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Stress Analysis on Spur Gears Using ANSYS Workbench 16.0

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Abstract: Gear is one of the most important machine elements in mechanical power transmission system. It is a rotating machine part having cut tooth which meshes with another tooth part in order to transmit torque. The application of gears has a wide range starting from tiny wrist watch to huge heavy machinery such as automobile, aerospace industry and marine engines due to its high degree of reliability and compactness. Spur gears have straight tooth and are parallel to the axis of the wheel. The bending stress and total deformation of gear tooth is considered to be the paramount objective of for modern gear design. A pair of spur gear tooth in action is generally subjected to two types of cyclic stress, contact stress and bending stress including fatigue. In this paper, the contact stress analysis of stainless steel spur gear by theoretical method using Hertz equation and Finite element analysis using ANSYS 16.0 workbench. The present work is an attempt to find root bending stress distribution, maximum allowable contact stress and total deformation of a stainless steel spur gear tooth.

Keywords: Stainless Steel Spur Gear, Static load, Contact Stress, Bending Stress, Total Deformation, ANSYS Workbench 16.0

1. INTRODUCTION

Gears are used in most types of machinery. Like nuts and bolts, gears are common machine elements that will be needed from time to time by almost all machine designers. They are mostly used to transmit torque and angular velocity. It would be appropriate to say that because of compactness and high degree of reliability, gears will predominate in future industrial machines as the most effective means of power transmission. Furthermore, refinement in the application of gear technology is necessary due to the sudden shift from heavy industry such as shipbuilding to automobile manufacturing and office automation tools.

As there is always a demand for enhanced service life of gears in industry, more efficient, reliable and light-weight gears needed to be designed and manufactured. Designing gear is highly complicated and intellectual field. For decades, several measures have been adopted to enhance the service life of gears such as heat treatment, adjusting micro geometry. Many physical factors accumulate to cause a gear failure, including the material of the gear. Selecting different materials for gears plays an important role in gear technology. Material selected for making a gear must satisfy two conditions: (1) manufacturability and processing requirement: (2) achieving required service life. Manufacturability requirement includes its forgeability and its response to heat treatment. Whereas, to achieve required service life, gears should transmit power to a satisfactory level when working in loading conditions as well as fulfilling mechanical property requirement such as fatigue, strength and response to heat treatment.

In industry, gear designers have been working hard for years to achieve precise gearing without error and to produce maximum service life. To reach the most refined level of gear design, designers refer to the standard such as DIN, AGMA, IS, ISO. These standards are strongly influenced by several safety factors. To reduce the cost of actual prototypes and field testing of gears, analysis software was introduced. Analysis software such as ANSYS is capable of performing finite element analysis (FEA) over not only gear teeth but each part of the gear body such as the rim. This software also provides information of bending stresses, contact stresses along with transmission error. To minimize the modeling time, preprocessor software that helps to create the geometry required for FEA, such as Solidworks could be used. Solidworks generates the three-dimensional spur gears easily. After designing and saving the geometry in solidworks, it is easy to import the same file into ANSYS. Advances in software development have opened a new era of gear analysis simulation. Computer simulation results have helped to achieve more accurate gear tooth profiles before manufacturing a practical prototype of a gear.

2. FINITE ELEMENT METHOD

The finite element method is numerical analysis technical of optioning approximate solution to a wide verity of engineering problems. Because of its diversity and flexibility as an analysis tool , it is receiving much attention in engineering school and industries in more and more engineering situation today , we find that it is necessary to obtain approximate solution to problems rather than exact close from solution it is not possible to obtain analytical mathematical solutions are many engineering's problems. An analytical solution is a mathematical expression that gives value of the desire unknown quantity an any location in the body, as consequence it is valid for infinite number of location in the body. For problem involving complex material properties and bounder condition, the engineer resource to numerical method that provide approximate that eatable solution.

