Evaluation of Bending Stress at Fillet Region of an Asymmetric Gear with a Hole as Stress Relieving Feature using a FEA Software ANSYS

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ABSTRACT
Gear is a vital and inevitable component in a machine. It is been a subject of research from the very first day of development of gear that, how to minimize gear failure? Many researches have already been done on the improvement of gears' stress carrying capacity. When a pair of gear meshes with each other, failure occurs due to cracks generated at the region of undercut caused by bending stress. Bending stress can be minimized by introducing a stress reliving feature on the gear surface. In few machines like Helicopter and Airplane gears are mainly used for forward rotation. In asymmetric gear one of the curved surfaces where engagement with other gear takes place is kept larger than the not mating curved surface. The curved surface where mating takes place with other gear is called drive surface and the curved surface where mating does not takes place is called coast surface. So, a smaller asymmetric gear transmits same torque transmitted by a larger symmetric gear and thus it is useful for aviation vehicle where weight of the vehicle is very important. In this work the above mentioned fact that, an asymmetric gear transmits same torque transmitted by a larger symmetric gear, has been proved by comparing bending stress carrying capacity by a symmetric gear and an asymmetric gear. The bending stress carrying capacity is calculated in ANSYS by structural analysis of the gear tooth under a given loading condition. The load value and gear geometry parameters have been taken loading condition mentioned in reference [17].In the present work ANSYS has been used as a FEA software and 3-Dimensional model of the gear geometry has been considered. The bending stress thus found out for symmetric as well as for the asymmetric gear tooth.

Keywords
Gear, Stress, FEA Software, Ansys.

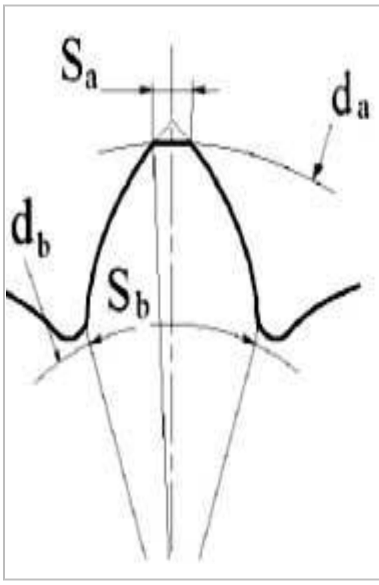
1. INTRODUCTION

1.1 Gear Design
The design of gears is a long process. The traditional gear design offers benefits like:

- Interchangeability of the gears
- Low tool inventory
- Easy gear design process

Actually traditional gear design relies heavily on operating conditions and performance parameters. These necessitated the need for an alternative gear design.

1.1.1Direct Gear Design Method
In case of Direct Gear Design, a gear tooth is not defined by using the typical generating rack parameters such as module, diametral pitch or a pressure angle. Instead the gear tooth is represented by 2 involutes of a base circle (see the image) and the circular distance (also called base tooth thickness) between them. The tooth height is restricted by the outside diameter for avoiding a pointed tooth tip and making available a preferred tip tooth thickness.



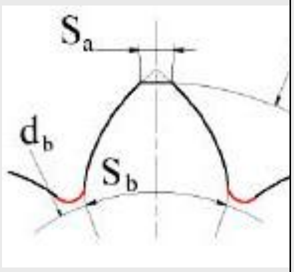
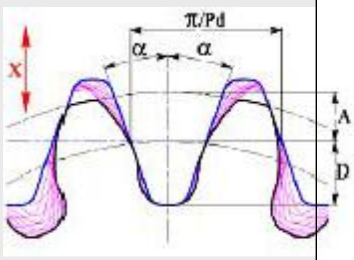
Sa= tip tooth thickness
Sb= base tooth thickness
da= outside diameter
db = the involutes

Fig1: Symmetric Gear with involute Profile.

1.1.1.1Advantages of a direct gear design
- An increase in the load capacity (15 to 30%)
- Weight and size reduction (10 to 20%)
- A longer life
- Reduction in noise and vibration (a finer pitch along with more teeth leads to higher contact ratio for a given center distance).
- Increase in the gear efficiency (atleast1 to 2% per stage)

1.1.1.2Traditional vs. Direct Gear Design

The following table highlights the points of differences between a direct and traditional gear design.

