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Design of Helical Gear and Analysis on Gear Tooth

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ABSTRACT

Gears are mainly used to transmit the power in mechanical power transmission systems. These gears play a most predominant role in many automobile and micro electro mechanical systems. One of the main reason of the failure in the gear is bending stresses and vibrations also to be taken into account. But the stresses are occurred due to the contact between two gears while power transmission process is started.

Due to meshing between two gears contact stresses are evolved, which are determined by using analyzing software called ANSYS. Finding stresses has become most popular in research on gears to minimize the vibrations, bending stresses and also reducing the mass percentage in gears. These stresses are used to find the optimum design in the gears which reduces the chances of failure. The model is generated by using CATIAV5 OR PRO-E,and ANSYS is used for numerical analysis.

The analytical study is based on Hertz's equation. Study is conducted by varying the geometrical profile of the teeth and to find the change in contact stresses between gears. It is therefore observed that more contact stresses are obtained in modified gears. Both the results calculated using ANSYS and compared according to the given moment of inertia.

1. INTRODUCTION

Gears are most commonly used for power transmission in all the modern devices. These toothed wheels are used to change the speed or power between input and output. They have gained wide range of acceptance in all kinds of applications and have been used extensively in the high-speed marine engines.

In the present era of sophisticated technology, gear design has evolved to a high degree of perfection.

The design and manufacture of precision cut gears, made from materials of high strength, have made it possible to produce gears which are capable of transmitting extremely large loads at extremely high circumferential speeds with very little noise, vibration and other undesirable aspects of gear drives. A gear is a toothed wheel having a special tooth space of profile enabling it to mesh smoothly with other gears and power transmission takes place from one shaft to other by means of successive engagement of teeth.

Gears operate in pairs, the smallest of the pair being called "pinion" and the larger one "gear". Usually the pinion drives the gear and the system acts as a speed reducer and torque converter.

2. GEOMETRY OF HELICAL GEARS

Helical gears offer a refinement over spur gears. The leading edges of the teeth are not parallel to the axis of rotation, but are set at an angle. Since the gear is curved, this angling causes the tooth shape to be a segment of a helix. Helical gears can be meshed in a parallel or crossed orientations. The former refers to when the shafts are parallel to each other; this is the most common orientation. In the latter, the shafts are non-parallel.

Quite commonly helical gears are used with the helix angle of one having the negative of the helix angle of the other; such a pair might also be referred to as having a right-handed helix and a left-handed helix of equal angles. The two equal but opposite angles add to zero: the angle between shafts is zero – that is, the

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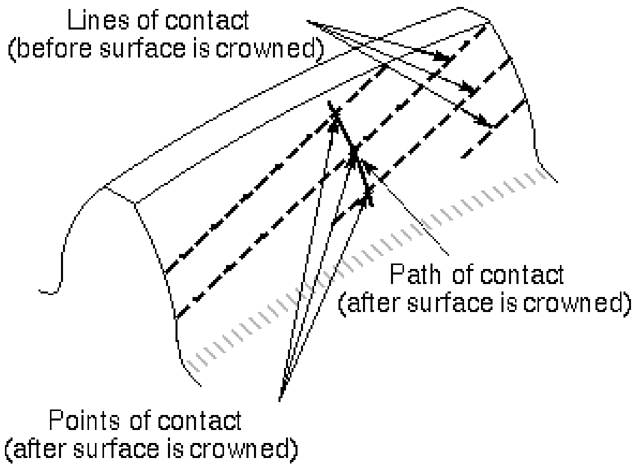
shafts are parallel. Where the sum or the difference (as described in the equations above) is not zero the shafts are crossed. For shafts crossed at right angles the helix angles are of the same hand because they must add to 90 degrees.