2.1 Terms used in Gear

The pitch circle of a gear is the circle that represents the size of the corresponding friction roller that could replace the gear. As two gears mate, their pitch circles are tangent, with a point of contact on the line that connects the center of both circles. The pitch point is the point of contact of the two pitch circles. The pitch diameter, d, of a gear is simply the diameter of the pitch circle. Because the kinematics of a spur gear are identical to an analogous friction roller, the pitch diameter is a widely referenced gear parameter. However, because the pitch circle is located near the middle of the gear teeth, the pitch diameter cannot be measured directly from a gear. The number of teeth, N, is simply the total number of teeth on the gear. Obviously, this value must be an integer because fractional teeth cannot be used. The circular pitch can be calculated from the number of teeth and the pitch diameter of a gear.

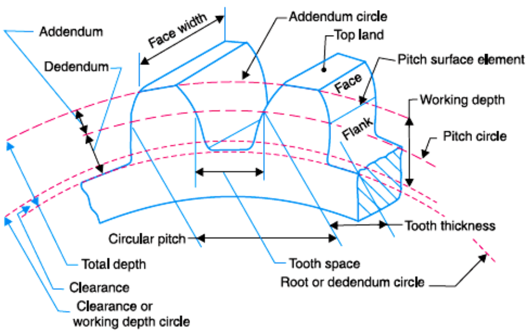

Figure 1. Terms used in Gears

The circular pitch, p_c is the distance measured along the pitch circle from a point on one tooth to the corresponding point on the adjacent tooth of the gear.

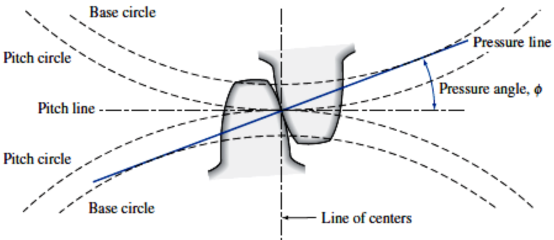

Figure 2. Pressure angle

The pressure angle, φ, is the angle between a line tangent to both pitch circles of mating gears and a line perpendicular to the surfaces of the teeth at the contact point.

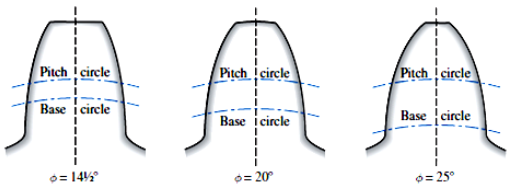

Figure 3. Pressure angle influence on tooth shape

The line tangent to the pitch circles is termed the pitch line. The line perpendicular to the surfaces of the teeth at the contact point is termed the pressure line or line of contact.

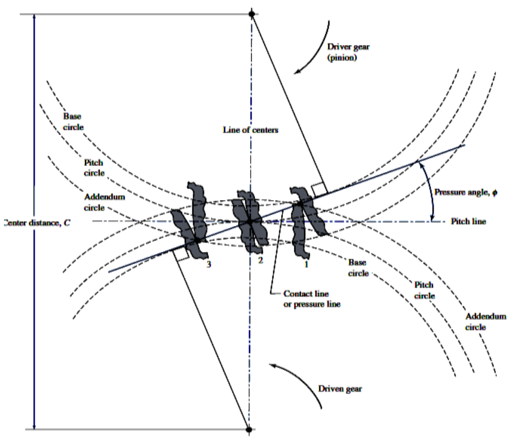

Figure 4. Gear Mating Process

Therefore, the pressure angle is measured between the pitch line and the pressure line. The pressure angle is illustrated in Figure 2. The pressure angle affects the relative shape of a gear tooth. Although gears can be manufactured in a wide range of pressure angles, most gears are standardized at 20° and 25°. Gears with 14.5° pressure angles were widely used but are now considered obsolete. They are still manufactured as replacements for older gear trains still in use. Because the pressure angle affects the shape of a tooth, two mating gears must also have the same pressure angle. Recall that forces are transmitted perpendicular to the surfaces in contact. Therefore, the force acting on a tooth is along the pressure line.

2.2 Standard Proportions of Gear Systems

The following table shows the standard proportions in module(m) for the four gear systems.

