Thesis Report

On

SPUR GEAR TOOTH STRESS ANALYSIS AND STRESS REDUCTION USING STRESS REDUCING GEOMETRICAL FEATURES

Submitted in the partial fulfillment of requirement for the award of the degree of

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IN

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Submitted by

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CERTIFICATE

This is to certify that the thesis report entitled **SPUR GEAR TOOTH STRESS ANALYSIS AND STRESS REDUCTION USING STRESS REDUCING GEOMETRICAL FEATURES** submitted by **Mr. MANOJ HARIHARAN** in the partial fulfillment of the requirement for the award of the degree of **Master of Engineering in CAD/CAM AND ROBOTICS** to **Thapar Institute of Engineering and Technology (Deemed University), Patiala,** is a record of candidate's own work carried out by him under my supervision and guidance. The matter embodied in this report has not been submitted in part or full to any other university or institute for the award of any degree.

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ABSTRACT

Gears have a wide variety of applications. Their applications vary from watches to very large mechanical units like the lifting devices and automotives. Gears generally fail when the working stress exceeds the maximum permissible stress. Number of studies has been done by various authors to analyse the gear for stresses. Gears have been analysed for different points of contact on the tooth profile and the corresponding points of contact on the pinion.

In this study the technology of gears is presented along with the various types of failures that gears have. The causes of these failures are studied and one type of stress related failure due to fatigue failure of a gear tooth due to stress concentration is detailed. Various studies have been done for variation of moment. Studies have also been done to find the highest point single tooth contact. The point after being found is point of application of force. Then the variation stress in root fillet region is found, which is then used for the study of variation of various parameters of stress reducing features. The effect and use of stress relief feature in geometry of gear is studied as reported by researchers.

A study of the optimum size and location of the stress relief features for pinion and gear is proposed which help in reducing the fatigue failure in gears.

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CHAPTER-1

INTRODUCTION

1.1 GEAR TECHNOLOGY

Gears are the most common means of transmitting power in the modern mechanical engineering world. They vary from a tiny size used in watches to the large gears used in watches to the large gears used in lifting mechanisms and speed reducers. They form vital elements of main and ancillary mechanisms in many machines such as automobiles, tractors, metal cutting machine tools etc. Toothed gears are used to change the speed and power ratio as well as direction between input and output.

Law of Gearing

A primary requirement of gears is the constancy of angular velocities or proportionality of position transmission. Precision instruments require positioning fidelity. High-speed and/or high-power gear trains also require transmission at constant angular velocities in order to avoid severe dynamic problems. Constant velocity (i.e., constant ratio) motion transmission is defined as "conjugate action" of the gear tooth profiles. A geometric relationship can be derived for the form of the tooth profiles to provide conjugate action, which is summarized as the Law of Gearing as follows:

"A common normal to the tooth profiles at their point of contact must, in all positions of the contacting teeth, pass through a fixed point on the line-of-centers called the pitch point."

Any two curves or profiles engaging each other and satisfying the law of gearing are conjugate curves.

Fig 1.1 shows two mating gear teeth, in which

- Tooth profile 1 drives tooth profile 2 by acting at the instantaneous contact point K.
- N_1N_2 is the common normal of the two profiles.
- N_1 is the foot of the perpendicular from O_1 to N_1N_2
- N_2 is the foot of the perpendicular from O_2 to N_1N_2.

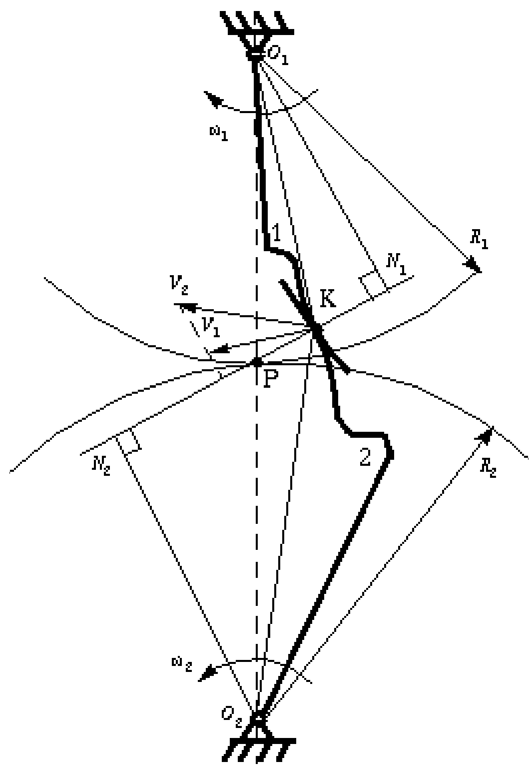


Fig 1.1 Law of Gearing

Although the two profiles have different velocities V_1 and V_2 at point K, their velocities along N_1N_2 are equal in both magnitude and direction. Otherwise the two tooth profiles would separate from each other. Therefore,

$$O_1N_1 \cdot \omega_1 = O_2N_2 \cdot \omega_2$$

or

$$\frac{\omega_1}{\omega_2} = \frac{O_2N_2}{O_1N_1}$$

It is noticed that the intersection of the tangency N_1N_2 and the line of center O_1O_2 is at point P, and

$$\Delta O_1 N_1 P \sim \Delta O_2 N_2 P$$

Thus, the relationship between the angular velocities of the driving gear to the driven gear, or velocity ratio, of a pair of mating teeth is

$$\frac{\omega_1}{\omega_2} = \frac{O_2 P}{O_1 P}$$

Point P is very important to the velocity ratio, and it is called the pitch point. Pitch point divides the line between the line of centers and its position decides the velocity ratio of the two teeth. The above expression is the fundamental law of gear-tooth action.

