformity in distribution of applied load along the meshing tooth face by being concentrated on one side of tooth face. The present work concentrated on the analysis of stresses generated on transmitting gear tooth, also studied the effect of misalignment angle on stress distribution and its concentration. This is important for the gear design and those who works in gear maintenance , because fracture is expected to initiate and propagate at locations of stress concentration . ANSYS program using finite element technique had been used, as this program is efficient and accurate tool in stress analysis, especially for complicated shapes. Gear tooth model had been analyzed using finite element method in three dimensions. After calculating transmitted load and dynamic load, misalignment angle had been changed from (0°,0.2°,0.3°,0.4°,0.5°) then its effect on distribution of applied load had been calculated. The finite element program (ANSYS) had been executed for cases of misalignment angle (0°,0.2°,0.3°,0.4°,0.5°). The results showed clearly, that the stresses distribution and its concentration on tooth changed with misalignment angle and the equivalent stress is direct proportional with the misalignment angle. According to the values of generated stresses, the tooth fracture can be predicted.

Keywords: Machine Design , Gear Design , Finite Element Method, Stresses Analysis, ANSYS software, Misalignment

تأثير عدم المحاذاة في عمود الدوران على توزيع الاجهادات للتروس المستقيمة

الخلاصة

تعد مسألة عدم المحاذاة في عمود الدوران من المشاكل التشغيلية الشائعة والمتكررة في معظم المكائن، اذ تؤدي الى توليد اهتزازات واحمال ديناميكية اضافية على اسنان التروس الناقلة للحركة. والى عدم انتظام توزع الحمل المسلط على امتداد سطح السن المعشق بتركزه على جانب من سطح السن. يركز البحث الحالي حول تحليل الاجهادات المتولدة على سن ترس ناقل للحركة ، ودراسة تاثير التغير في زاوية عدم المحاذاة على توزع هذه الاجهادات وتركزها نظرا لما لدراسة تركز الاجهادات من اهمية بالنسبة لعاملين في مجالات تصميم التروس وتشغيلها وصيانتها. اذ يتوقع ان يبدأ الكسر ويتنامى عند مناطق تركز الاجهادات. تم في هذا البحث استخدام طريقة العناطر المحددة من خلال برنامج ANSYS بوصفها اداة فاعلة ودقيقة في تحليل الاجهادات ولاسيما للاشكال المعقدة. اجري تحليل نموذج سن الترس بتقنية العناصر المحددة بثلاثة ابعاد ، وبعد حساب الحمل المنقول والحمل الديناميكي الاضافي تم تغيير زاوية

* Pumps Engineering Department, Technical College, Al-Musaib /Babylon

https://doi.org/10.30684/etj.28.7.5
2412-0758/University of Technology-Iraq, Baghdad, Iraq

عدم المحاذاة (0° ، 0.2° ، 0.3° ، 0.4° ، 0.5°) ، وحساب تاثيرها على توزع الحمل المسلط على السن الناقل للحركة. تم تنفيذ برنامج الـــ ANSYS لعدة حالات من زوايا عدم المحاذاة (0° ، 0.2° ، 0.3° ، 0.4° ، 0.5°)، وبعد عرض النتائج المستحصلة ، يمكن ملاحظة التغير الذي يطرأ على توزع الاجهادات وتركزها بوضوح على سن الترس عند تغير زاوية عدم المحاذاة ، وان الاجهادات المكافئة تتغير طرديا مع زاوية عدم المحاذاة ، وبموجب قيم هذه الاجهادات المتولدة يتم تحديد امكانية حدوث كسر في السن .

Nomenclature

T… applied torque
P… transmitted power
w … angular velocity
r_b ... radius of base circle
r … radius of pitch circle
y … pressure angle
m… module
Z… number of teeth
F_i … load at contact point
d… pitch circle diameter (P.C.D.)
δ ... total deflection at contact point
δ_B … bending deflection
δ_S … shearing deflection
δ_G … deflection due to the inclination of foundation under load
δ_P … compressive deflection
ρ_2, ρ_1 radii of curvature for the
B… width of gear
S_m… the location of applied load
E… Young modulus
u … possion’s ration
K… tooth stiffness at contact point
F_{dyn}… dynamic load
F_t … tangential component of applied load
V… linear velocity
c… parameter depending on the material of gearing teeth and pressure angle
e… error in the shape of tooth
a… constant depending on pressure angle
E_p, E_g Young modulus for the pinion and gear
B' … active gear width of contact
C_z … spring constant
f_{RW} … directional error
α … angle of misalignment

Introduction

The reason of increasing the stresses concentration on gear teeth is the misalignment of the shafts that transmitting the movement, this misalignment made non uniform distribution of transmitted loads along gear face, which is caused generation of the stresses concentration in specific region. Several researches and studies have been done and used different ways for the generated stresses analysis on the movement transmitting gear teeth. Lewis Equation which is treated the spur gear as a cantilever beam and it is the main method for calculating the stresses in the root of the gear. Also several studies have been done using Photoelasticity Technique to determine the stresses concentration. The widely used of finite element method let the researchers to applied it on stress analysis of spur gear . Tae H. Chang and A. Kubo [1] studies the stresses distribution in spur gear , and it is depended on the finite element method to analyze the stresses and deformations in the gear and specify the expected failure locations. V. Simon [2] studies the loads and stresses distributions on the teeth for spur and helical gears. M.Q. Abdullah [3] studies the analysis of stresses and deformations on the gear teeth using finite element method, in this study a complete program is built to generate the symmetry and unsymmetry spur gear and analyze the stresses distribution on it. D.G. Lewicki and R. Ballarini [4] studies the stresses distributions on the spur gear using finite element methods.

In this paper the effect of misalignment of shaft on the stresses distribution and its concentration on the spur gear using finite element method during ANSYS program is investigated. Also the effect of shaft misalignment angle on the loads distribution and on the stresses distribution and its concentration on the movement transmitted gear teeth is studied.

Transmitted Load by Meshing of Teeth

The load distribution on the gear width [5] is determined as the following steps:

$$\text{Torque(T)} = \frac{\text{Power(P)}}{\omega(\text{rad/sec})} \quad \text{.... (1)}$$

The values of transmitted power and running speed are specified in order to calculate the torque, in this study the values of P=400 kW and N=5000 rpm

So, from Eq.(1) , T = 764.331 N.m

The vertical load (F_i) which is transmitted by teeth meshing, can be calculated by:

$$F_i = T / r_b \quad \text{....... (2)}$$

$$r_b = r.\cos(\psi) \quad \text{....... (3)}$$

It is required to specify the basic radius (r_b), Pressure angle(y), module(m) and number of teeth (Z) , in order to calculate F_i , so in this study the following data are used : Pressure angle (ψ) = 20° , Module (m) = 4 mm , No. of teeth (Z) = 50 teeth , (P.C.D) $d=m.Z$ = 200 mm , $r= d/2$ = 100 mm, hence, from Eq.(3) r_b = 93.969 mm , substituted into Eq.(2) , the value of normal load can be get

F_i=8133.843 N

Now to calculate the share of normal load at any contact point as follows:

