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Analysis of an Asymmetric Spur Gear with an Elliptical Hole as Stress Relieving Feature Using ANSYS APDL as Tool

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***Abstract*— Gears are essential parts in any machine. It is used to transmit torque from one point of application to another point of application of a mechanical system through shaft. The shaft which is mainly used for transmitting mechanical power from one point to another point in a machine may be needed to rotate in a single direction only. In that case asymmetric gears of comparatively smaller size may be used in place of symmetric gears. Size of asymmetric gears may be reduced further by introducing stress relieving features. In the present work asymmetric gear with elliptical hole as stress relieving feature has been analysed using Finite Element Method. A FEA software ANSYS has been used for this purpose.**
***Keywords*— Asymmetric gear, Stress relieving feature, Circular hole, Elliptical hole, Von-Misses stress.**

I. INTRODUCTION

In designing aviation vehicle, civil or defense, vehicle weight is a prime aspect to consider first. Specially, when a war plane or a helicopter is being designed weight is kept as less as possible. It is because; war plane or helicopter is needed to change their dynamic state very flexibly and spontaneously and with very high speed. So, it is needed to keep momentum as small as possible but without compromising the magnitude of velocity. To keep overall weight of the plane or helicopter, many methods have been adopted. To reduce weight of whole the plane weight of various sub-systems of the whole flying system are to be reduced. Weight of fuselage can be reduced by using composite material instead of using an isotropic material. Weight of other sub-systems like, 'Transmission System', 'Engine' etc can also be reduced by doing a research on material and designing. Here in this work a research has been done on the designing of gears in transmission system to reduce its size without reducing the torque transmission capability.

Russia started very early doing research on the above mentioned topic. So many researches were done by many Russian scientist and engineers in 1970s and 80s. They started with a concept of asymmetric gear design to be used in transmission system of aircraft. E. B. Vulgakov [1] published a paper on improvement of the characteristics of gear design in the year of 1974. Then in the year of 1984 I. A. Bolotovsky, O. F. Vasil'eva and V. P.Kotelnikov[2] did a work on asymmetric involute gear and published a paper. Work of A. L. Kapelevich[3] on asymmetric is also very remarkable which he did and published in the year of 1987. There is a fabulous work in gear manufacturing done by Mabie H. H., Rogers C. A., and Reinholtz C. F [4] and they published it in the year of 1990. In the year of 1993 N. Canesan and S. Vijayarangan [5] did an analysis on composite spur-gear with three-dimensional finite element method. Another beautiful work on analysis of asymmetric gear tooth has been done by G.DiFrancesco and S.Marini[6] in the year of 1997 and was published in 'Gear Technology'. Kapelevich A.L. [7] did a remarkable work on involute spar gear with asymmetric tooth profile in the year of 2000 and published it in the renowned journal named "Mechanism and Machine Theory". In the same year another important work was done on the process of noise reduction of asymmetric gear drive during gear meshing. That was done by Litvin, F.L., Q. Lian and A.L. Kapelevich[8] and they published their work as a research paper in the journal "Computer Methods in Applied Mechanics and Engineering". In the year of 2001 so many works were done on the analysis of asymmetric gear. G. Gang and T. Nakanishi [9] did a work on "Enhancement of Bending load Carrying Capacity of Gears Using an Asymmetric Involute Tooth," and published their work in "The JSME International Conference on Motion and Transmissions (MPT2001-Fukuoka), Fukuoka, JAPAN". Yeh,T.,Yang and D. Tong [10] did a work on "Design of new tooth profiles for high-load capacity gears" which was published in the journal "Mechanism and Machine Theory". Another work was done in that same year that is in 2001 on Tooth contact analysis of an asymmetric gear jointly by US Army Research Laboratory and NASA [11]. The document number of that research work is NASA/TM-2001-210614, ARL-TR-2373. In next two years that is in 2002 and 2003 few works were done on direct gear design method to design spar and helical gear. A.L. Kapelevich and R.E. Kleiss [12] did a work "Direct Gear Design for Spur and