Direct Gear Design	**Traditional Gear Design**
	
Basic Principle	**Basic Principle**
Gear design is governed by application (performance based parameters).	Gear design is governed by manufacturing (based on parameters of cutting tool profile).
Custom Application Gears • Metal and plastic moulded, forged, powder metal, die cast gears. • High production machined gears. • Gears that has special requirements and also gears for extreme applications.	Custom Application Gears • Stock gears • Gearboxes that has interchangeable gear sets (such as old machine tools) • Prototype Mechanical drive • Machined gears with low production

1.2 Asymmetric Gears Design

Direct Gear Design is perfectly applicable for gears that have asymmetric teeth. This is due to the reason that in case of Asymmetric Gears, no standard is applicable for asymmetric generating racks. The approach of Direct Gear Design for asymmetric gears is identical for symmetric gears. It must be noted the degree or extent of asymmetry and the selection of drive profile for these gears is totally dependent on the application. Unlike traditional gears, Asymmetric gear design is not limited by the effects of standardized tooling or the approach of tool based design.

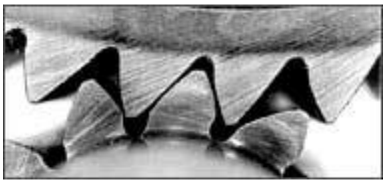


Fig 2: Asymmetric Gear meshing.

2. PROBLEM IDENTIFICATION

Gear is a vital and inevitable component in a machine. It is been a subject of research from the very first day of development of gear that, how to minimize gear failure? Many researchers have already been done on the improvement of gears' stress carrying capacity. When a pair of gear meshes with each other, failure occurs due to cracks generated at the region of undercut caused by bending stress.

In few machines like Helicopter and Airplane gears are mainly used for forward rotation. In these cases asymmetric gear is very useful to increase the overall efficiency of the machine.

In this work an asymmetric gear has been tested virtually with ANSYS under a predefined loading and it has been investigated how bending stress changes at the fillet region of the asymmetric gear

3. PROJECT METHODOLOGY

In the present work the analysis of a symmetric gear tooth and an asymmetric gear tooth of same dimension as taken by Frederick W. Brown et al (Reference [17]) has been done using ANSYS to find out maximum bending stress at the fillet region of gear tooth. Modeling of these gears has been done using a 3-D modeling software Pro-Engineer Wildfire 5.0 parametrically with the gear design parameters as mentioned in reference [17]. To model parametrically first the parameters have been collected or referred from reference [17]. The parameters have been mentioned below in a table.

Table 1: Parameters of Symmetric and Asymmetric Gear

Parameters	**Symmetric Toothed Gear**	**Asymmetric toothed Gear**
Number of Tooth (N)	32	32
Diametral Pitch (p)	0.21	0.21
Drive Pressure Angle ($\varnothing_d$)	25°	35°
Coast Pressure Angle ($\varnothing_c$)	25°	15°
Load	6950.334N	6950.334N
Load Application Radius	81.28mm	81.28mm

With the help of the involute curve a partial gear profile has been generated and then extruded to generate the partial 3-D model of gear teeth.

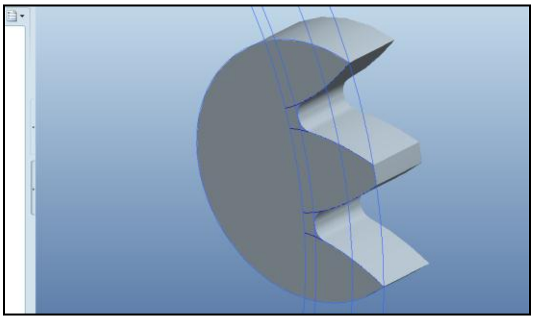


Fig 3: 3-D profile of partial gear teeth.

Now, the partial 3-D model has been imported to the analysis software ANSYS for the purpose of simulation to find out Von-Misses stress at fillet region of the gear tooth under a loading condition mentioned in reference [17]. Actually load is applied at a point in vicinity of pitch radius but, as it is not possible to manipulate the meshing procedure so that a node or a series of nodes could be available at the point of actual loading, load is considered to be located at gear tooth tip. For this assumption a modification is need for pressure angle. The equation for calculation of modified pressure angle has been mentioned below to make the above consideration or assumption effective.

The equation is:

$$\emptyset_m = \emptyset - \frac{s_a}{2r_a} \qquad (1.1)$$

Where-

$\emptyset_m$ is modified pressure angle.

$\emptyset$ is actual pressure angle.

s_a is tooth thickness at addendum circle.

r_a is addendum circle radius.

From the above equation the modified pressure angle has been calculated as 24.05°.