Total Deflection (δ) at Contact Point

The total deflection at contact point can be calculated as follows[6]:

$$\delta = \delta_B + \delta_S + \delta_G + \delta_P \quad \text{...... (4)}$$

where,

$$\delta_B = \frac{12\ F_i \cos^2(\psi)\ z}{E\,B\,S_F^3}[S^2{}_m + \frac{z^2}{3} - S_m z] + \frac{6\ F_i \cos^2(\psi)\ (w-z)^3}{E\,B\,S_F^3} \left[\frac{(w-S_m)}{(w-z)}\left\{4 - \frac{(w-S_m)}{(w-z)}\right\} - 2\log_e\left[\frac{(w-S_m)}{(w-z)}\right] - 3 \right] \quad \text{.....(5)}$$

$$\delta_S = \frac{2(1+\upsilon)\ F_i \cos^2(\psi)}{E\,B\,S_F}\left[z + (w-z)\log_e\left(\frac{w-z}{w-S_m}\right)\right] \quad \text{...(6)}$$

$$w = \frac{nS_F - zS_K}{S_F - S_K} \quad \text{.... (7)}$$

$$\delta_G = \frac{24F_i\ S^2{}_m\ \cos^2(\psi)}{\pi\ EBS_F} \quad \text{.... (8)}$$

$$\delta_P = \frac{4\ F_i\ (1+\upsilon^2)}{\pi EB}\left(\frac{\rho_2}{\rho_1+\rho_2}\right) \text{... (9)}$$

where :

δ_B = Bending deflection

δ_S = Shearing deflection

δ_G = Deflection due to the inclination of foundation under load

δ_P = Compressive deflection

ρ_2, ρ_1 = Radii of curvature for the two teeth gearing at contact point

Fig.(1) demonstrated the terms of teeth which are used in calculated the deflection . The following data are used in this study:

S_m = 6.338 mm , S_F = 8.891 mm , S_K = 4.93 mm, n = 9 mm , z = 1 mm

w = 18.957 mm , B = 40 mm , $y = 20°$ .Also , the Young modulus (E) and Possion's ratio (u) of the material of tooth is :

E = 207000 N/mm^2 , $u = 0.3$

Tooth Stiffness at Contact Point

To Calculated the tooth stiffness at contact point as follows[6]:

$$K = \frac{F_i}{\delta} \quad \text{..... (10)}$$

If the applied load equal to the normal load , hence the load at contact point (c) equal to : $F_c = F_i$ =8133.843 N so, it can be explained the stiffness of gear at point (c) using the material of tooth and its size as follows:

$$\frac{\delta_B}{F_i} = 181.635\text{x}10^{-9} \quad \text{mm/N} \ ,$$

$$\frac{\delta_S}{F_i} = 228.745\text{x}10^{-9} \quad \text{mm/N}$$

$$\frac{\delta_G}{F_i} = 3682.788\text{x}10^{-9} \quad \text{mm/N} \ ,$$

$$\frac{\delta_P}{F_i} = 83.848\text{x}10^{-9} \quad \text{mm/N}$$

$$\frac{\delta_{tot}}{F_i} = \frac{\delta_B}{F_i} + \frac{\delta_S}{F_i} + \frac{\delta_G}{F_i} + \frac{\delta_P}{F_i}$$

$$= 4177.018\text{x}10^{-9} \text{ mm/N}$$

$$K = \frac{1}{\delta_{tot}/F_i} = 239405.2 \quad \text{N/mm}$$

$$= K_c$$

where

K_c = stiffness at point (c) .

Dynamic Load

In order to demonstrate the effect of the dynamic load , Buckingham equation [7] is used, according to this equation, the dynamic load is depending on the linear velocity (V), tangent load (F_t), tooth width (B) and the material properties of the gear .It can be shown the Buckingham equation as follows[8]:

$$F_{dyn} = F_t + \frac{21V(Bce+F_t)}{21V+\sqrt{Bce+F_t}} \text{...... (11)}$$

$$F_t = F_i \cos(\Psi) \quad \text{....... (12)}$$

where :

e: error in the shape of tooth

It can be found from tolerance table in Refs[7,8] corresponding to m= 4 mm that $e = 2\times10^{-6}$m c: Parameter depending on the material of gearing teeth and the pressure angle [8],

$$c = \frac{a\,(E_p\,E_g)}{(E_p + E_g)} \quad \ldots\ldots (13)$$

a: constant depending on pressure angle , when $\psi = 20°$, $a\,(\Psi = 20°) = 0.111$

E_p, E_g: Young modulus for the pinion and gear , $E_p = E_g = 207 \times 10^9\ N/m^2$

Substituting the values of a, E_p, E_g in Eq.(13), the value of c can be obtained : $c = 11488.5\ N/mm^2$

Hence , the dynamic load (Eq.(11) will be : $F_{dyn} = 16204.94\ N$

i.e that the applied load on the analyzed gear increased by twice than the transmitted load after the dynamic effect is calculated.

Active Gear Width of Contact

When the teeth is meshing in correct way the load is distributed uniformly along the gear width as shown in Fig.(2), and when the misalignment is happen between shafts, so the meshing of teeth will be not correct and the contact will be subside in one side from the gear face without the other side as demonstrated in Fig.(3).

The alteration of the applied distribution load that concentrated on one side and decreasing gradually to the other side and it can be approximated the applied distribution load as a cubic parabolic distribution. Fig.(4) explained this distribution on the active gear width of contact (B') and can be described in the following equation[9] :

$$F(x) = F_{max}\,.[1 - (x/B')^3] \quad \ldots\ldots (14)$$

It can be defined the tangential load on the gear width as follows [9] :

$$F = F_m\,.B = \int_0^{B'} F(x).dx = \frac{3}{4} B' F_{max} \ldots (15)$$

Hence,

$$F_m = \frac{F}{B} \quad \ldots\ldots (16)$$

$$F_{max} = \frac{4\,F}{3\,B'} \quad \ldots\ldots (17)$$

It is found experimentally that (F_{max}) can be calculated as [9]:

$$F_{max} = C_z.f_{RW}\,\frac{B'}{B} \quad \ldots\ldots (18)$$

where :

C_z : Spring constant $(N/\mu mm)$ and can be defined as [9]:

$$C_z = \frac{K}{B} \times 10^{-3} \quad \ldots\ldots (19)$$

K : gear stiffness at the location of applied load $(N/mm.)$

f_{RW} : Directional error (μ) and can be defined as [9]:

$$f_{RW} = 10^3.\alpha.B \quad \ldots\ldots (20)$$

α : angle of misalignment (Radian)

From Eq.(17) and Eq.(18) , it can be deduced that :

$$B' = \sqrt{\frac{4\,F\,B}{3\,f_{RW}\,.C_z}} \quad \ldots\ldots (21)$$

This equation represent the active gear width of contact, when the misalignment of gears is occurred .

Load Distribution on Gear in Case of Shaft Misalignment [10-14]

The load distribution on the gear in case of misalignment can be specified by putting a thin layer of soft material, which is explained the pressure zone on gear face during the meshing of teeth, and when the load is applied on the teeth , it can be shown that the stamp on the thin layer as illustrated in Fig.(5).