1.2 Conjugate Action

Mating gear teeth acting against each other to produce rotary motion are similar to cams. When the tooth profiles, or cams, are designed so as to produce a constant angular-velocity ratio during meshing, these are said to have conjugate action. In theory, at least, it is possible arbitrarily to select any profile for one tooth and then to find a profile for the meshing tooth which will give conjugate action. One of these solutions is the involute profile, which, with few exceptions, is in universal use for gear teeth.

There are two forms of tooth profile commonly used:-

a) Cycloidal teeth
b) Involute teeth

An advantage of the cycloidal teeth over the involute one is that wear of cycloidal tooth is not as fast as with involute tooth. For this reason, gears transmitting very large amount of power are sometimes cut with cycloidal teeth. On the other hand, involute teeth are very easy to manufacture and the actual distance between the centers may deviate slightly from the theoretical distance without affecting the velocity ratio or general performance. Because of this distinct advantage, gears with involute cut teeth are used much more than those with cycloidal teeth.

Involute properties

There are almost an infinite number of curves that can be developed to satisfy the law of gearing, and many different curve forms have been tried in the past. Modern gearing (except for clock gears) is based on involute teeth. This is due to three major advantages of the involute curve:

5 Conjugate action is independent of changes in center distance.
6 The form of the basic rack tooth is straight-sided, and therefore is relatively simple and can be accurately made; as a generating tool it imparts high accuracy to the cut gear tooth.
7 One cutter can generate all gear teeth numbers of the same pitch.

An involute may be generated as shown in fig. 1.2 a partial flange B is attached to the cylinder A, around which is wrapped a cord def which is held tight. Point b on the cord represents the tracing point, and as the cord is wrapped and unwrapped about the cylinder, point b will trace out the involute curve ac.

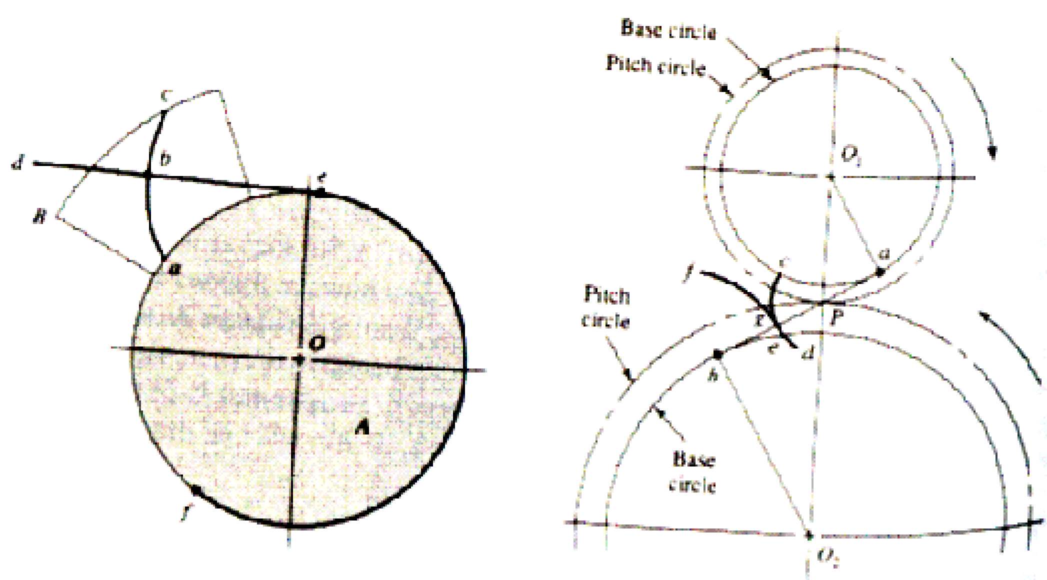


Figure 1.2 Generation of an involute and involute action

The radius of the curvature of the involute varies continuously, being zero at point ‘a’ and a maximum at point c. at point b the radius is equal to the distance be, since point “be” is instantaneously rotating about point e. Thus generating line “de” is normal to the involute at all point of intersection and at the same time, is always tangent to the cylinder A. The circle on which the involute is generated is called the base circle.