Helical Gears" and published in journal Gear Technology (September/October 2003, 29-35). Kapelevich A.L. and Shekhtman Y.V. [13] did a work on "Direct Gear Design: Bending Stress Minimization" and that was published in Gear Technology, September/October 2003, 44-49. In the year of 2005 K. Cavdar, F. Karpat and F.C. Babalik [14] performed a fabulous work on "Computer aided analysis of bending strength of involute spur gears with asymmetric profile" and published their work in Journal of Mechanical Design 127 (3) (2005) 477-484. In the year of 2007 and 2008 many works on asymmetric gear were done. Flavia CHIRA, Vasile Tisan and Anamaria Dascalescu [15] did a work on "Modeling of the Asymmetric Gears using Applications in Matlab and Autolisp" and published in ANNALS Of THE ORADEA UNIVERSITY, Fascicle of Management and Technological Engineering, Volume VI (XVI), 2007. Kapelevich A.L. [16] did a work on "Direct Design Approach for High Performance Gear Transmissions" and was published in Gear Solutions, January 2008, 22-31. This article was presented at the Global Powertrain Congress 2007 June 17-19, 2007, Berlin, Germany and published in the Global Powertrain Congress Proceedings, Vol. 39-42, 66-71. A.S. Novikov, A.G. Paikin, V.L. Dorofeyev,V.M. Ananiev and A.L. Kapelevich [18] did a work on "Application of Gears with Asymmetric Teeth in Turboprop Engine Gearbox" and published his work in Gear Technology, January/February, 2008. A very recent work of Mr. Sumit Agrawal and Dr. R. L. Himte on " Evaluation of Bending Stress at Fillet Region of an Asymmetric Gear with a Hole as Stress Relieving Feature using a FEA Software ANSYS" has been referred in this work as a base paper. In their work they have studied effect of stress reliving feature on asymmetric gear. They considered circular hole as stress reliving feature and in the present work elliptical hole has been considered as stress reliving feature. Besides above mentioned references in this project many research papers and books and documentations have been referred. All those papers and books and documentations have been mentioned in the chapter 'REFERENCE'. Among all those references work of Frederick W. Brown, Scott R. Davidson, David B. Hanes and Dale J. Weires have been considered as base paper here. In their work Frederick W. Brown et al tested and analyzed an involute gear tooth with asymmetric profile practically and with a FEA method (other than ANSYS). In the present work research of Frederick W. Brown et al has been reproduced using ANSYS and a modification of asymmetric gear tooth design has been adopted and validated with help of FEA software ANSYS.

II. PARAMETRIC MODELING OF AN ASYMMETRIC INVOLUTE GEAR

From reference [17] all the required parameters as shown in table below have been collected to model the gear parametrically.

Table 1: Parameters of the Asymmetric Gear from Ref. [17]

Parameters	Symmetric Toothed Gear	Asymmetric toothed Gear
Number of Tooth (N)	32	32
Diametral Pitch (p)	0.21	0.21
Drive Pressure Angle (ϕd)	25°	35°
Coast Pressure Angle (ϕc)	25°	15°
Pitch Diameter (Dp)	152.4mm	152.4mm
Drive Base Diameter (Dbd)	138.12mm	124.84mm
Coast Base Diameter	138.12mm	147.21mm
Outside Diameter	162.56mm	162.21mm
Root Diameter	141.5034mm	141.1732mm
Fillet Radius	1.8796mm	1.9812mm
Face width	9.525mm	9.525mm
Torque	564923.94N-mm	564923.94N-mm
Load Application Radius	81.28mm	81.28mm

In reference with above mentioned parameters first a portion of involute gear with symmetric profile has been created in a 3-D modeling software named PTC Creo 2.0. To create the 3-D model parametrically above mentioned parameters (as mentioned in table1) have been introduced or put into the software.

To generate involute profile in Pro/E here, only 'Number of Teeth (N)', 'Diametral Pitch (P)' and 'Pressure angle (PHI)' have been considered as input parameters and other parameters like:

Pitch Diameter (DP)

Addendum (A)

Duodenum (B)

Addendum circle diameter (DA)

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Root circle diameter (DR)
Base circle diameter (DB)
Fillet radius (R)
Face width (F)

have been calculated from equations of gear design mentioned below and those equations have been mentioned in the Creo editor which has been shown in figure below.