In this research the load distribution is calculated on the gear for several cases of misalignment angle $\alpha = 0.2°$, $0.3°$, $0.4°$, $0.5°$,where (B') is determined for each α (Eq.(21)) and then find the centered forces , which is represented the actual distributed load.

In order to facilitate representation of the load distribution on the gear width, the equivalent centered forces is calculated at specific points on contact gear width (B'), the width is divided into several intervals , where in each interval center the centered force is calculated.

The centered force can be found by the following equation:

$$F = \int_{x_1}^{x_2} F_{max} \left[1 - \left(\frac{x}{B'} \right)^3 \right] dx \qquad \dots (22)$$

Fig.(6) demonstrates the location of the coordinate (x) on the gearing teeth.

After calculation the centered forces in which it sum equal to the tangential distributed load and to find the normal component of tangential centered forces [10] using the following equation.

$$F_n = F_t \tan(\Psi) \qquad \dots\dots(23)$$

Thus, to find the distribution forces on the gear width during misalignment , equations(22) and (23) are used with Simpson Rule to achieve the integration in Eq.(22) and the following forces can be get as in Tables(1),(2),(3) & (4), and is illustrated in Fig.(7).

Model Generation by ANSYS

The ultimate purpose of a finite element analysis is to re-create mathematically the behavior of an actual engineering system[15]. In other words, the analysis must be an accurate mathematical model of a physical prototype[16].In the broadest sense, the model comprises all the nodes, elements, material properties, real constants, boundary conditions and the other features that used to represent the physical system.

In ANSYS terminology, the term model generation usually takes on the narrower meaning of generating the nodes and elements that represent the special volume and connectivity of the actual system. Thus, model generation in this study will mean the process of defining the geometric configuration of the model's nodes and elements. The program offers the following approaches to model generation[17-18]:

a) Creating a solid model
b) Using direct generation
c) Importing a model created in a computer-aided design CAD system.

The method used in this research is a creating solid model. In solid
modeling one can describe the boundaries of the model, establish controls over the size and desired shape elements automatically, i.e. drawing the two dimensional gear model and meshing using meshtool and dragging it in z-direction to form three dimensional gear model. Solid modeling is usually more powerful and versatile than other modeling, and is commonly the preferred method for generation models. The

three Dimensional model of gear is done by drawing and meshing two dimension gear plane with element plan42 and then dragging by solid45 to form three dimensional gear. Fig.(8) shows the gear model in ANSYS. Fig.(9) shows the load distribution when the misalignment angle (α) is zero, i.e. there is alignment of shafts, and it can be seen that the distribution of load is uniform. The load distribution is applied in ANSYS according to equations (22) and (23) and its value depend on the misalignment angle (α), these equations are programmed in ANSYS software by Ansys Parametric Design Language (APDL) as follows :

```
/prep7
! Data for Force F
P=400*1000  ! power = 400 kW
N=5000        ! rpm
omg=5000*(2*3.14/60)
T=P/omg
*afun,deg : Psi=20
m=4  ! mm : Z=50
dpcd=m*Z  ! mm  : r=dpcd/2    ! mm
rb=r*cos(psi)  ! mm : Fi=T/(rb/1000)
!Date for stiffness K
Sm=6.338 : Sf=8.891
Sk=4.93 : nn=9
zz=1 : w=18.957
B=40 : E=207000 :posi=0.3
! DeltaB
A1=12*Fi*zz*(cos(psi))**2
A2=E*B*Sf**3
A3=Sm**2+((zz**2)/3)-(Sm*zz)
A4=6*Fi*(cos(psi))**2*(w-zz)**3
A5=(w-Sm)/(w-zz)
DeltB=((A1/A2)*A3)+(A4/A2)*(A5*
(4-A5)-2*log(A5)-3)
DB=DeltB/Fi
! DeltaS
B1=2*(1+posi)*Fi*(cos(psi))**2
B2=E*B*Sf
B3=(w-zz)/(w-Sm)
DeltS=(B1/B2)*(zz+(w-zz)*log(B3))
DS=DeltS/Fi
! DeltaG
C1=24*Fi*Sm**2*(cos(psi))**2
C2=3.14*E*B*Sf
DeltG=C1/C2
DG= DeltG/Fi
! DeltaP
D1=4*Fi*(1+posi**2)
D2=3.14*E*B
D3= rb/(rb+rb)
DeltP=(D1/D2)*D3
DP= DeltP/Fi
Dtot=DB+DS+DG+DP
Kc=1/Dtot
! Data for dynamic force Fdyn
Ft=Fi*cos(psi)
ee=2e-3 : aa=0.111: Ep=E:Eg=E
cc= (aa*Ep*Eg)/(Ep+Eg)
V= omg*rb
E1=21*V*((B*cc*ee)+Ft)
E2=21*V+sqrt((B*cc*ee)+Ft)
Fdyn=Ft+(E1/E2)
! Bdash
Alf=0.00349: fRW=1000*Alf*B
Cz= (Kc/B)*1e-3: F1=4*Fdyn*B
F2=3*fRW*Cz : Bdash=sqrt(F1/F2)
Fmax=(4*Fdyn)/(3*Bdash)
!   the   function   Fmax*(1-
(x/Bdash)**3)
X1=0: X2=1: N=1  ! no. of interval
*Dim,Xs,,10 : *Dim,Ys,,10
K1=0:  K1=K1+1
N1=N+K1-1 : L=(X2-X1)/N1
H=L:  A1=0
! LOOP
*DO,I,1,N1 :  *DO,J,1,2
Xs(J)=X1+(I-1)*L+(J-1)*H
Xso=Xs(J) :
! FUNCTION
Yso=Fmax*(1-(Xso/Bdash)**3)
Ys(J)=Yso
*ENDDO
```

```
A1=A1+H/2*(Ys(1)+Ys(2))
*ENDDO
Ftag=A1
Fnor=Ftag*tan(psi)
Finish
```

Tables(1),(2),(3)&(4) are contained the values of loads with (α = 0.2° , 0.3° , 0.4° & 0.5°). Fig.(10) shows the load distribution according to the angles of misalignment. This figure shows that the loads become less strong gradually in only one side of the gear face, while it is concentrated on the other side.

Results and Discussions

Misalignment of shaft causes changes in the teeth gearing .This changes the load distribution on the teeth. Manufacturing errors causes alternating loads on the gear. Local deformation at the point of applying the load and misalignment of the shaft causes stress concentration (i.e. the stresses are concentrated on one side of the teeth while gradually decreasing on the other side) . Consequently, sudden fracture in the tooth occurs. Generally, there appears at the side of stress concentration, wearing or pitting and consequent damage. The contact region between the teeth of a sound gear may be represented by straight line parallel to the gear's axes of rotation, i.e. load distribution along the gear width is uniform (Fig.(9)). Misalignment is due to manufacturing errors , bending in the involutes curve of the gear …etc, with consequent changes in the teeth gearing making concentrated load on one side of the teeth (Fig.(10)) .