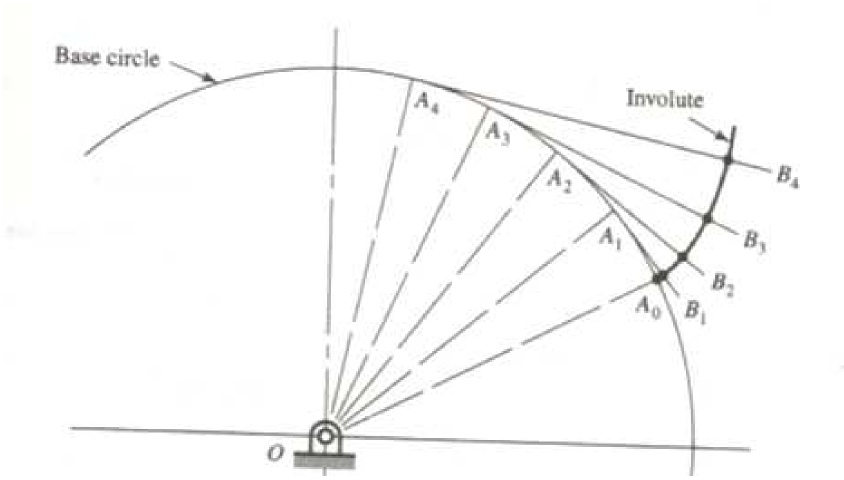


Figure 1.3 Construction of an Involute curve

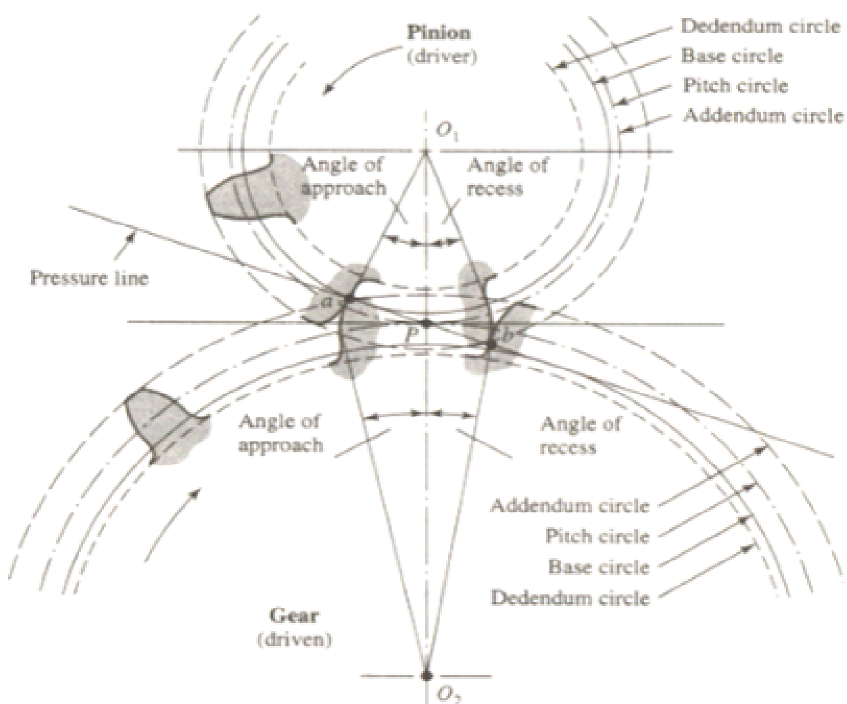


Figure 1.4 Tooth Action

The design of the gear is usually done for the permissible bending stress on which basis the face width of the gear is calculated. The value of the bending stress is taken equal to the elastic limit of the material in bending divided by a factor of safety. The value of the factor of safety varies from 1.5 to 3.0 and depends on the type of application.

For calculating the face width of the gear, the following formula is considered.

$$F = W_t / K_v * m * Y * \sigma_b$$

Where, F = Face width

W_t = Tangential load

K_v = Velocity factor

m = Module

Y = Lewis form factor

σ_b = Permissible bending stress

1.3 SPUR GEARS

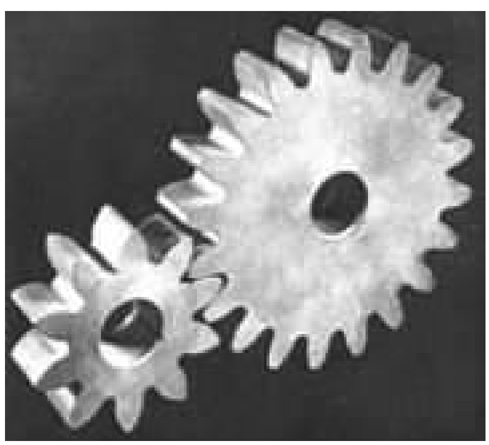


Fig 1.5 Spur gears

Spur gears*: -* This is a cylindrical shaped gear in which the teeth are parallel to the axis. It has the largest applications and, also, it is the easiest to manufacture. Spur gears are the most common type used. Tooth contact is primarily rolling, with sliding occurring during engagement and disengagement. Some noise is normal, but it may become objectionable at high speeds.