3.1 Helical gear nomenclature



3.2 Helical Gear geometrical proportions

- p = Circular pitch = $d_g . p / z_g = d_p . p / z_p$
- p_n = Normal circular pitch = $p . \cos\beta$
- P_n =Normal diametrical pitch = $P / \cos\beta$
- p_x = Axial pitch = $p_c / \tan\beta$
- m_n =Normal module = $m / \cos\beta$
- α_n = Normal pressure angle = $\tan^{-1} (\tan\alpha . \cos\beta)$
- β =Helix angle
- d_g = Pitch diameter gear = $z_g . m$
- d_p = Pitch diameter pinion = $z_p . m$
- a =Center distance = $(z_p + z_g)^* m_n /2 \cos\beta$
- a_a = Addendum = m
- a_f =Dedendum = 1.25^*m

General Procedures to Create an Involute Curve

The sequence of procedures employed to generate the involute curve are illustrated as follows: -

1. Set up the geometric parameters Number of teeth Diametric Pitch Pressure angle Pitch diameter Face width Helix angle

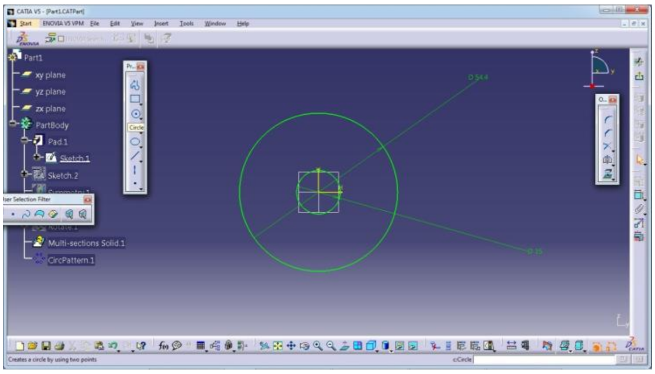
2. Create the basic geometry such as addendum, dedendum and pitch circles in support of the gear tooth.

3. Define the involute tooth profile with datum curve by equation using cylindrical coordinate system.

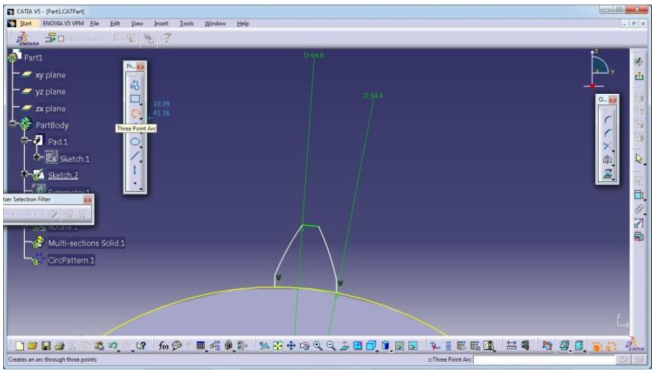
4. Create the tooth solid feature with a cut and extrusion. Additional helical datum

Number of teeth	25
Diametral pitch (p) [mm]	60
Pressure angle	20 degree
Addendum [mm]	1/p
Dedendum [mm]	1.25/p
Helix angle	12 degree