Fig.(11) explain the stress distribution on the gear width when α = 0° i.e. shaft alignment, and it can be seen that the stress distribution is uniform along the gear face. Fig.(12) explain the stress distribution on the gear width when α = 0.2°, Fig.(12-a) shows the distribution of stress is not uniform along the gear face, and Fig(12-b) shows the deformation occurs in spur gear during misalignment. Fig.(13) demonstrates that when α = 0.3°, stress distribution on the gear width occurs; the non-uniformity of stress distribution along the gear face is demonstrated by Fig.(13-a), while occurrence of deformation in spur gear during misalignment is demonstrated by Fig(13-b).

Fig.(14) demonstrates that when α = 0.4°, stress distribution on the gear width occurs; the non-uniformity of stress distribution along the gear face is demonstrated by Fig.(14-a), while occurrence of deformation in spur gear during misalignment is demonstrated by Fig(14-b).

Fig.(15) demonstrates that when α = 0.5°, stress distribution on the gear width occurs; the non-uniformity of stress distribution along the gear face is demonstrated by Fig.(15-a), while occurrence of deformation in spur gear during misalignment is demonstrated by Fig(15-b)

Fig.(16) shows the relationship between the misalignment angle and the equivalent stress at contact point on spur gear, it can be shown that when the misalignment angle increases the equivalent stresses increase directly. Fig.(17) shows the relationship between the misalignment angle and max. deflection at contact point on spur gear , it can be seen that the deflection is directly proportional to the misalignment angle.

Fig.(18) shows the comparison between the present results and the result from Ref. [3] for the tooth thickness and shear stress at zero angle of misalignment i.e. alignment shafts. It can be shown that there are differences in the results, that due to the mesh used in Ref.[3] is coarse mesh and due to that their results were far from the ANSYS results.

Conclusions

From this study the following conclusion can be get :

1-The values of equivalent stresses and its distribution is changed with changing the misalignment angle, where the stresses concentration is increased in the contact region and in the root of the tooth with increasing the misalignment angle, this is occurred in the side of subside the load and decreasing in the other side of the gear face.

2- increasing the deformation of gear with increasing misalignment angle.

3- increasing the probability of fracturing the gear in the root with increasing the misalignment angle.

References

[1] Chang T.H. and Kubo A., " Simple Stress Formulae for a Thin Rimmed Spur Gear" , Journal of Mechanisms, Transmissions, and Automation in Design, Vol.107,No.3, pp. 406-411, 1985.

[2] Simon V., " Load and Stress Distributions in Spur and Helical Gears", Journal of Mechanisms, Transmissions, and Automation in Design", Vol.110,No. 2, pp. 197-202, 1988.

[3] Abdullah M.Q, " Analysis of Gear Tooth Stresses using Finite Element Technique", M.Sc. thesis, University of Al-Nahreen,1994.

[4] Lewicki D.G., and Ballarini R., " Effect of Rim Thickness on Gear Crack Propagation Path", 7th International Power Transmission and Gearing Conference, U.S.A., California, 1996.

[5] Maitra G.M., " Hand Book of Gear Design", Tata McGraw-Hill Company, India, New Delhi , 1997.

[6] Elkholy A.H., " Tooth Load Sharing in High Contact Ratio Spur Gears", Journal of Mechanisms, Transmissions, and Automation in Design , Vol. 107, No. 1, pp.11-16, 1985.

[7] Buckingham E., " Analytical Mechanics of Gears", McGraw-Hill, U.S.A., New York, 1949.

[8] Pandya N.C., and Shah C.S., " Elements of Machine Design", Charotar Publishing House, India,1986.

[9] Niemann G., " Machine Elements", Allied Publishers Private, Ltd., India, New Delhi, 1980 .

[10] Al-Musawi A.K., " Vibration Analysis and the Allowable Limits in Rotating Machinery", M.Sc. thesis , University of Baghdad , 1993.

[11] " Shaft Aligment is a Profitable Form of Preventative Maintenance", SKF Bearing Maintenance Handbook, Hane Training, Inc., 2001.

[12] Wowk V., " Specifying Shaft Alignment", McGraw-Hill Company, U.S.A., New York, 2000.

[13] Shigley J.E., and Mischke C.R., " Mechanical Engineering Design", McGraw-Hill Company , U.S.A. , New York, 1989.

[14] Polak S., " Gear box & Gear System Problems", Gears Conference, U.K., London, 1999.

[15] Saeed Mouveni "Finite Element analysis" theory and application With ANSYA, 1999.

[16] Tim Langlais "ANSYS Short course ", 1999, from internet.

[17] User's manual of FEA/ANSYS/Version 5.4/1997.

[18] Basic structural, Training Manual, (1996, ANSYS Inc., (ANSYS on-line-help).

Table (1) when $\alpha = 0.2$, $B' = 32$ **mm**

Point	1	2	3	4	5	6	7	8	9	10	11	12	13
Ft	671	671	671	670	669	668	666	663	659	654	648	640	632
Fn	244	244	244	244	243	243	242	241	239	238	235	233	230

Point	14	15	16	17	18	19	20	21	22	23	24	25	26
Ft	621	610	596	580	563	543	521	497	470	441	409	374	336
Fn	226	222	217	211	205	197	189	181	171	160	149	136	122

Point	27	28	29	30	31	32
Ft	295	251	203	152	40	21
Fn	107	91	74	55	14	7

Table(2) when $\alpha = 0.3$, $B' = 26$ **mm**

Point	1	2	3	4	5	6	7	8	9	10	11	12	13
Ft	822	822	821	820	818	814	809	803	794	783	769	753	733
Fn	299	299	299	298	297	296	294	292	289	285	280	274	266

Point	14	15	16	17	18	19	20	21	22	23	24	25	26
Ft	710	683	652	618	578	534	485	431	370	305	233	154	69
Fn	258	248	237	224	210	194	176	156	135	111	84	56	25

Table (3) when $\alpha = 0.4$, $B' = 22$ **mm**

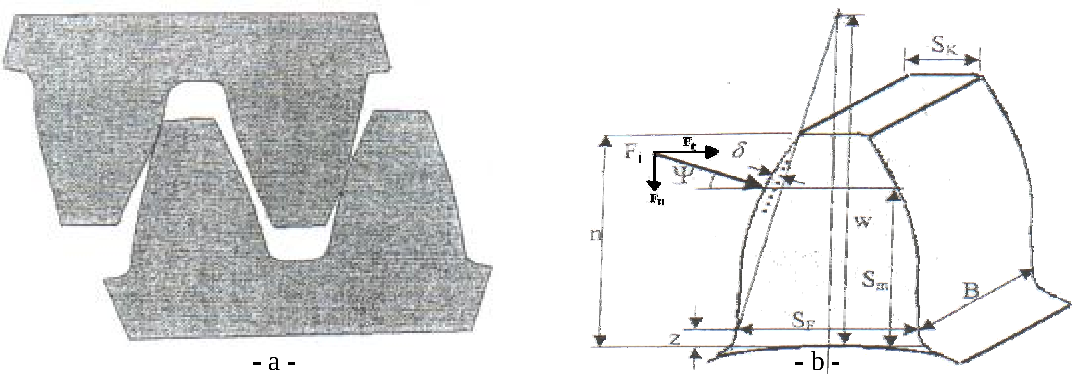
Point	1	2	3	4	5	6	7	8	9	10	11	12	13
Ft	949	949	947	945	941	935	926	914	899	879	855	826	791
Fn	345	345	345	344	342	340	337	333	327	320	311	300	288