Table.1 Standard proportions of Gear systems

S. No.	Particulars	$14\frac{1}{2}°$ composite or full depth involute system	20° full depth involute system	20° stub involute system
1.	Addenddm	1 *m*	1 *m*	0.8 *m*
2.	Dedendum	1.25 *m*	1.25 *m*	1 *m*
3.	Working depth	2 *m*	2 *m*	1.60 *m*
4.	Minimum total depth	2.25 *m*	2.25 *m*	1.80 *m*
5.	Tooth thickness	1.5708 *m*	1.5708 *m*	1.5708 *m*
6.	Minimum clearance	0.25 *m*	0.25 *m*	0.2 *m*
7.	Fillet radius at root	0.4 *m*	0.4 *m*	0.4 *m*

2.3 Arc of Contact

The arc of contact is the path traced by a point on the pitch circle from the beginning to the end of engagement of a given pair of teeth. The arc GP is known as arc of approach and the arc PH is called arc of recess. The angles subtended by these arcs at O1 are called angle of approach and angle of recess respectively.

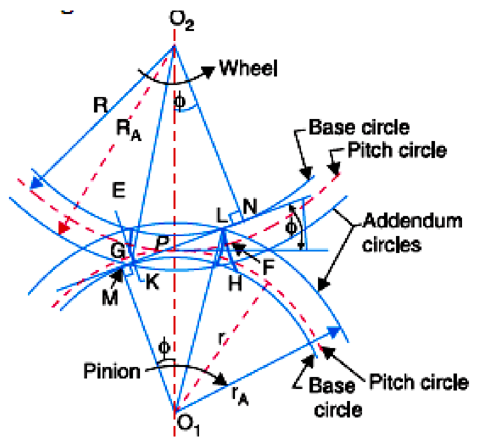

Figure 5. Path of contact

$$\text{Length of the arc of contact} = \frac{\text{length of path of contact}}{\cos\emptyset}$$

2.4 Contact Ratio

The contact ratio or the number of pairs of teeth in contact is defined as the ratio of the length of arc of contact to the circular pitch. Mathematically;

$$\text{Contact ratio} = \frac{\text{length of arc of contact}}{p_c}$$

Where p_c = circular pitch = π m; m = module

3. DESIGN SPECIFICATION

For calculating bending stress, contact stress and total deformation we have taken a standard model for designing the spur gear tooth. The following data is given for the design of

20˚ full depth spur gear made of stainless steel transmitting torque at 10 kW power with 3000rpm.

Table 2. Specification of Spur Gear Tooth

Dicription	Units	Symbol	Value
No. of teeth on pinion	-	N_p	20
No. of teeth on gear	-	N_g	20
Pressure angle	degree	ϕ	20
Module	mm	m	6
Addendum	mm	h_a	m = 6
Dedendum	mm	h_d	1.25m = 7.5
Pitch circle diameter	mm	d_p	mN = 120
Pitch circle radius	mm	r_p	60
Base circle diameter	mm	d_b	$d_p\cos\phi$ = 112.76
Addendum circle diameter	mm	d_a	d_p + 2m= 132
Dedendum circle diameter	mm	d_d	d_p-(2+π/N)m=107.06
Face width	mm	b	54
Tooth thickness	mm	t	1.5808m = 9.485
Fillet radius	mm	r_f	0.4m=2.4

Specification of Spur Gear Tooth and Material Properties of Stainless Steel are described in Table 2 and 3.

Table 3. Material Properties of Stainless Steel

Property	Value	Unit
Density	7750	kg/m^3
Coeff; of thermal expansion	1.7e-05	C^{-1}
Young's Modulus	1.93e+11	Pa
Poison's ratio	0.31	Pa
Bulk Modulus	1.69e+11	Pa
Shear Modulus	7.3664e+10	Pa
Tensile Yield Strength	2.07e+08	Pa
Compressive Yield Strength	2.07e+08	Pa
Ultimate tensile Strength	5.86e+08	pa

3.1 Theoretical Calculation

3.1.1 Contact stress calculation using Hertz Equation

Earle Buckingham(1926) used the Hertz theory to determine the contact stress between a pair of teeth while transmitting power by treating the pair of teeth in contact as cylinders of radii equal to the radii of curvature of the mating involutes at the pitch point. According to Hertz theory, when the two cylinders are pressed together, the contact stress is given by,

$$\sigma_c = \frac{2F}{\pi BL} \quad \text{(i)}$$

$$B = \sqrt{\left[\frac{2F\left(\frac{1-\mu_1^2}{E_1}+\frac{1-\mu_2^2}{E_2}\right)}{\pi L\left(\frac{1}{d_1}+\frac{1}{d_2}\right)}\right]} \quad \text{(ii)}$$

where σ_c = maximum value of contact stress (N/mm^2)
F = force pressing two cylinders together (N)
B = half width of deformation (mm)
L = axial length of cylinders (mm)
d_1, d_2 = diameters of two cylinders (mm)
E_1, E_2 = moduli of elasticity of two cylinder materials (N/mm^2)
μ_1,μ_2 = poisson's ratio of materials