4. THE FOLLOWING ARE THE PICS OF SEQUENCIAL DESIGN PROCEDURE IN CATIA



1 helical gear base diagram



helical gear involute

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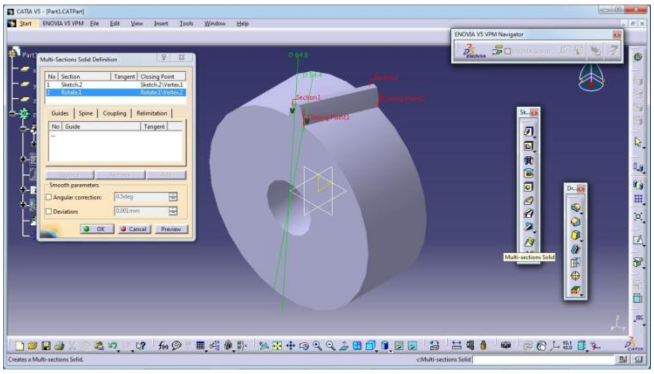
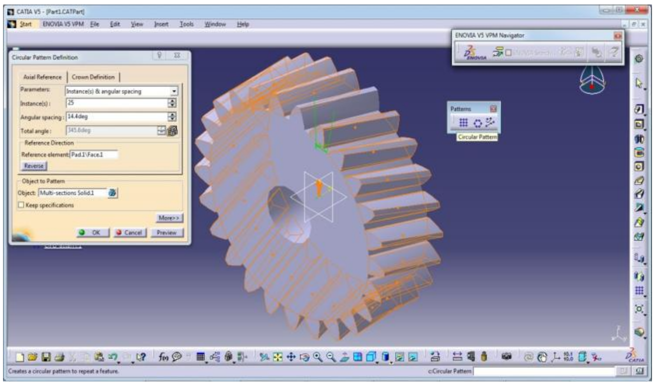
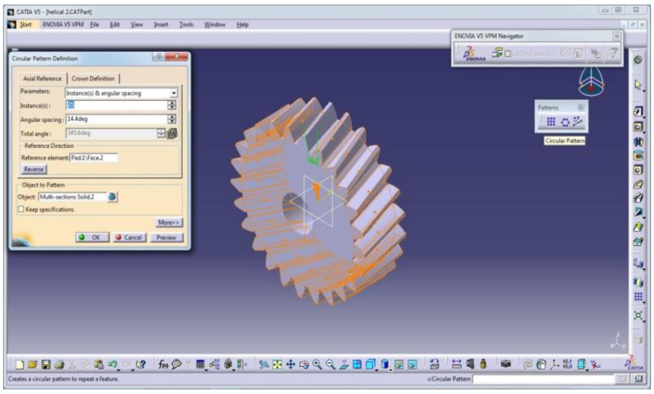