Point	14	15	16	17	18	19	20	21	22
Ft	750	702	648	586	516	438	351	254	147
Fn	273	255	236	213	188	159	127	92	53

Table(4) when $\alpha = 0.5$, $B' = 20$ mm

Point	1	2	3	4	5	6	7	8	9	10	11
Ft	1061	1061	1059	1056	1049	1040	1026	1008	983	952	914
Fn	386	386	385	384	382	378	373	366	358	346	333

Point	12	13	14	15	16	17	18	19	20
Ft	869	814	750	676	590	494	384	261	125
Fn	316	296	273	246	215	179	139	95	45



- a - - b -

Figure (1) Tooth terms [6]

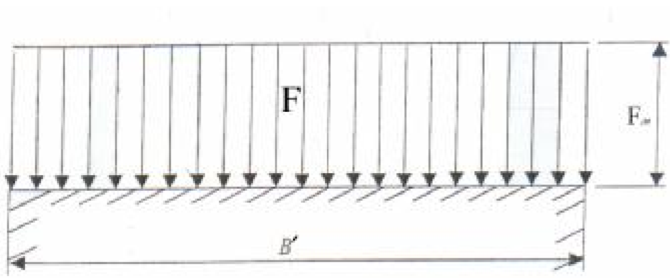


Figure (2) Uniform distributed of load in along the gear width

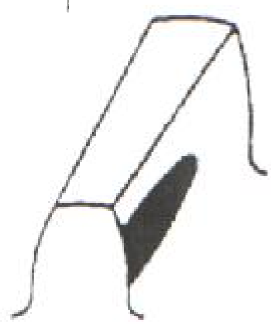


Figure (3) Subside of the contact one side of the gear

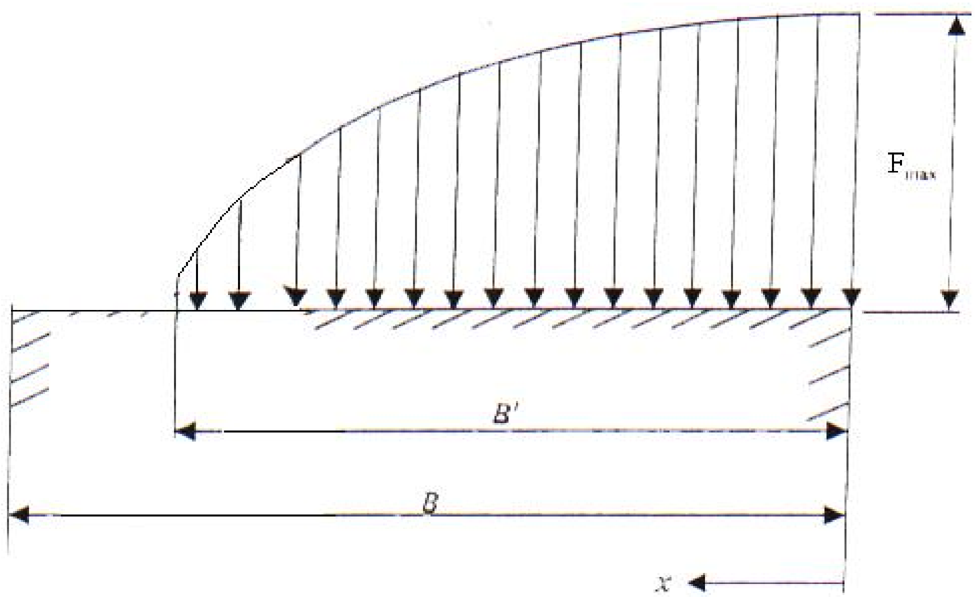


Figure (4) Load distribution along the active gear width in contact when misalignment is occur

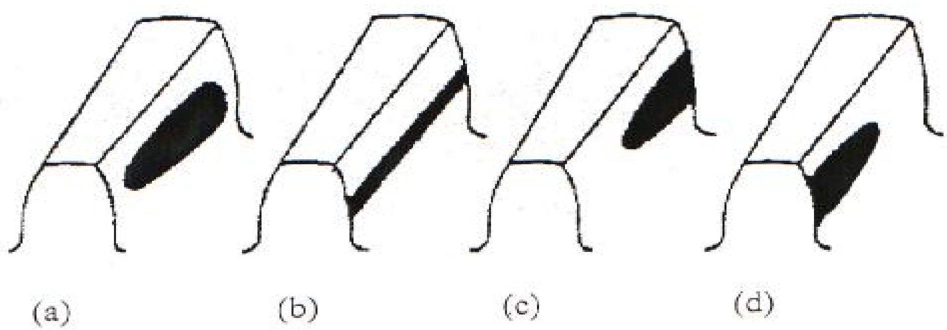


Figure (5) Configurations of load the regions on the tooth face during misalignment