The load has been applied in two resolved directions. One in X-direction and other in Y-direction. As there are seven nodes on the edge of the tooth, the loads have further been divided by seven.. After application of load the back rounded portion of the gear tooth has been fixed by imposing 'All Degree of Freedom' to zero. Figure below depicts the gear tooth with DOF imposed.

Mechanical properties of alloy steel Grade-9310: Young's modulus (E): 2e5N/mm^2
Density(ρ) : 8e-6Kg/mm^3 Poisson's ratio(ν) : 0.3, Yield Strength : 439.9N/mm^2

4. DISCUSSIONS AND CONCLUSION

On completion of load imposing and DOF imposing the FEA model of gear tooth has been solved in ANSYS. From the solution following post processing has been derived.

i) Contour plotting of Von-Misses Stress over whole tooth model.

ii) Graph plotting of Von-Misses stress for a cross-section.

Figure below shows the Von-Misses stress of gear tooth.

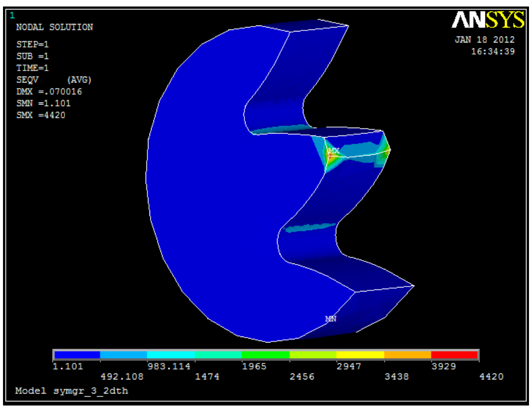


Fig 4: Von-Misses Stress Distribution of symmetric gear.

To find out bending stress at the fillet region it is needed to plot a graph between Von-Misses stress at different nodes of any section and distance of those nodes from a reference point.

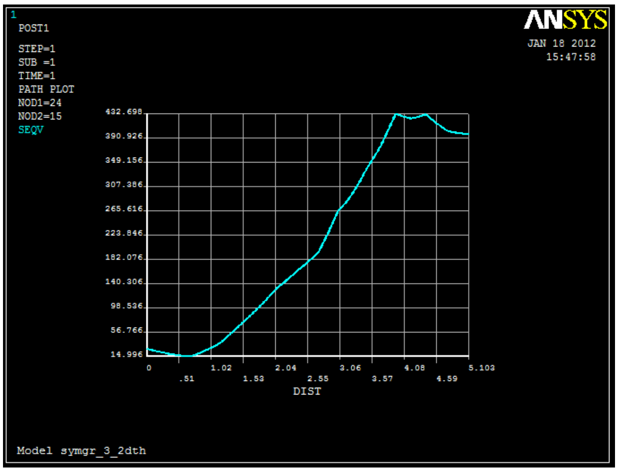


Fig 5: Graph of Von-Misses Stress Distribution at fillet area of symmetric gear.

From the above figure it is clear that maximum bending stress is 432.698 N/mm^2 and from contour plot it can be surely said that the maximum bending stress occurs at fillet area.

After validation of symmetric gear tooth an asymmetric gear tooth has been generated in Pro/Engineer Wildfire 5.0 software using parameters as mentioned in table 1.

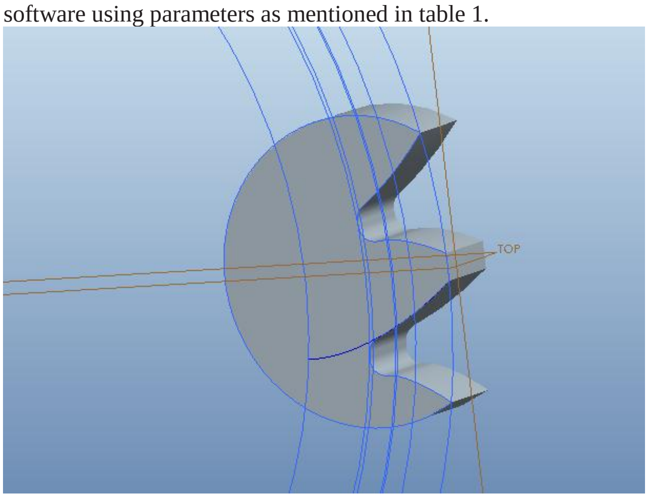


Fig 6: 3-dimensional model of an asymmetric gear tooth.

Figure below shows the Von-Misses stress of gear tooth.