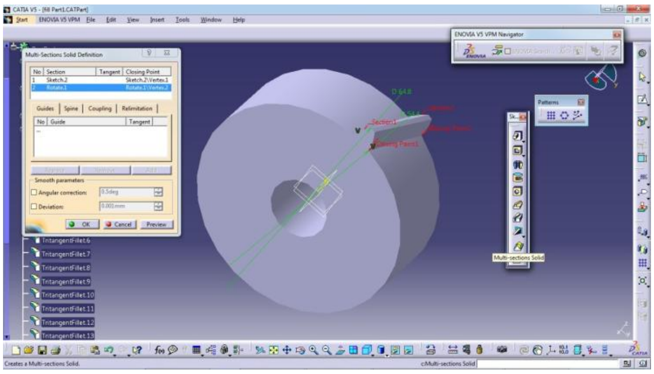
helical gear teeth



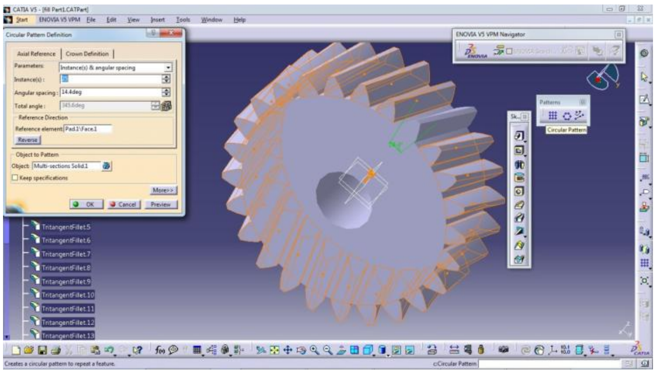
helical gear teeth generation



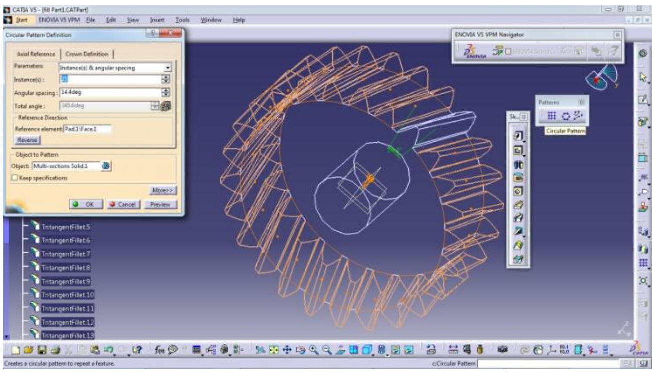
driven helical gear teeth generation



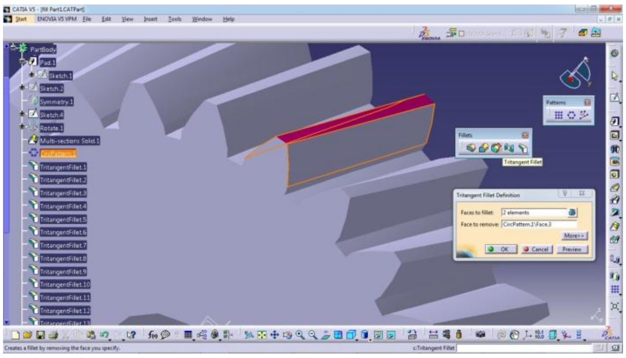
driving helical gear with fillet...teeth generation



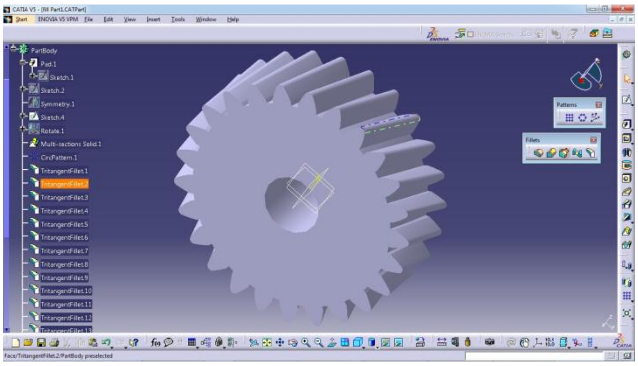
driving helical gear with fillet...teeth generation using circular pattern



driving helical gear with fillet...teeth generation using circular pattern (wireframe)



driving helical gear with fillet



driven helical gear with fillet complete

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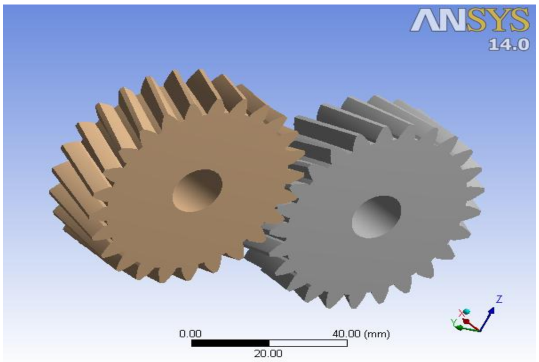
7.1 INTRODUCTION TO ANSYS:

ANSYS Stands for Analysis System Product.

Dr. John Swanson founded ANSYS. Inc in 1970 with a vision to commercialize the concept of computer simulated engineering, establishing himself as one of the pioneers of Finite Element Analysis (FEA). ANSYS inc. supports the ongoing development of innovative technology and delivers flexible, enterprise wide engineering systems that enable companies to solve the full range of analysis problem, maximizing their existing investments in software and hardware. ANSYS Inc. continues its role as a technical innovator. It also supports a process-centric approach to design and manufacturing, allowing the users to avoid expensive and time-consuming "built and break" cycles. ANSYS analysis and simulation tools give customers ease-of-use, data compatibility, multi platform support and coupled field multi-physics capabilities.

ANSYS RESULTS



Properties			
Volume	52368 mm³	21205 mm³	52368 mm³
Mass	0.40585 kg	0.16434 kg	0.40585 kg
Centroid X	6.9988e-005 mm	2.2122e-015 mm	3.8961e-005 mm
Centroid Y	-60.22 mm	-6.3815e-017 mm	3.1123e-005 mm
Centroid Z	2.2936 mm	1.1904e-016 mm	1.4959e-006 mm
Moment of Inertia Ip1	195.07 kg·mm²	4.5605 kg·mm²	195.02 kg·mm²
Moment of Inertia Ip2	111.1 kg·mm²	198.34 kg·mm²	111.07 kg·mm²
Moment of Inertia Ip3	111.09 kg·mm²	198.34 kg·mm²	111.07 kg·mm²
Statistics			
Nodes	14332	0	14449
Elements	8011	0	8067
Mesh Metric	None		