Substituting the value of half width of deformation B in equation(i) and squaring both sides,

$$\sigma_c^2 = \frac{F}{\pi L}\left[\frac{\left(\frac{1}{r_1}+\frac{1}{r_2}\right)}{\left(\frac{1-\mu_1}{E_1}+\frac{1-\mu_2}{E_2}\right)}\right]$$

(iii)

If the material of both cylinders are the same, then the moduli of elasticity and poisson's ratio will be equal. Substituting $E_1 = E_2 = E$ and $\mu_1 = \mu_2 = \mu$ in eq.(iii),

$$\sigma_c^2 = \frac{F}{2\pi L}\left[\frac{\left(\frac{1}{r_1}+\frac{1}{r_2}\right)}{\left(\frac{1-\mu}{E}\right)}\right]$$

(iv)

Now applying this equation to a pair of spur gear teeth in contact, replacing the radii r_1 and $_{r2}$ by the radii of curvature at the pitch point.

$$r_1 = \frac{d_{pp}\sin\emptyset}{2} \quad \text{and} \quad r_2 = \frac{d_{pg}\sin\emptyset}{2}$$

where d_{pp} = pitch circle diameter of pinion
d_{pg} = pitch circle diameter of gear

Since the pinion and gear have equal geometry in all respects as given in table 2, therefore

$d_{pp} = d_{pg} = d_p$,

$$r_1 = \frac{d_p\sin\emptyset}{2} \quad \text{and} \quad r_2 = \frac{d_p\sin\emptyset}{2}$$

$$\Rightarrow \quad r_1 = r_1 = r = \frac{d_p\sin\emptyset}{2}$$

(v)

⇨ substitute in eq.(iv)we get

$$\sigma_c^2 = \frac{1}{\pi(1-\mu}\left[\frac{FE}{Lr}\right]$$

(vi)

For both materials as stainless steel, from table 3 poisson's ratio μ = 0.31and substitute in eq.(vi) and solving;

$$\sigma_c = 0.6792\left(\frac{FE}{Lr}\right)^{\frac{1}{2}}$$

(vii)

From table 3 modulus of elasticity E = 193000MPa and substitute in (vii),

$$\sigma_c = 0.6792\left(\frac{F\times 193000}{Lr}\right)^{\frac{1}{2}}$$

$$\sigma_c = 298.386\left(\frac{F}{Lr}\right)^{\frac{1}{2}}$$

(viii)

from (v) $\quad r = \frac{d_p\sin\emptyset}{2} = r_p\sin\emptyset$

Therefore $\quad \sigma_c = 298.386\left(\frac{F}{Lrp\sin\phi}\right)^{\frac{1}{2}}$

(ix)

Now, $F = \frac{F_t}{\cos\emptyset}$ where Ft is the tangential component of the resultant force F between two meshing teeth. Substitute this value in (ix),

$$\sigma_c = 298.386\left(\frac{F_t}{Lr_p\sin\emptyset\cos\emptyset}\right)^{\frac{1}{2}}$$

(x)

Also the axial length L is equal to the face width b of spur gears, therefore replacing L by b in (x),

$$\sigma_c = 298.386\left(\frac{F_t}{br_p\sin\emptyset\cos\emptyset}\right)^{\frac{1}{2}}$$

(xi)

Equation (xi) is the general mathematical model for evaluating the contact stress for a pair of stainless steel spur gear teeth in contact, for the equal geometry and dimensions of pinion and gear in mesh.