Pitch Circle Diameter (DP) = Number of Teeth (N)/ Diametral Pitch (P)

Addendum (A) = 1/Diametral Pitch (P)

Duodenum (B) = 1.157/ Diametral Pitch (P)

Addendum Circle Dia (Da) = DP + 2*A

Duodenum Circle Dia or Root Circle Dia (Dr) = DP – 2*B

Base Circle Dia (Db) = DP * Cos(PHI)

Fillet radius (r) = 0.4 *A

Face Width (F) = 0.0625*DP

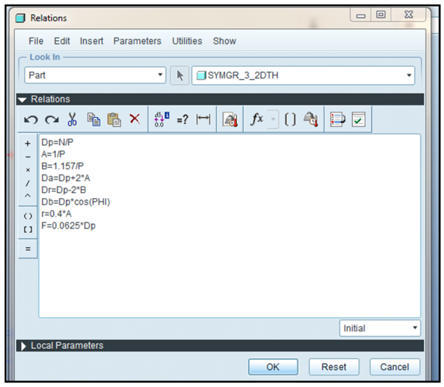

Fig 1: Gear design equations have been mentioned in Pro/E.

After mentioning gear design equations an involute curve has been created from datum curve generation tool using involute curve equation in cylindrical co-ordinate system as mentioned below.

Mathematical model of an asymmetric involute gear tooth has been validated as per the work of Frederick W. Brown et al (Reference [17]). To validate work of Frederick W. Brown et al[17] an asymmetric gear tooth has been created in PTC Creo 2.0 software using parameters as mentioned in table1 and using asymmetric involute curve equations as mentioned below.

For Drive direction

todeg=180/pi

alpha=t*sqrt((Da/Dbd)^2-1)

alpha2=sqrt((Dp/Dbd)^2-1)

r=0.5*Dbd*sqrt(1+alpha^2)

theta=alpha*todeg-atan(alpha)-(alpha2*todeg-atan(alpha2))-(90/n)-1

z=0

For Coast direction

todeg=180/pi

alpha= t*sqrt((Da/Dbc)^2-1)

alpha2=sqrt((Dp/Dbc)^2-1)

r=0.5*Dbc*sqrt(1+alpha^2)

theta=alpha*todeg-atan(alpha)-(alpha2*todeg-atan(alpha2))-(90/n)+.4

z=0

Using the above equations involute curve for asymmetric gear has been generated in Pro-Engineer software and using the involute curves thus generated, a profile of involute asymmetric gear with one gear tooth has been generated. Finally a 3-D model of asymmetric gear tooth has been created using the profile as mentioned above.

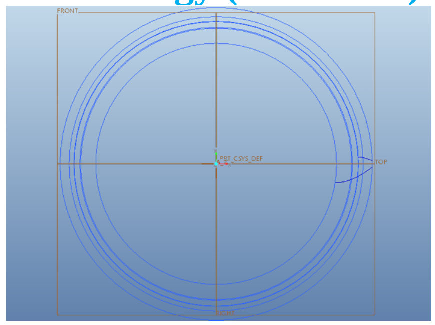

Fig 2: Involute curves for asymmetric gear.

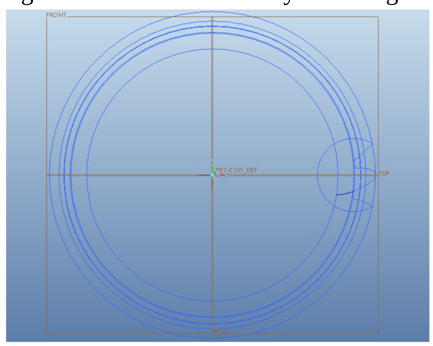

Fig 3: Partial profile of asymmetric involute gear tooth.

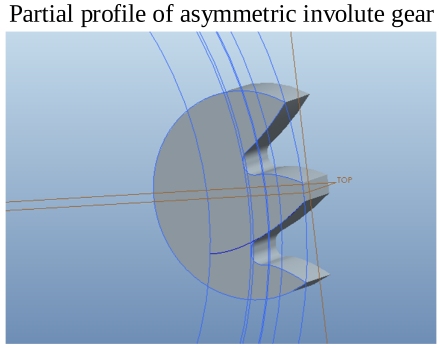

Fig 4: 3-dimensional model of an asymmetric gear tooth.