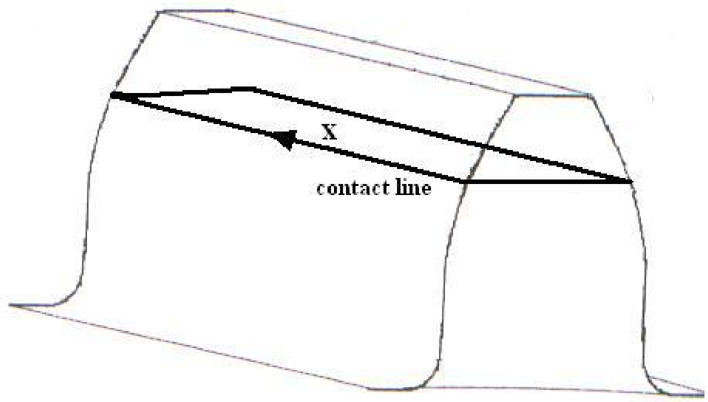


Figure (6) Configurations of the contact line between the gearing tooth

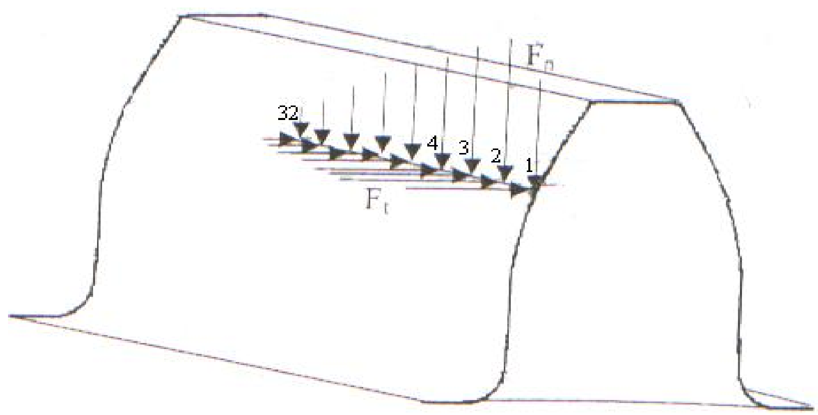


Figure (7) The components of equivalent forces for misalignment gear

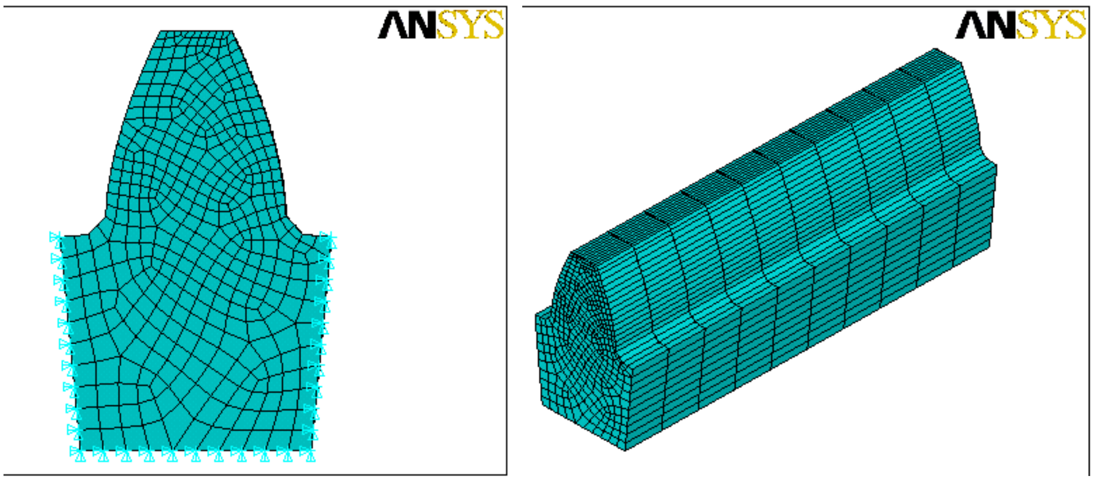


Figure (8) Gear Model in ANSYS

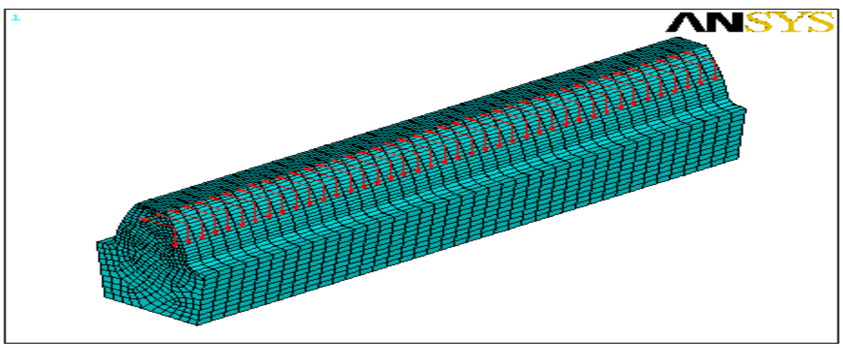
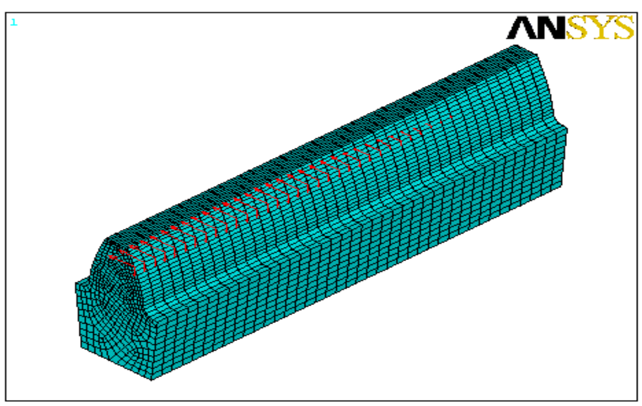
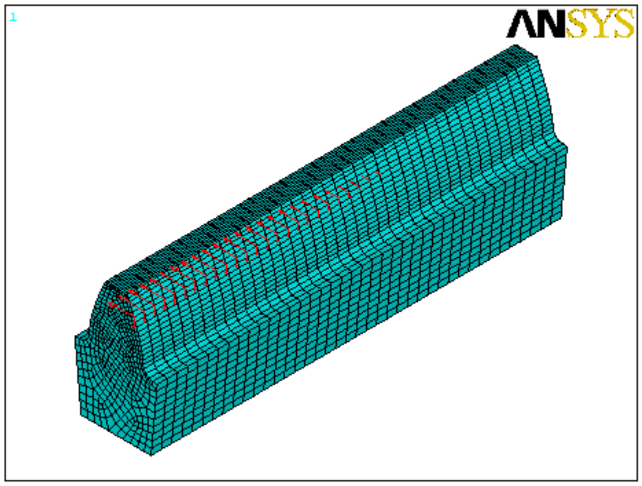


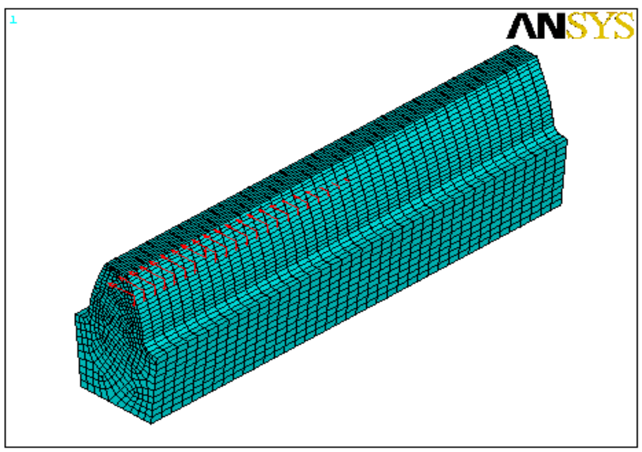
Figure (9) The load distributions at misalignment angle equal zero