Considering the power to be transmitted P = 10 kW and the rotational speed of the pinion n_p = 3000 rpm, the tangential component of force can be obtained from;

$$P = \frac{2\pi n_p T}{60\times10^6} \qquad \text{(xii)}$$

Where T is transmitting torque in N-mm.
=> T = 31831 N-mm
Also $F_t = \frac{2T}{d_p}$ and from table 2, d_p = 120 mm
=> F_t = 530.517 N
Again b = 54 mm , $\phi = 20^\circ$ and r_p = 60 mm are substituted in (xi) σ_c = 212.98 MPa
Allowable maximum contact stress,

$$\sigma_{c,all} = \frac{\sigma_c}{S.F}$$ where S.F is safety factor.

Taking S.F as 2.736, $\sigma_{c,all}$ = 77.84 MPa
Again from eq.(xi)

$$\sigma_c = 298.386\left(\frac{F_t}{br_p sin\phi cos\phi}\right)^{\frac{1}{2}}$$

$$F_t = br_p\sin\phi\cos\phi\left(\frac{\sigma_c}{298.386}\right)^2$$

(xiii)
Replacing σ_c by the maximum allowable Hertz stress σ_H in(xii), the tooth surface strength of pinion is;

$$F_{ts} = br_p\sin\phi\cos\phi\left(\frac{\sigma_H}{298.386}\right)^2$$

(xiv)
The maximum allowable Hertz stress for stainless steel spur gears is 405.015 N/mm^2.

F_{ts} = 1918.527 N

For the design to be safe, the tooth surface strength F_{ts} must be greater than the dynamic load on gear tooth F_d. The dynamic load on gear tooth is given by;

$$F_d = \frac{21v(Ceb+F_t)}{21v+\sqrt{Ceb+F_t}}$$

(xv)
Where v = pitch line velocity (m/s)
C = deformation factor (N/mm^2)
e = sum of error between two meshing teeth (mm)
b = face width of tooth (mm)
F_t = tangential component of force (N)
The deformation factor C is given by,

$$C = \frac{k}{\left[\frac{1}{E_1}+\frac{1}{E_2}\right]}$$

where k = 0.111 constant depending on form of tooth

⇨ C = 10711.5 N/mm^2

Pitch line velocity $v = \frac{\pi d_p n}{60\times10^3}$; n = 3000 rpm and d_p = 120mm, v = 18.84 m/s
The error e is a function of the quality of the gear and the method of manufacturing. There are twelve different grades of gears as listed in table 4.

Table 4. Tolerances on the adjacent pitch

Grade	e(microns)
1	$0.80 + 0.06\phi$
2	$1.25 + 0.10\phi$
3	$2.00 + 0.16\phi$
4	$3.20 + 0.25\phi$
5	$5.00 + 0.40\phi$
6	$8.00 + 0.63\phi$
7	$11.00 + 0.90\phi$
8	$16.00 + 1.25\phi$
9	$22.00 + 1.80\phi$
10	$32.00 + 2.50\phi$
11	$45.00 + 3.55\phi$
12	$63.00 + 5.00\phi$

The tolerance factor ϕ is given by,

$$\phi = m + 0.25\sqrt{d_p}$$

where m = module, d_p = pitch circle diameter

⇨ for top precision as grade 1;
e = $2e_p = 2e_g = 2(0.8 + 0.06\phi) = 2.6486\times10^{-3}$ mm
from eq.(xv) the dynamic load F_d = 1850.15 N

F_{ts} = 1918.517 N > F_d = 1850.15 N
Therefore, the design is safe from surface durability consideration.

3.2 Assumption of Lewis Equation

The analysis of bending stress in gear tooth was done by Mr. Wilfred Lewis in his paper, ‘The investigation of the strength of gear tooth’ submitted at the Engineers club of Philadelphia in 1892. Even today, the Lewis equation is considered as the basic equation in the design of gears. Lewis considered gear tooth as a cantilever beam with static normal force F applied at the tip. Assumptions made in the derivation are:

1. The full load is applied to the tip of a single tooth in static condition.
2. The radial component is negligible.
3. The load is distributed uniformly across the full face width.
4. Forces due to tooth sliding friction are negligible.
5. Stress concentration in the tooth fillet is negligible.