III. STRUCTURAL ANALYSIS OF THE ASYMMETRIC GEAR UNDER A GIVEN LOADING CONDITION

3-dimensional model of partial gear is created in Creo 2.0 and was imported in ANSYS for structural simulation to find out maximum bending stress at fillet region. To do this loading conditions have been considered as per Frederick W. Brown et al (Reference [17]. In ANSYS the asymmetric tooth has been meshed with 10-node tetrahedral element named SOLID92. To mesh a model there are many schemes available in ANSYS. But for any irregular body or asymmetric object meshing is usually done by the default scheme already put in the software. Here the asymmetric involute gear tooth has been meshed using the default scheme already present in the software.

As per Frederick W. Brown et. al. (Reference [17]) torque applied to the gear tooth is 564923.94N-mm and it has been applied at the end of the gear tooth. (i.e torque radius is 81.28mm). From the given torque and torque application radius, longitudinal force calculated is 6950.334 N which has been applied at tooth top at modified pressure angle. In reality load is actually exerted on a line of contact passing through a point near pitch circle. But it is not possible as the mashing is unstructured and so a series of nodes cannot be available long a line near pitch circle. To avoid this problem load in the simulation is imposed at tip of the gear model with a modified pressure angle. To make the above consideration or assumption effective following equation has been implemented or used. The equation is:

$$\emptyset_m = \emptyset - \frac{s_a}{2r_a} \qquad (1)$$

Where-

ϕm is modified pressure angle.

ϕ is actual pressure angle.

sa is tooth thickness at addendum circle.

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ra is addendum circle radius.

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

After importing the 3-D gear model in ANSYS software it has been meshed in finite elements. Figure below shows the meshed view of gear tooth.

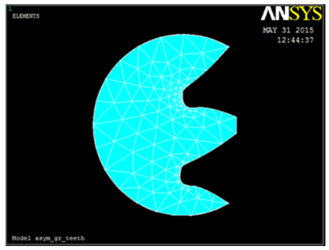

Fig 5: Meshed view of gear tooth in 2-D orientation.

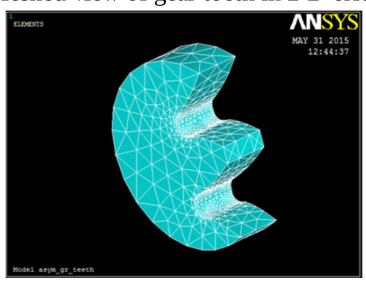

Fig 6: Meshed view of gear tooth in 3-D orientation.

On completion of meshing the gear tooth has been imposed with the load in the same way as mentioned before. After imposing loads and boundary conditions the model has been solved in ANSYS with its inbuilt solver. After solution deflection of gear tooth and Von-Misses stress have been found out. Figure below shows the deflection of gear tooth under a given loading condition.

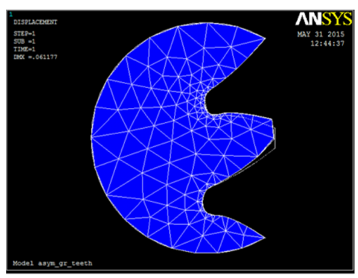

Fig 7: Deformation of gear tooth of the asymmetric gear under load

Above figure clearly depicts that maximum deflection occurs at the gear tooth's extreme edge and its value is 0.061177mm and this value is less than the magnitude of deflection occurred in symmetric gear under same loading conditions. After calculation of deflection Von-Misses stress has been derived and shown as contour plotting over all the portion of the gear tooth. Figure below is the contour plotting of Von-Misses stress on the gear tooth.

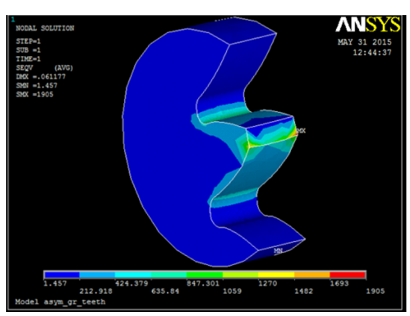

Fig 9: Von-Misses Stress Distribution of asymmetric gear.

From the above figure it is clear that maximum stress occurs at the tip of the gear tooth. But here bending stress is needed to be calculated. To calculate bending stress at the fillet region a graph between Von-Misses stress of each point at the edge of fillet region and their distance from a reference has been plotted. Figure below shows the graph.