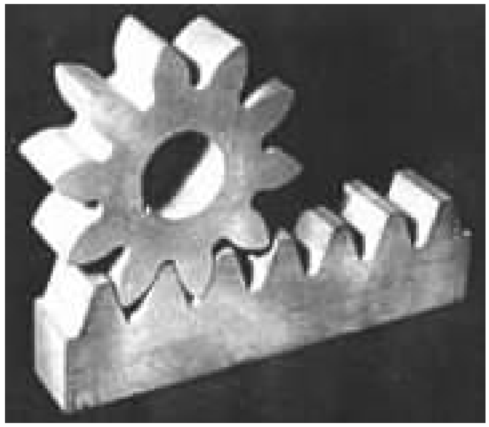


Fig 1.6 Rack and Pinion gears

Rack and pinion*:* - This is a linear shaped gear which can mesh with a spur gear with any number of teeth. The spur rack is a portion of a spur gear with an infinite radius. Rack and pinion gears are essentially a variation of spur gears and have similar lubrication requirements.

1.4 GEAR NOMENCLATURE

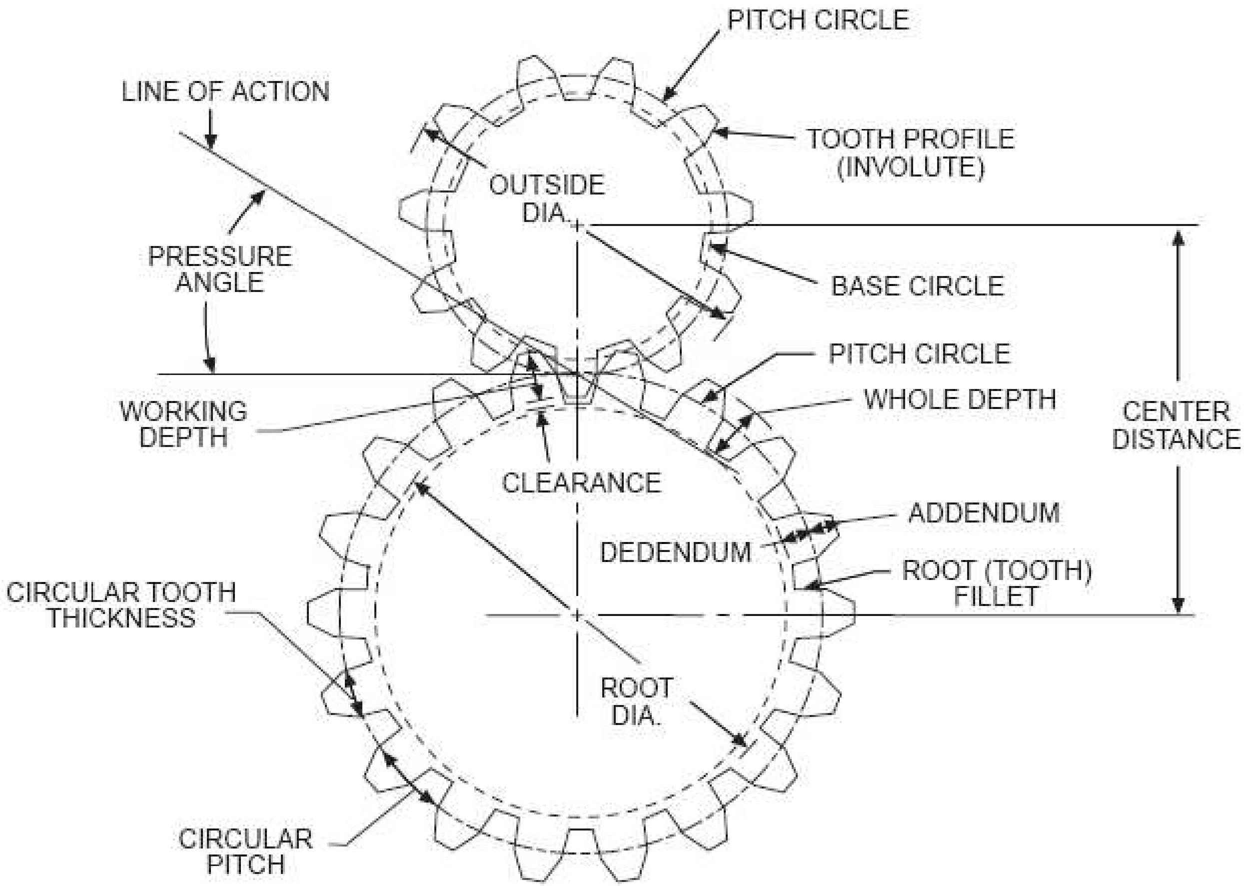


Fig 1.7 Gear Nomenclature

ADDENDUM (a) is the height by which a tooth projects beyond the pitch circle or pitch line.

BASE DIAMETER (D_b) is the diameter of the base cylinder from which the involute portion of a tooth profile is generated.