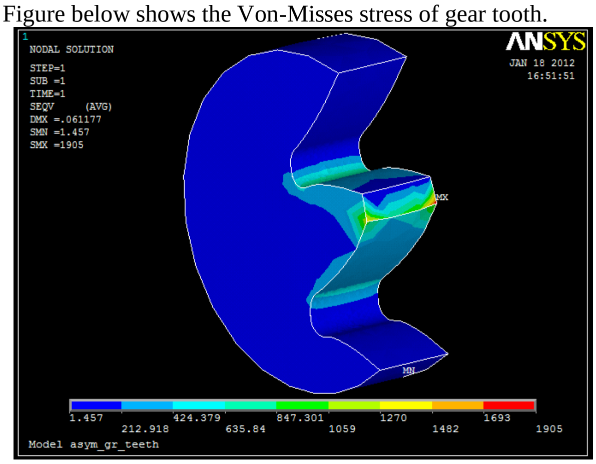


Fig 7: Von-Misses Stress Distribution of asymmetric gear.

After completion of 3-d model generation it has been imported in ANSYS for structural analysis to find out the maximum bending stress generated at the fillet region of the gear tooth under a loading condition as per Frederick W. Brown et al (Reference [17]). After imposing loads and boundary conditions i.e. degree of freedom the model has been solved.

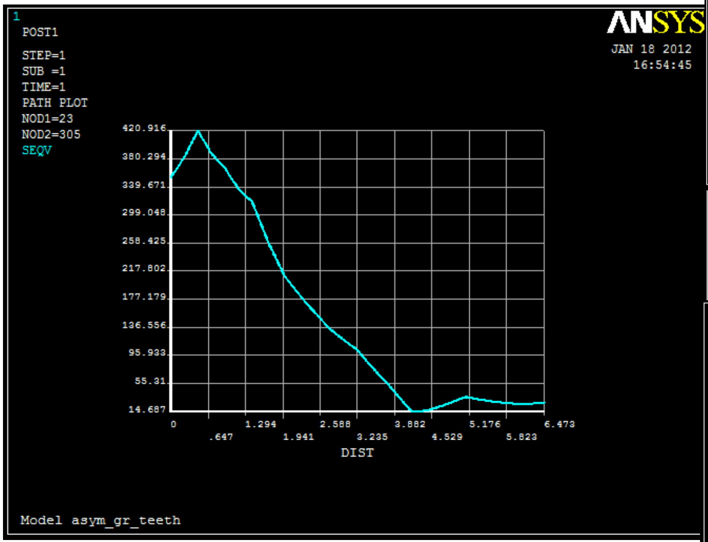


Fig 8: Graph of Von-Misses Stress Distribution at fillet area of asymmetric gear.

The above graph shows that the maximum bending stress at the fillet region is 420.916 N/mm^2 and this value is less than the magnitude of stress occurred in symmetric gear under same loading conditions.

5. ASYMMETRIC GEAR WITH STRESS RELIEVING FEATURES

A circular hole has been introduce on the face of the asymmetric gear tooth as a modification in the design of asymmetric gear tooth for reduction of bending stress at the fillet region of the gear tooth. A hole at the vicinity of the fillet region reduces the stress concentration created at the fillet. The theoretical explanation behind this idea of modification is that, any abrupt change in the dimension of any structure always creates stress concentration. Now the matter of research is to fix up the shape, dimensions and position of hole with respect to the gear parameter.

3-dimensional model of asymmetric gear tooth with circular hole has been created in Pro/Engineer Wildfire 5.0 parametrically. Dimension of hole has been manipulated by parameter 'HR' which represents hole's radius and position has been manipulated by two parameters 'HPR' & 'THETA' which represent hole's pitch-circle radius and angular position of hole with respect to x-axis. For the first trial p_1 value has been taken 0.5 and p_2 has been considered as 0.44. Angular position of hole with respect to x-axis has been considered as 7.86°.

$$HPR = D_p \times p_2 \qquad HR = r \times p_1$$

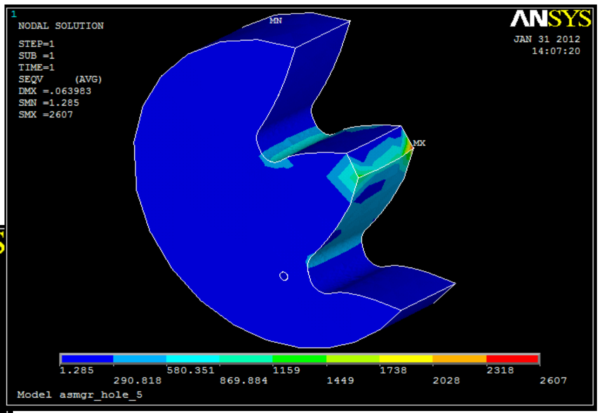


Fig 9: Contour plotting of stress on whole the gear tooth and near hole.