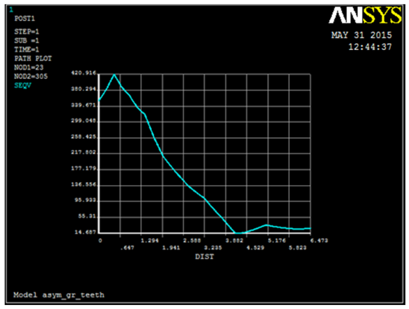

Fig 10: Graph of Von-Misses Stress Distribution at fillet area of symmetric gear.

The above graph shows that the maximum bending stress at the fillet region is 420.916 N/mm2 which agrees the result found by Frederick W. Brown et al (Reference [17]) in their work. So, it can be said mathematical model of asymmetric gear tooth with involute profile has been validated for further FEA analysis.

IV. STRUCTURAL ANALYSIS OF THE ASYMMETRIC GEAR WITH CIRCULAR STRESS RELIVING FEATURE

In this section a design modification which has been adopted by S. Agrawal et. al. [24] has been discussed. In their work S Agrawal et. al have introduced a circular hole on the face of the asymmetric gear in the vicinity of the fillet area and have found an beneficial position and size of the hole after few trials. Below is the FEA analysis of asymmetric gear with circular hole as mentioned in Reff [24]. S. Agrawal et al [24] has controlled the dimension and position of the circular hole in their work through few parameters like 'HR' representing radius of hole, 'HPR' pitch circle radius of hole-center and 'THETA' representing angular position of hole with respect to the axis of gear.

Here hole's radius 'HR' and hole's pitch circle radius 'HPR' have been parameterized with radius if fillet of gear tooth 'r' and diameter of pitch circle of gear tooth 'DP' respectively like following. THETA is different for different trials.

$HR = r \times p_1$

$$HPR = D_p \times p_2 \qquad (5.2)$$

Where, p1 and p2 are parameters control the configuration and position of hole.

In the final trial of the work of S agrawl et al [24] p1 value has been taken 0.5 and p2 has been considered as 0.445. Using these parameters a three dimensional model of asymmetric gear with a hole has been created in the same way as mentioned for trial one above. Figures below show two view of three dimensional model of asymmetric gear tooth with hole created in second trial.

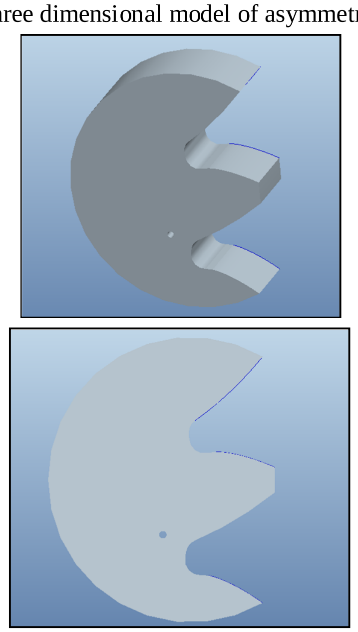

Fig 11: Isometric view and Front view of 3-D asymmetric gear with hole in 3rd trial

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After generating three dimensional model of asymmetric gear with a hole for third trial position and size in PTC Creo 2.0 (as show in figure above), it has been imported in ANSYS for meshing. After doing Meshing and imposing same boundary conditions and loading conditions following result are derived. Figures below show the contour plotting for Von-Misses stress on whole the gear tooth and stress around the hole.

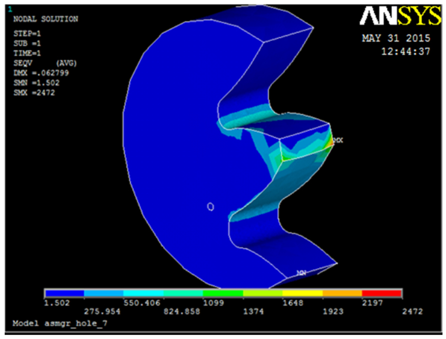

Fig 12: Contour plotting of stress on whole the gear tooth and near hole.