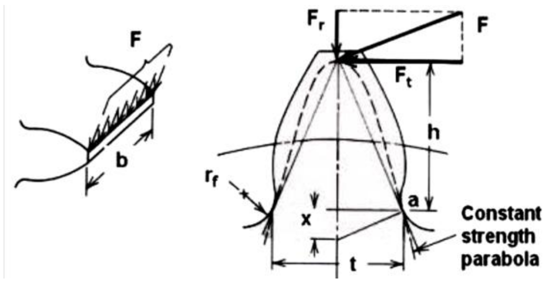

Figure 6. Force on a gear tooth

In the above Figure 3.1, the following notations are used: F is the Full load, F_r and F_t are the Radial and Tangential component of the full load, h, b and t are the height, face-width and thickness of the tooth at critical section respectively.

3.2.1 Calculation of Bending Stress

In calculation of bending stress of tooth we consider the Lewis assumption as discussed above in 3.2. From the Figure 3 at point ‘a’

Bending moment $M_b = Ft * h$ (1)
Area moment of inertia I = $bt^3/12$ (2)
Then the bending stress is given by $\sigma_b = \frac{6hF_t}{bt^2}$

from the Eq. (1) and Eq. (2). After the rearranging we have

$$F_t = b*\sigma_b*\frac{t^2}{6h} \qquad (3)$$

Multiplying numerator and denominator by module (m) from the Eq. (3) we have the tangential component of the force is

given by $F_t = m*b*\sigma_b*\frac{t^2}{6hm}$ (4)

$Y = \frac{t^2}{6hm}$ is known as Lewis form factor. Equation (4) can be

rewritten as $F_t = m * b * \sigma * Y$ (5)
When the tangential force increased the stress also increases. When the stress reaches the permissible magnitude of bending

stress the corresponding force F_t is known as Beam strength and denoted by S_b. So replacing F_t in the Eq. (5) we have

$$S_b = m * b * \sigma * Y. \quad (6)$$

Consider torque T = 31831 N-mm at 3000 rpm. The tangential load F_t can be found as

$$F_t = 2*T/d_p \; ; \qquad F_t = 530.517 \text{ N} ,$$

The value of bending stress is given from Eq.(5) if form factor Y = 0.32, σ_b = 5.117 MPa.

Ultimate tensile strength of gear material is 586 MPa.
Consider safety factor as 3, the allowable bending stress is 195.33 MPa $>$ 5.117 MPa.

So the design is satisfactory.

As the gear and pinion are identical there is no need to check the following relation i.e. strength of gear $<$ strength of pinion.

3.2.2 Calculation of Total Deformation

It is observed that the cross section of the gear tooth varies from free end to the fixed end. Lewis has assumed it as a constant strength parabola. Using Castigliano's Theorem total deformation of the tooth can be found with minor error. For linearly elastic structure, where external forces only cause deformations, the complementary energy is equal to the strain energy. For such structures, the Castigliano's first theorem may be stated as the first partial derivative of the strain energy of the structure with respect to any particular force gives the displacement of the point of application of that force in the direction of its line of action. The theory applies to both linear and rotational deflection $\delta = \frac{\delta u}{\delta F}$. It should be clear that Castigliano's theorem finds the deflection at the point of application of the load in the direction of the load.

Here U is the strain energy given by U = $\int_0^l \frac{M^2}{2EI}\, dx$, where M is the moment due to the load. Consider the parabolic tooth of height h and thickness t. The equation of parabola $y^2 = 4*a*x$, the boundary condition at x = h, y = t/2. After substitution

$y^2 = (t^2 x / 4 h)$ and $y^3 = (t / 2)^3 (x / h)^{1.5}$

Putting $M = F_t * x$, $I = (2/3)*b*y^3$, the strain energy will be

$$U = \frac{8h^3 F_t}{Ebt^3} \quad (7)$$

Again deflection is given by $\delta = \frac{16 F_t h^3}{Ebt^3}$ (8)

4. STATIC ANALYSIS

4.1 Meshing

Meshing is basically the division of the entire model into small cell so that at each and every cell the equations are solved. It gives the accurate solution and also improves the quality of solution. Here the element size of 1 mm with medium smoothing is considered for mesh generation.

4.2 Boundary Conditions

Based on the assumptions of Lewis equation, the boundary conditions are set in ANSYS Workbench. The fixed support is used at the root end of the tooth and the force is applied on the face having components in Y and Z directions.