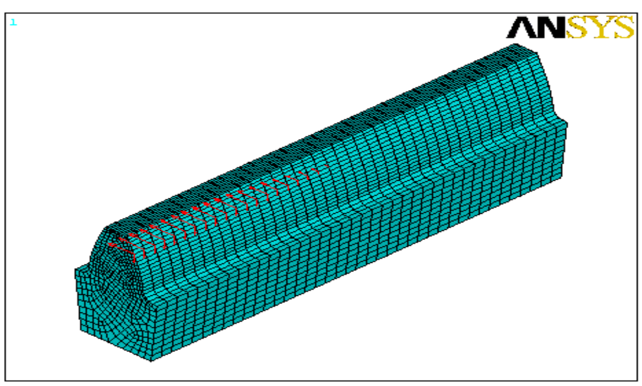
$\alpha = 0.2°$



$\alpha = 0.3°$

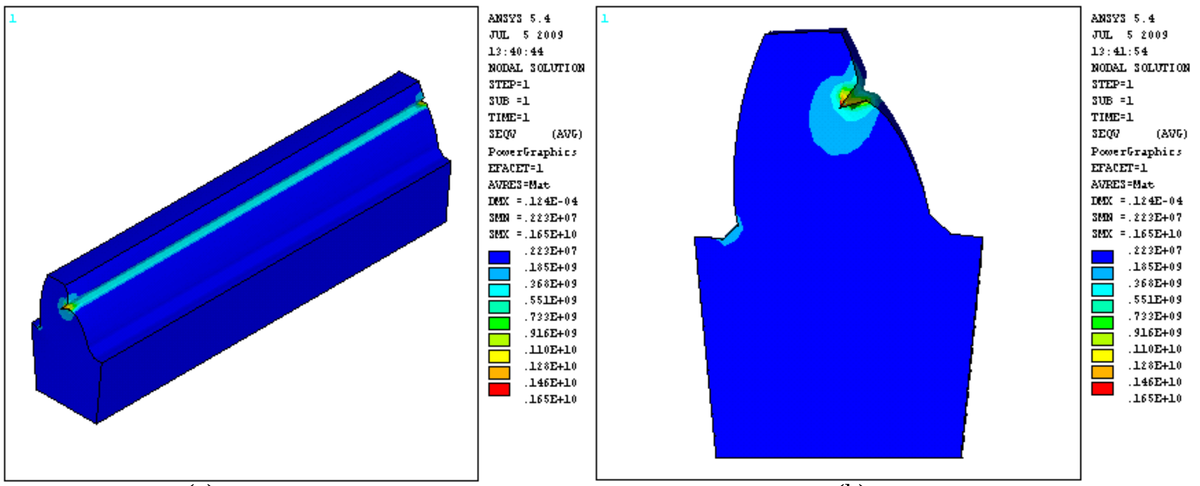


$\alpha = 0.4°$



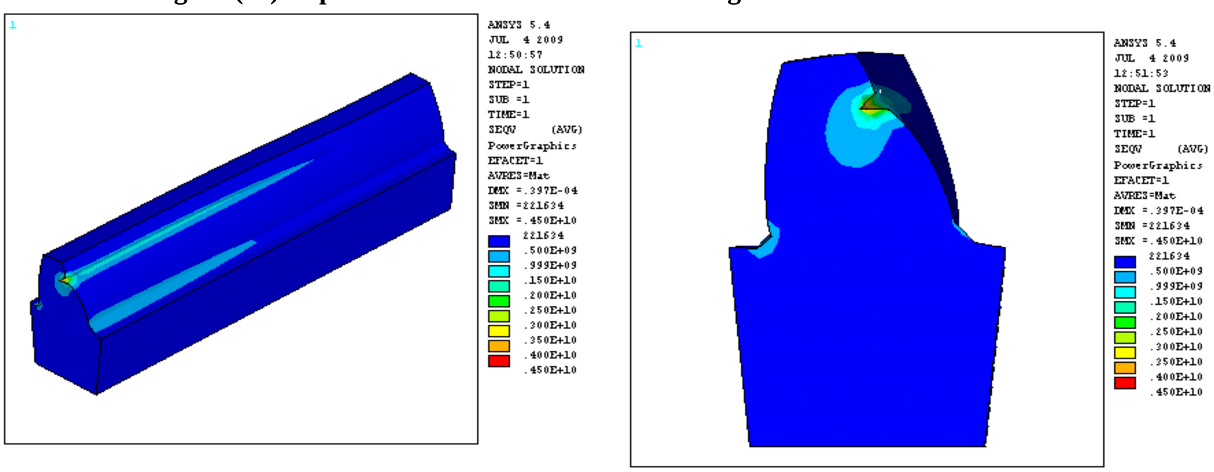
$\alpha = 0.5°$

Figure (10) The load distributions at different misalignment angles



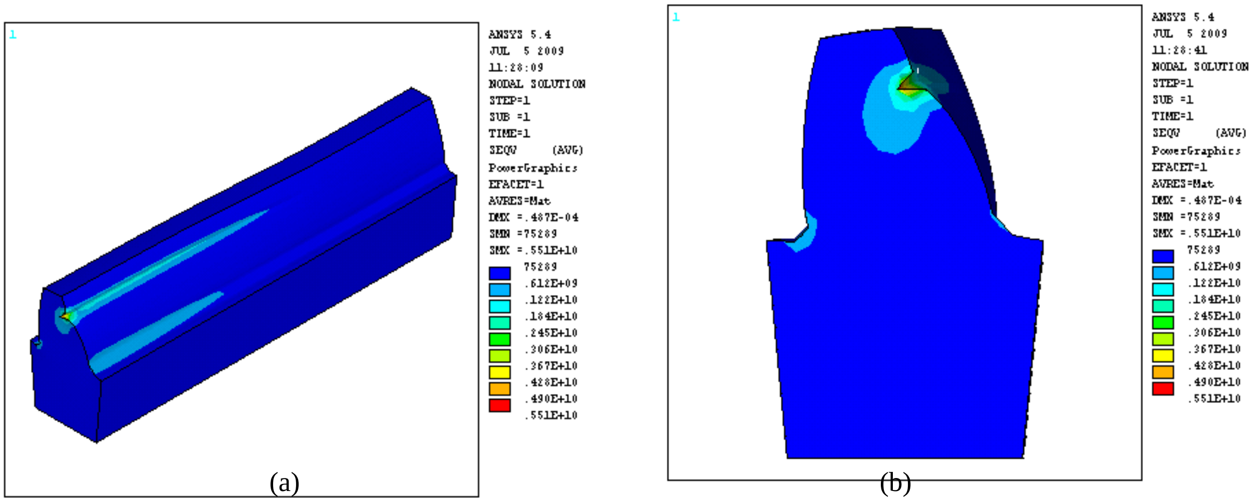
(a) (b)

Figure (11) Explain the stress distribution on the gear width when α = 0°



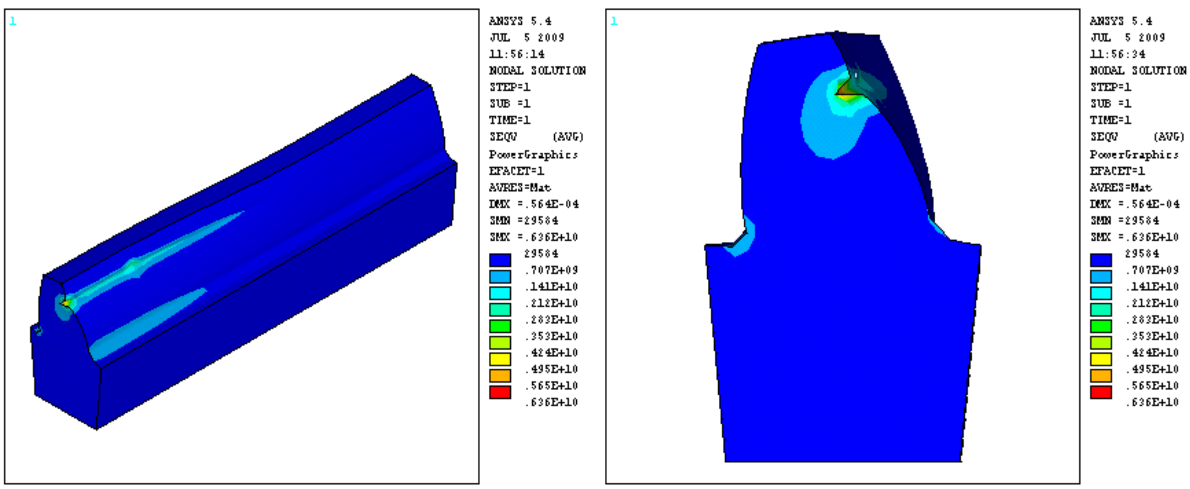
(a) (b)