From the above figure it is clearly depicted that maximum stress occurs at tooth tip but bending stress occurs at the fillet area. The software calculates all the stresses but on contour plot it only shows the magnitude of maximum stress. 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. Figure below shows the graph of Von-Misses stress at different point on the edge of fillet at section of gear tooth versus distances of those points from a reference.

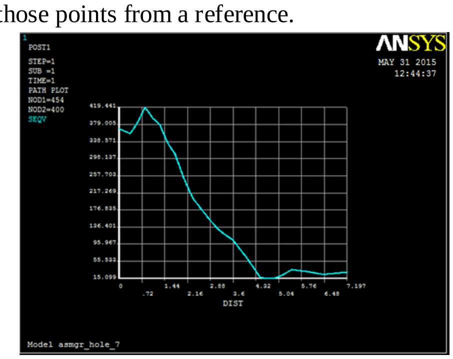

Fig 13: Graph of Von-Misses Stress Distribution at fillet area of asymmetric gear with hole at third trial.

From the above analysis done in this chapter it has been shown that result for maximum bending stress occurred at the fillet region of the gear when circular hole as stress reliving feature is not considered is well agreed with the result found by Frederick W. Brown, Scott R. Davidson, David B. Hanes and Dale J. Weires in their work (Ref. [17]). Whatever discrepancies have been raised between the result of this work and work of Frederick W. Brown et al [17] is due to the fact that Frederick W. Brown et al considered 2-Dimensional model of gear in their analysis and in the present work 3-Dimensional model has been created and analyzed. After validating the work of Frederick et al [17], work of S. Agrawal et al [24] has also been validated considering circular hole as a stress reliving feature.

V. ASYMMETRIC GEAR WITH ELLIPTICAL STRESS RELIEVING FEATURES

In the previous section it has been shown that S Agrawal et al [24] considered a circular hole to reduce Von-Misses stress at the fillet region of an asymmetric gear. As they have only tried with circular hole here in the present work an effort has been made to investigate the influence of an elliptical hole of different sizes and orientations on the Von-Misses stress at fillet region of the gear. In this work shape of the hole has been considered as elliptical. The dimension and position of the elliptical hole has been found out by trial and error method using FEA software ANSYS for which bending stress has been decreased further. 3-dimensional model of asymmetric gear tooth with elliptical hole has been created in PTC Creo Parametric software parametrically. Dimension of hole has been manipulated by parameters ‘HMJD’ and ‘HMND’ which represent elliptical hole’s major dia and minor dia. Position has been manipulated by two parameters ‘HPD’ & ‘HPANG’ which represent hole’s pitch-circle diameter and angular position of hole with respect to x-axis. Another parameter named ‘HAANG’ has been introduced to specify the angle between the axis of hole-center and hole’s major axis.

Here hole’s major dia ‘HMJD’, minor dia ‘HMND’ and hole’s pitch circle Dia ‘HPD’ have been parameterized with radius if fillet of gear tooth ‘FR’ and diameter of pitch circle of gear tooth ‘Dp’ respectively like following. ‘HPANG’ and ‘HAANG’ are different for different trials.

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$$HPD = D_p \times par_1$$

$$HMJD = FR \times par_2$$

$$HMND = FR \times par_3$$

Where, par1, par2 and par3 are parameters control the configuration and position of hole. Parameter 'par1' defines radial distance of hole-center from the center of gear. Parameter 'par2' and 'par3' define major axis and minor axis of elliptical hole.

For the first trial 'par1' value has been taken 0.85, 'par2' and 'par3' have been considered as 0.8 and 0.4 respectively. These parameters have been added to the design equation of asymmetric gear of involute profile as mentioned in chapter three. After appending the equations mentioned in equ (6.1) the equation editor of PTC Creo Parametric software will look like following figure.

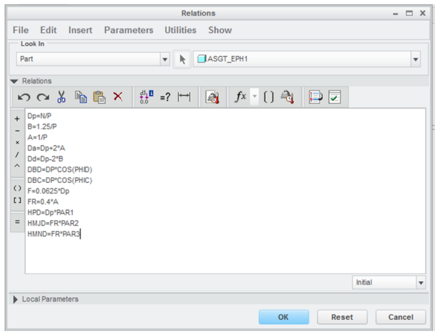

Fig 14: Equation editor in Creo with parametric equations for asymmetric gear with hole in First trial.