BACKLASH (B) is the amount by which the width of a tooth space exceeds the thickness of the engaging tooth on the pitch circles. As actually indicated by measuring devices, backlash may be determined variously in the transverse, normal, or axial-planes, and either in the direction of the pitch circles or on the line of action. Such measurements should be corrected to corresponding values on transverse pitch circles for general comparisons.

BORE LENGTH is the total length through a gear, sprocket, or coupling bore.

CIRCULAR PITCH (p) is the distance along the pitch circle or pitch line between corresponding profiles of adjacent teeth.

CIRCULAR THICKNESS (t) is the length of arc between the two sides of a gear tooth on the pitch circle, unless otherwise specified.

CLEARANCE-OPERATING (c) is the amount by which the dedendum in a given gear exceeds the addendum of its mating gear.

CONTACT RATIO (m_c) in general, the number of angular pitches through which a tooth surface rotates from the beginning to the end of contact.

DEDENDUM (b) is the depth of a tooth space below the pitch line. It is normally greater than the addendum of the mating gear to provide clearance.

DIAMETRAL PITCH (P) is the ratio of the number of teeth to the pitch diameter.

FACE WIDTH (F) is the length of the teeth in an axial plane.

FILLET RADIUS (r_f) is the radius of the fillet curve at the base of the gear tooth.

FULL DEPTH TEETH are those in which the working depth equals 2.000 divided by the normal diametrical pitch.

GEAR is a machine part with gear teeth. When two gears run together, the one with the larger number of teeth is called the gear.

HUB DIAMETER is outside diameter of a gear, sprocket or coupling hub.

HUB PROJECTION is the distance the hub extends beyond the gear face.

INVOLUTE TEETH of spur gears, helical gears and worms are those in which the active portion of the profile in the transverse plane is the involute of a circle.

LONG- AND SHORT-ADDENDUM TEETH are those of engaging gears (on a standard designed center distance) one of which has a long addendum and the other has a short addendum.

KEYWAY is the machined groove running the length of the bore. A similar groove is machined in the shaft and a key fits into this opening.

OUTSIDE DIAMETER (D_o) is the diameter of the addendum (outside) circle.

PITCH CIRCLE is the circle derived from a number of teeth and a specified diametrical or circular pitch. Circle on which spacing or tooth profiles is established and from which the tooth proportions are constructed.

PITCH CYLINDER is the cylinder of diameter equal to the pitch circle.

PINION is a machine part with gear teeth. When two gears run together, the one with the smaller number of teeth is called the pinion.

PITCH DIAMETER (D) is the diameter of the pitch circle. In parallel shaft gears, the pitch diameters can be determined directly from the center distance and the number of teeth.

PRESSURE ANGLE (ø) is the angle at a pitch point between the line of pressure which is normal to the tooth surface, and the plane tangent to the pitch surface. In involute teeth, pressure angle is often described also as the angle between the line of action and the line tangent to the pitch circle. Standard pressure angles are established in connection with standard gear-tooth proportions.

ROOT DIAMETER (D_r) is the diameter at the base of the tooth space.

PRESSURE ANGLE-OPERATING (ϕ_r) is determined by the center distance at which the gears operate. It is the pressure angle at the operating pitch diameter.

TIP RELIEF is an arbitrary modification of a tooth profile whereby a small amount of material is removed near the tip of the gear tooth.

UNDERCUT is a condition in generated gear teeth when any part of the fillet curve lies inside a line drawn tangent to the working profile at its point of juncture with the fillet.

WHOLE DEPTH (h_t) is the total depth of a tooth space, equal to addendum plus dedendum, equal to the working depth plus variance.

WORKING DEPTH (h_k) is the depth of engagement of two gears; that is, the sum of their addendums.

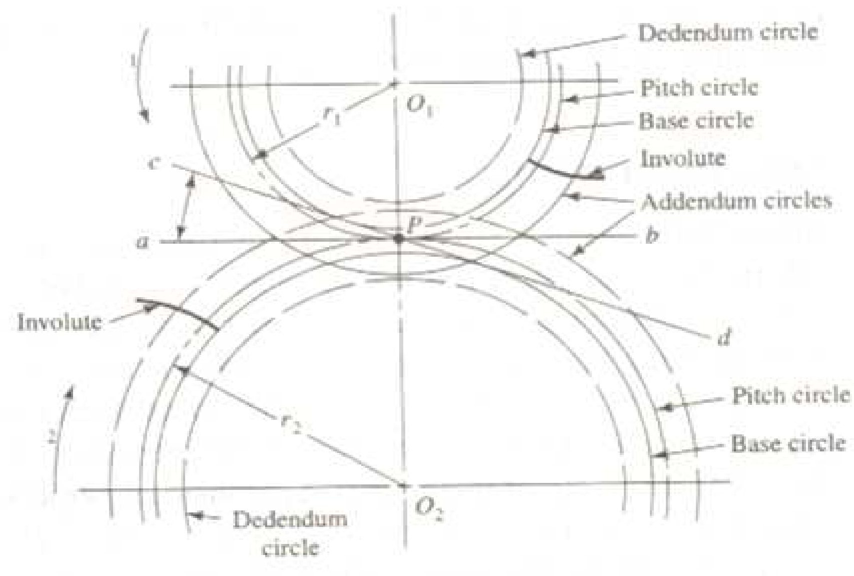


Figure 1.8 Circles of a Gear Layout

1.5 Highest Point of Single Tooth Contact (H Point):-The largest diameter on a spur gear at which a single tooth is in contact with the mating gear. Following figure shows the H point