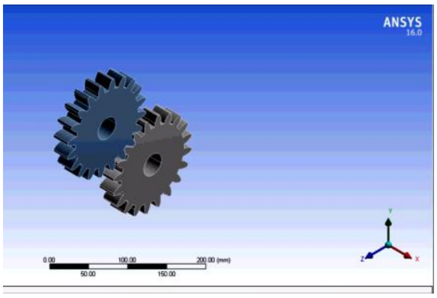

Figure 7. Gear Geometry

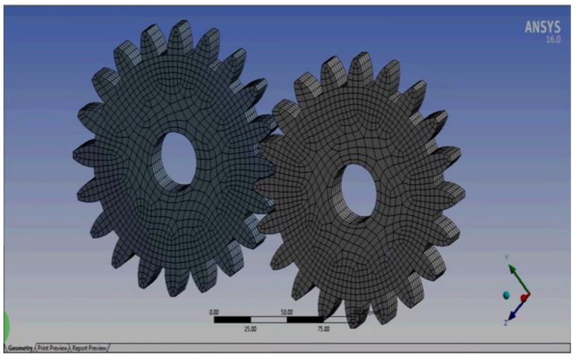

Figure 8. Meshing of Gears

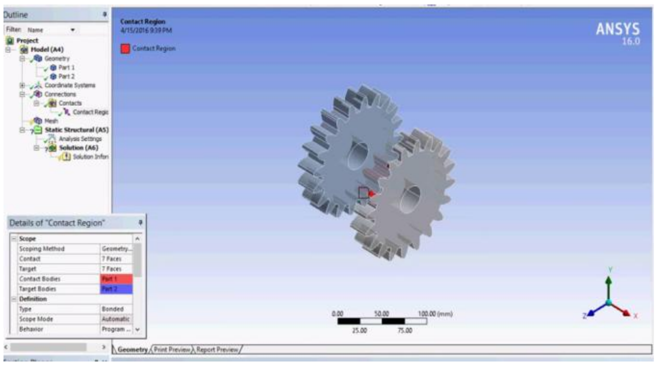

Figure 9. Contact region

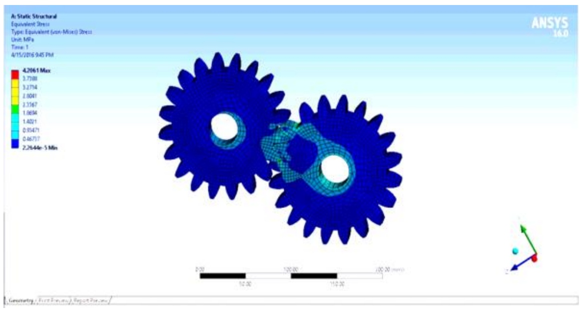

Figure 10. Distribution of Equivalent (Von Mises) stress

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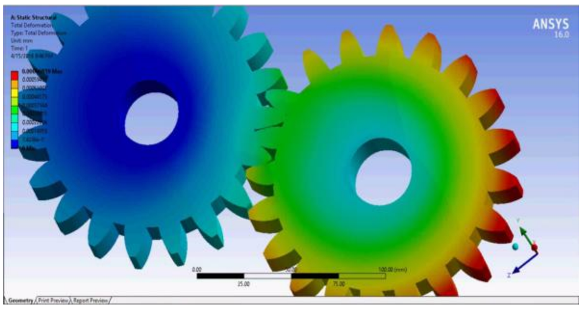

Figure 11. Total Deformation

FEA provides results that are comparable with theoretical analysis results as was in the contact stress analysis of spur gears in the present study. FEA can predict whether a product will break, wear out, or work the way it was designed. Hence, FEA can prove very helpful in the product development process by forecasting its behavior in operation. A number of research studies have been carried out in the context of spur gear using different types of materials. An extended version of the above work based on the same software can also be carried out for the analysis of shear stress between two mating gears.

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5. RESULTS AND CONCLUSION

In the present work, the spur gear tooth is modeled and is analyzed in the static structural domain of ANSYS software. The results found in the above figures are the maximum equivalent stress is 4.206 MPa and total deformation is 0.0006689 mm. It is concluded that for the given specification, the maximum bending stress and total deformation can be found for a torque specification.