After executing above equations radius hole's PCD becomes129.5283 mm. Major dia and Minor dia becomes 1.52 mm and 0.76 mm respectively. Angular position of hole with respect to x-axis has been considered as 7° and axis of hole with hole-center axis has been considered as 90º. All the parameters which have been taken in first trial are very much random and as a part of the trial and error method. After generation of all the required parameters first asymmetric gear tooth has been created by the same process as mentioned before Now sketch of the hole has been created on the face of the asymmetric gear with hole's dimension parameters and hole positioning parameters. Lastly the hole has been done through the face of the asymmetric gear tooth axially. Thus an asymmetric gear tooth of involute profile with a hole as a stress reliving feature has been modeled for first trial simulation in ANSYS. Figures below show two view of three dimensional model of asymmetric gear tooth with hole created in first trial.

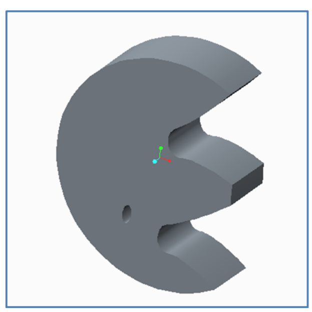

Fig 15: Isometric view of 3-D asymmetric gear with hole in 1st trial.

As the completion of 3-D model creation is done, the model has been imported in ANSYS for trial analysis to find out bending stress at fillet region. Meshing has been done with 10-nodes tetrahedral element named 'SOLID 92' as mentioned in before. Figure below shows the meshed view of asymmetric gear tooth with hole for first trial.

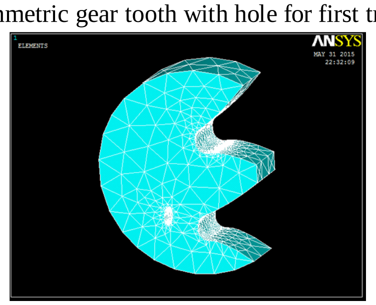

Fig 16: Isometric view of Meshed 3-D asymmetric gear tooth with hole in 1st trial.

To show the meshing at the vicinity of hole of the gear tooth a figure has been presented below the enlarged front view of

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meshed gear tooth clearly showing the mesh near the hole region.

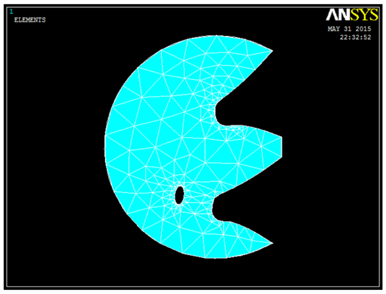

Fig 17: Enlarged front view of meshed gear tooth clearly showing the region near hole.

On completion of meshing of the asymmetric gear tooth with hole in first trial load has been imposed on gear tooth. Load has been distributed on each node of the extreme edge of the gear tooth. The value of loads on each node and method of loading is same as mentioned in chapter 5 for asymmetric gear without hole. After imposing loads on each node boundary conditions have been fixed in the same way as mentioned before. The curved part of the gear tooth has been held fixed by assigning all degree of freedom to zero. Now FEA model of asymmetric gear tooth with first trial hole has been solved in ANSYS. After solution bending stress has been derived from post-processing. Figures below shows the contour plotting for Von-Misses stress on whole the gear tooth and stress around the hole.

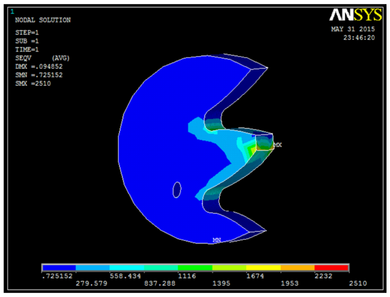

Fig 18: Contour plotting of stress on whole the gear tooth and near hole.

From the above figure it is clearly depicted that maximum stress occurs at tooth tip but bending stress occurs at the fillet area. The software calculates all the stresses but on contour plot it only shows the magnitude of maximum stress. Maximum bending stress at the fillet region has been determined by plotting the Von-Misses stress value at every node on the edge curve at the undercut region. The graph which has been generated by plotting the stress value of different nodes with respect to its distance from any reference point along the curve has been presented below.