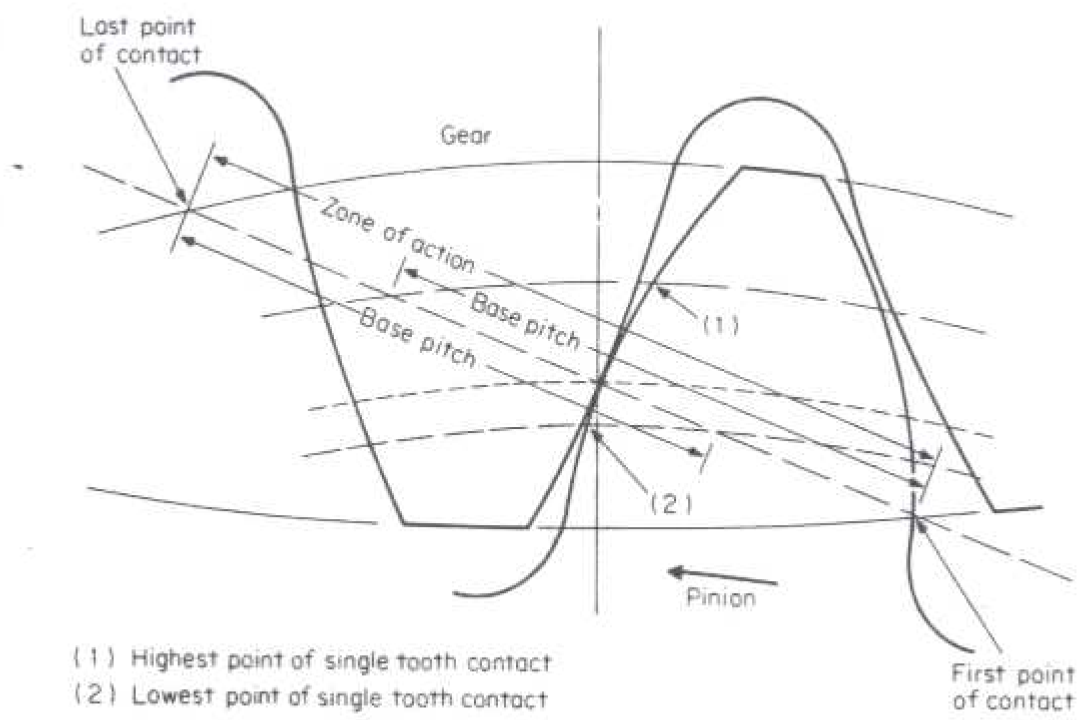


Figure 1.9 Definition for lowest and highest point of single tooth contact

1.6 GEAR DESIGN PROCEDURE

Following are the formulas used in Gear design:-

1. The Pitch Diameter d mm from the equation d=m x N (10^{-3})
2. The pitch line velocity V mm/sec. V= πdn/60.
3. The transmitted load W_t in Newton from the equation

   W_t=H/V

Where H= power in Watts

4. The velocity factor K_v = 6/(6+V)
5. The face width F in mm

   F= W_t/ (K_v.m.Y.σ_p)

   Where $_p$ = permissible bending Stress in the M Pa

6. The minimum and maximum face width 3p and 5p respectively in mm.

LEWIS EQUATION

Each tooth may be considered as a cantilever beam. The force F_n between the tooth surfaces is normal to the surface acting at the tip of tooth. This normal force F_n is resolved into two components, tangential F_t, and radial F_r, acting at a point A, the intersection of the line of action of the normal tooth load and the center of the tooth. F_t will induce a bending stress in the tooth and F_r will induce a uniform direct compressive stress, over the tooth section. The maximum value of bending stress is,

$$f_b = K_f \frac{6.F_t.h_o}{f.t_o^2}$$

Where, f = face width of tooth

K_f = stress concentration factor

h_o = maximum height

t_o = corresponding root thickness

The above equation after rearranging we get,

$$F_t = \frac{f_b.f.\pi.m.y}{K_f}, \text{ N}$$

BUCKINGHAM'S EQUATION

It is known that Buckingham based his design on the dynamic load and the endurance limit of the material. Although one way of considering dynamic effects is the introduction of velocity factor in the Lewis equation, the method of calculating the dynamic load as given by Buckingham is as follows :-

The maximum instantaneous load on the tooth, or the dynamic load, is

$$F_d = F_t + F_i = F_t + \frac{21.V(c.f + F_t)}{21.V + \sqrt{c.f + F_t}} \text{ , N}$$

where, c = a factor depending upon machining errors, and is given as,

$$c = \frac{k.e}{\frac{1}{E_p} + \frac{1}{E_g}} \text{ , N/m}$$

1.7 FATIGUE FAILURE IN GEARS

In high-cycle fatigue situations, materials performance is commonly characterized by an S-N curve, also known as a Wöhler curve. This is a graph of the magnitude of a cyclical stress (S) against the cycles to failure (N). Failure due to repeated loading is known as fatigue.