FIGURE 1

Model (A4) > Static Structural (A5) > Moment

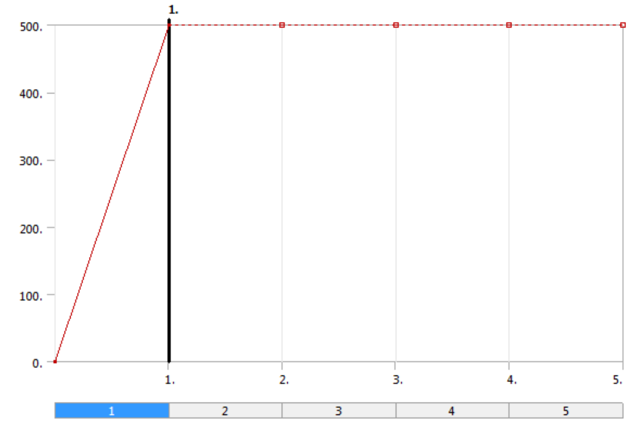
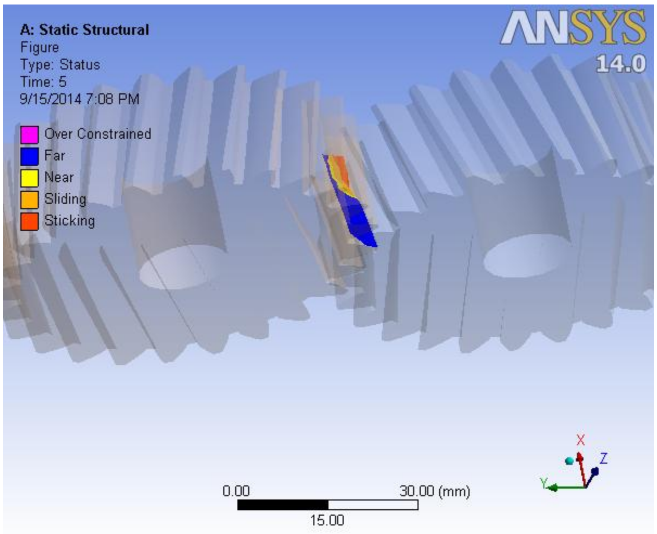


FIGURE 4

Model (A4) > Static Structural (A5) > Solution (A6) > Contact Tool > Status > Figure



Material Data

Stainless Steel

TABLE 22

Stainless Steel > Constants

Density	7.75e-006 kg mm^-3
Coefficient of Thermal Expansion	1.7e-005 C^-1
Specific Heat	4.8e+005 mJ kg^-1 C^-1
Thermal Conductivity	1.51e-002 W mm^-1 C^-1
Resistivity	7.7e-004 ohm mm