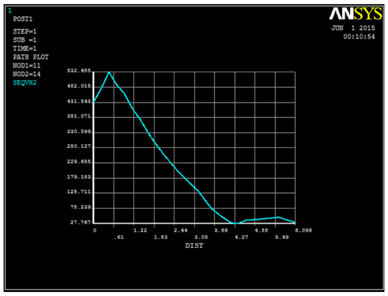

Fig 19: Graph of Von-Misses Stress Distribution at fillet area of asymmetric gear with elliptical hole for its first position.

From the above graph it is quite clear that the maximum stress value that is 532.458 N/mm2 is not a better value calculated by S. Agrawal et al in their work for the fillet region. The maximum bending stress calculated by S. Agrawal et al [24] is 419.441N/mm2 and the value calculated the first trial with hole is 532.485 N/mm2. So, a second trial has been given.

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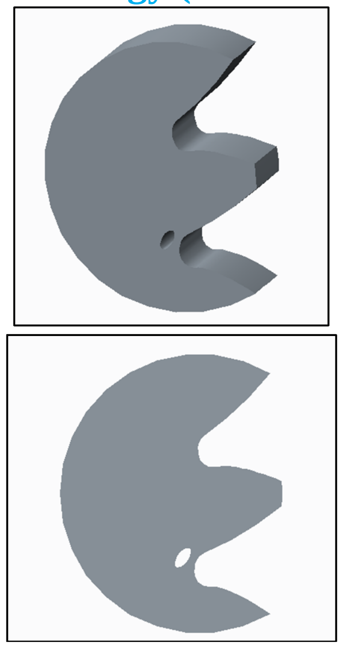

Fig 20: Isometric view and Front view of 3-D asymmetric gear with hole in 3rd trial.

After imposing loads on each node boundary conditions have been fixed in the same way as mentioned in chapter 5. The curved part of the gear tooth has been held fixed by assigning all degree of freedom to zero. Now FEA model of asymmetric gear tooth with second trial hole has been solved in ANSYS. After solution bending stress has been derived from post-processing. Figures below shows the contour plotting for Von-Misses stress on whole the gear tooth and stress around the hole.

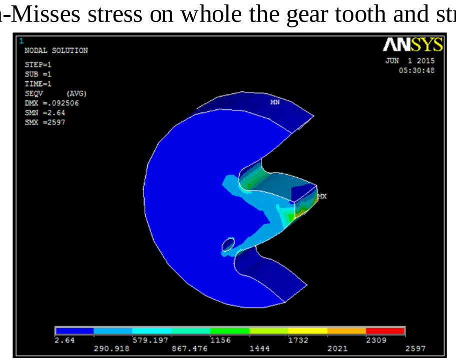

Fig 21: Contour plotting of stress on whole the gear tooth and near hole.

From the above figure it is clearly depicted that maximum stress occurs at tooth tip but bending stress occurs at the fillet area. The software calculates all the stresses but on contour plot it only shows the magnitude of maximum stress. 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. Figure below shows the graph of Von-Misses stress at different point on the edge of fillet at section of gear tooth versus distances of those points from a reference.

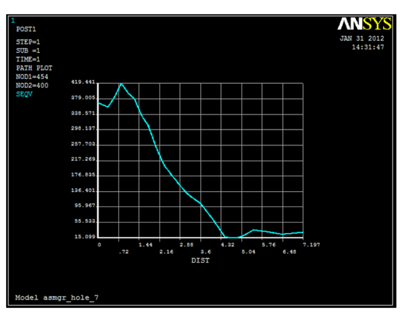

Fig 22: Graph of Von-Misses Stress Distribution at fillet area of asymmetric gear with elliptical hole for its third position

Above figure depicts the fact that maximum Von-Misses stress at fillet region when the elliptical hole as mentioned in third modification is incorporated gets increased that that derived in second modification.

VI. RESULT AND DISCUSSION

From the above study it is quite clear that though it has not been able to reduce the stress that the value determined by S. Agrawal et al [24] but it has been experienced from a series of simulation that the bending stress first decrease from trial 1 to trial 2 and then again increases from trial 2 to trial 3. So, there must be some region where more less stress may exist.

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