A small crack, a scratch, or some other such minor defect causes localized deformation. This deformation leads to a small crack if one was not initially present. After cyclic loading, that is, loading in the same way multiple times, the crack grows, and eventually the material fails. A fatigue life curve is a graphical representation of the cyclic loading. Simply, a fatigue life curve, also known as an S-N curve is a plot of the stress amplitude versus the number of cycles the material goes through before it fails. That is, for a certain stress, the material will fail within a certain number of cycles. Figure (1.10) is an example of a typical fatigue life curve.

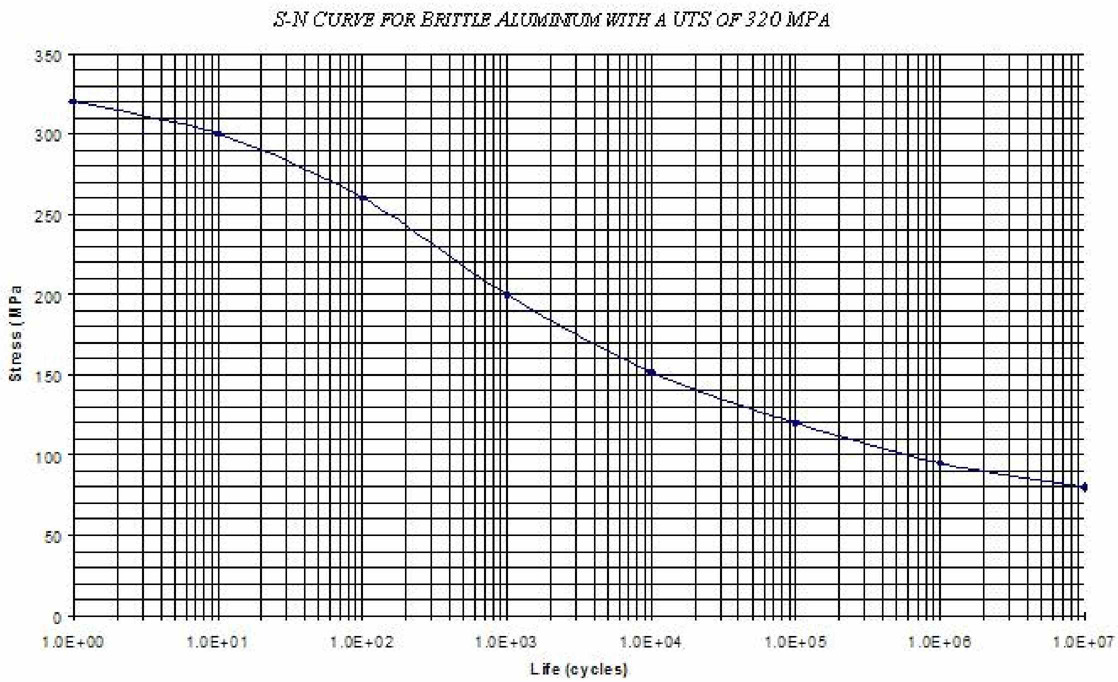


Figure 1.10 S-N Curve

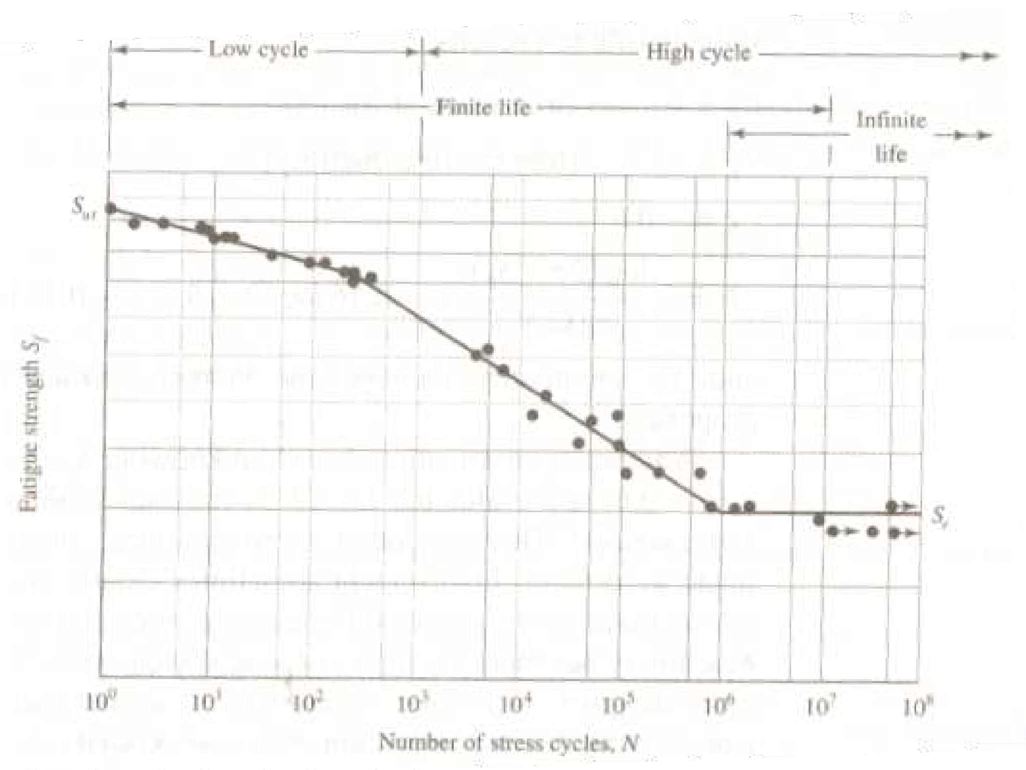


Figure 1.11 Bending fatigue (N vs S_f)

1.8 Design against fatigue failure

Dependable design against fatigue failure requires thorough education and supervised experience in mechanical engineering or materials science. There are three principle approaches to life assurance for mechanical parts that display increasing degrees of sophistication:

- Figure 1.21 Design to keep stress below threshold of fatigue limit (infinite lifetime concept);
- Figure 1.22 Design (conservatively) for a fixed life after which the user is instructed to replace the part with a new one (a so-called lifed part, finite lifetime concept, or "safe-life" design practice);