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Figure 1

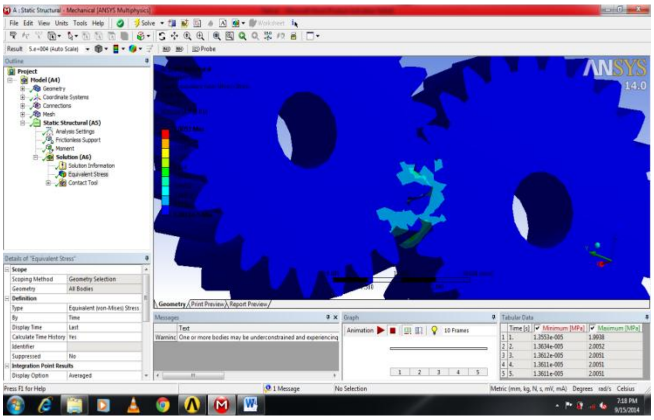
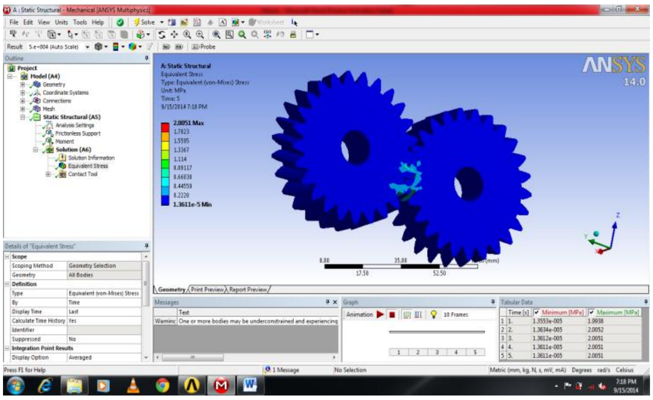
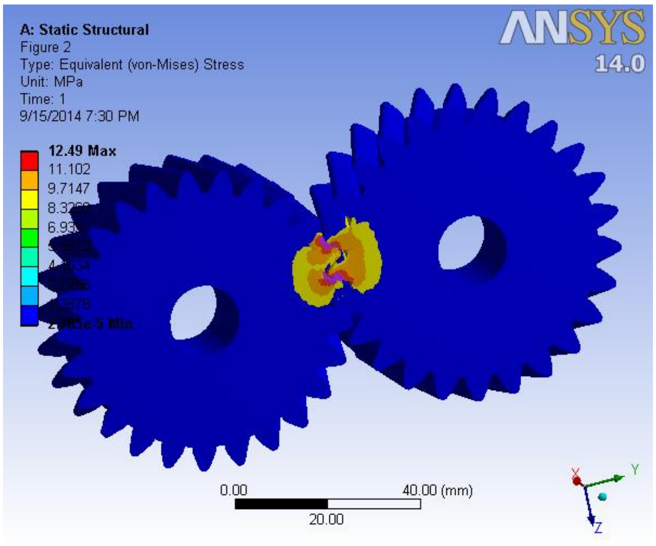


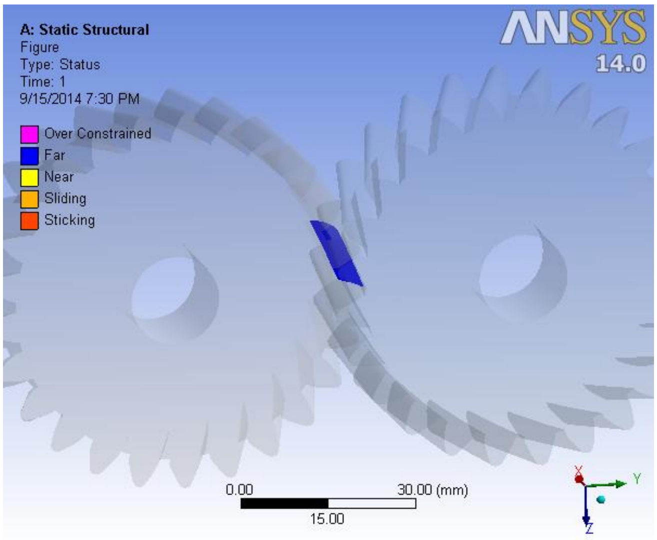
Figure 2



RESULT 2:
FIGURE 3

Model (A4) > Static Structural (A5) > Solution (A6) > Equivalent Stress > Figure 2





Comparison of Values of Modified Helical Gear and Normal Helical Gear

Table 1: Comparison Of Von – Misses Stresses of Modified Gear and Normal Gear	
Gear Type	**Von Misses Stresses**
Normal Helical Gear	2.0051 Mpa
Modified Helical Gear	12.49 Mpa

CONCLUSION

Gear analysis uses a number of assumptions, calculations and simplification which are intended to determine the maximum stress values in analytical method. In this paper parametric study is also made by varying the geometry of the teeth to investigate their effect of contact stresses in helical gears. As the strength of the gear tooth is important parameter to resist failure. In this study, it is shown that the effective method to estimate the contact stresses using three dimensional model of both the different gears and to verify the accuracy of this method.

The two different result obtained by the ansys with different geometries are compared. Based on the result from the contact stress analysis the hardness of the gear tooth profile can be improved to resist pitting failure: a phenomena in which a small particle are removed from the surface of the tooth that is because of the high contact stresses that are present between mating teeth, as of the obtained data the contact

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stresses which are acting on the modified helical gears are more when compared to the standard helical so these paper pretends to be failure theory by which the design aspects are to no changed to reduce the contact stresses.

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