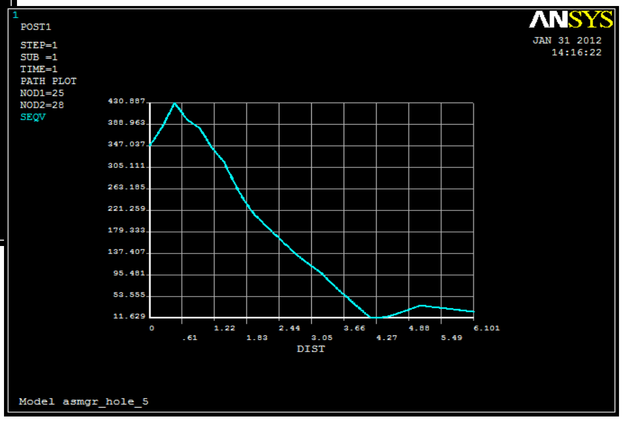


Fig 10: Graph of Von-Misses Stress Distribution at fillet area of asymmetric gear with hole at first trial.

From the above figure it is clear that maximum bending stress created at fillet region is not less than the value calculated for asymmetric gear without hole. The maximum bending stress calculated for asymmetric gear without hole was calculated 420.916N/mm^2 and the value calculated the first trial with hole is 430.007 N/mm^2. So, a second trial has been given. For this trial p_1 value has been taken 0.5 and p_2 has been considered as 0.44. Angular position of hole with respect to x-axis has been considered as 6°.

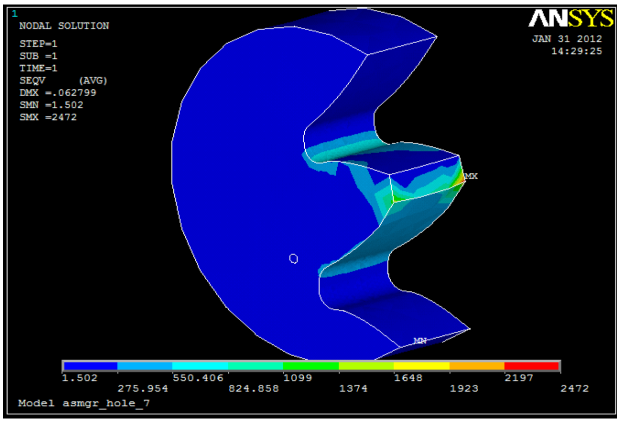


Fig 11: Contour plotting of stress on whole the gear tooth and near hole

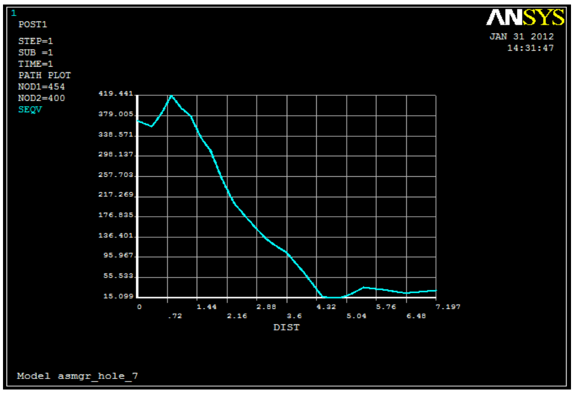


Fig 12: Graph of Von-Misses Stress Distribution at fillet area of asymmetric gear with hole at second trial.

From the above figure it is clear at the present configuration and position of hole described in second trial, maximum bending stress occurs at fillet of the asymmetric gear is 419.441N/mm^2 ,which is less than the maximum bending stress value calculated for asymmetric gear tooth without any hole.

From the above mentioned work it is clear that if a stress relieving feature, like a hole, of suitable shape and dimensions can be put at correct position, bending stress of an asymmetric gear can be reduced further. Another fact has been revealed from the above work is that, the hole should be at vicinity of fillet region and also it should not be very near to the fillet because removal of material at very near to the fillet may make the region breakage prone.

Also it is the fact understood from this work that a hole away from fillet region would not be able to reduce the maximum bending stress at the fillet region. So, it is need to put the hole suitably at correct position and with correct dimensions.

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