Figure (12) Explain the stress distribution on the gear width when α = 0.2°



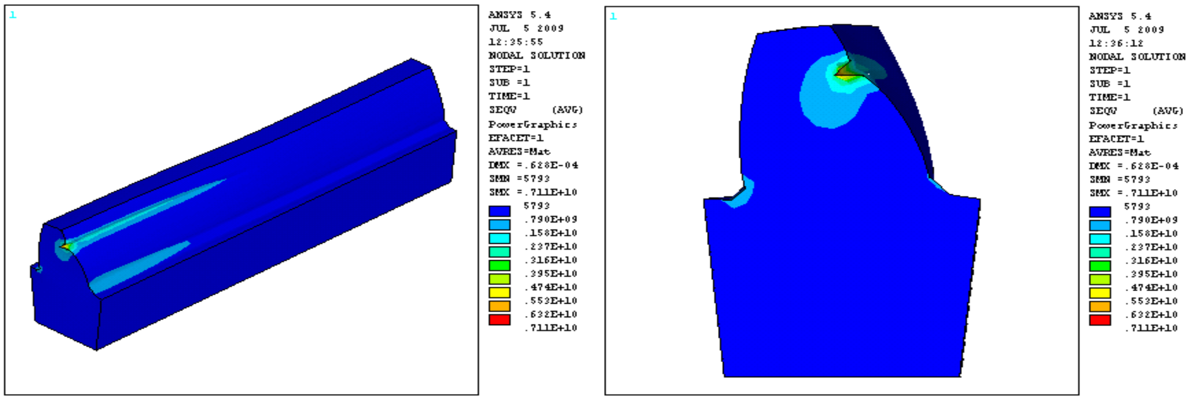
(a) (b)

Figure (13) Explain the stress distribution on the gear width when α = 0.3°



(a) (b)

Figure (14) Explain the stress distribution on the gear width when α = 0.4°



(a) (b)

Figure (15) Explain the stress distribution on the gear width when α = 0.5°

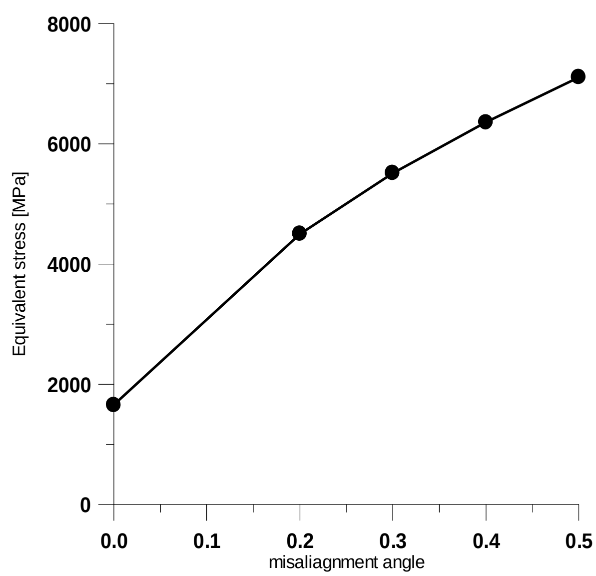


Figure (16) The relationship between the misalignment angle and the equivalent stress on spur gear

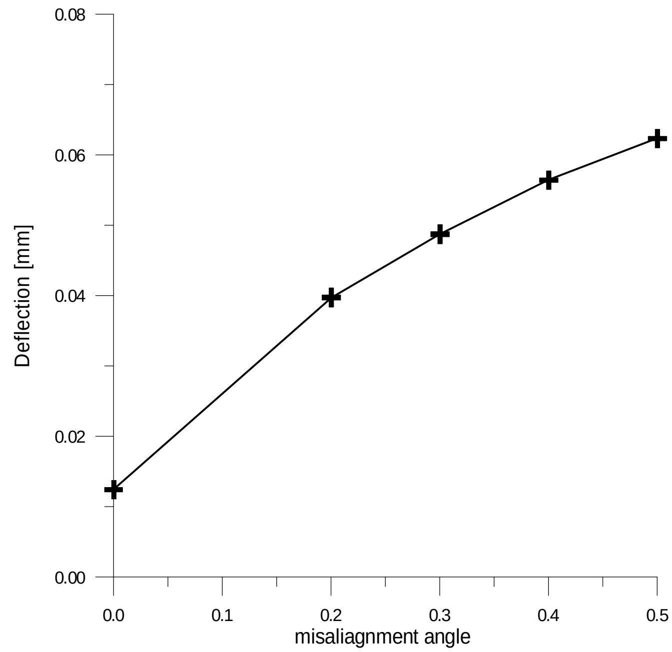


Figure (17) The relationship between the misalignment angle and the deflection on spur gear

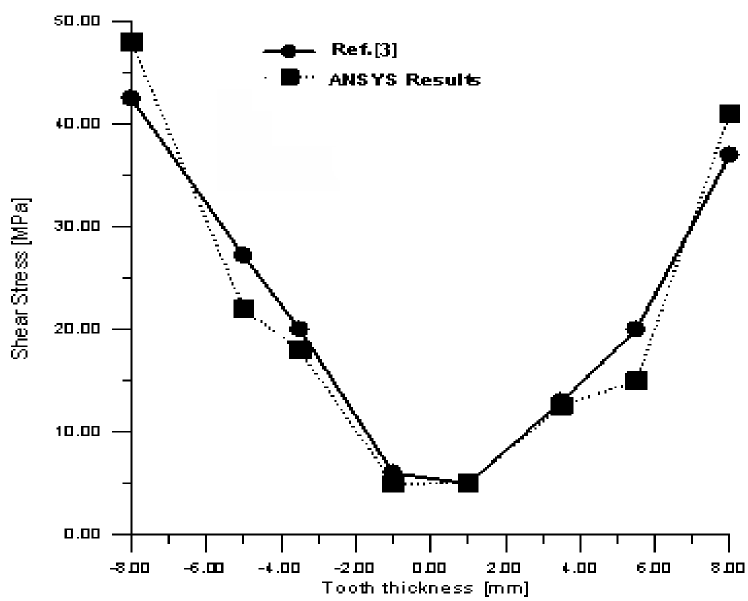


Figure (18) The relationship between the tooth thickness and shear stress when misalignment angle = 0