- Figure 1.23 Instruct the user to inspect the part periodically for cracks and to replace the part once a crack exceeds a critical length. This approach usually uses the technologies of nondestructive testing and requires an accurate prediction of the rate of crack-growth between inspections. This is often referred to as damage tolerant design or "retirement-for-cause".

1.9 Factors That Affect Fatigue Life and Solutions

The Mean Stress is defined as:

$$\sigma_m = \frac{\sigma_{max} - \sigma_{min}}{2}$$

The Mean stress has the affect that as the mean stress is increased, fatigue life decreases. This occurs because the stress applies is greater. The scratches and other imperfections on the surface will cause a decrease in the life of a material. Therefore making an effort to reduce these imperfections by reducing sharp corners, eliminating unnecessary drilling and stamping, shot peening, and most of all careful fabrication and handling of the material. Another Surface treatment is called case hardening, which increases surface hardness and fatigue life. This is achieved by exposing the component to a carbon-rich atmosphere at high temperatures. Carbon diffuses into the material filling interstices and other vacancies in the material, up to 1 mm in depth.

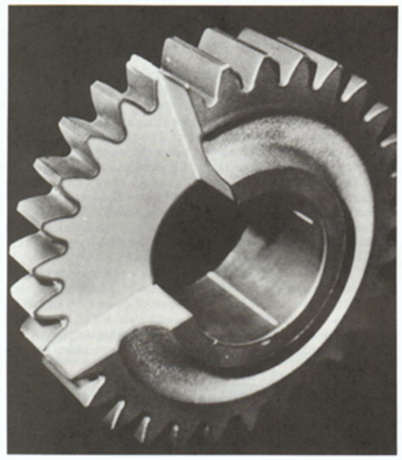


Figure 1.12 Case Hardening of Gears

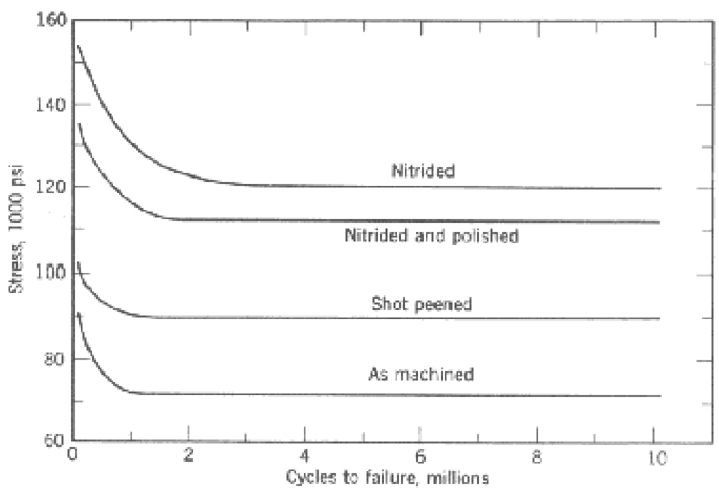


Figure 1.13 Effect of Case Surface treatments on fatigue failure

1.10 Endurance Limit: - The stress level below which a specimen will withstand cyclic stress indefinitely without exhibiting fatigue failure. Rigid, elastic, low damping materials such as thermosetting plastics and some crystalline thermoplastics do not exhibit an endurance limit / Fatigue Limit.

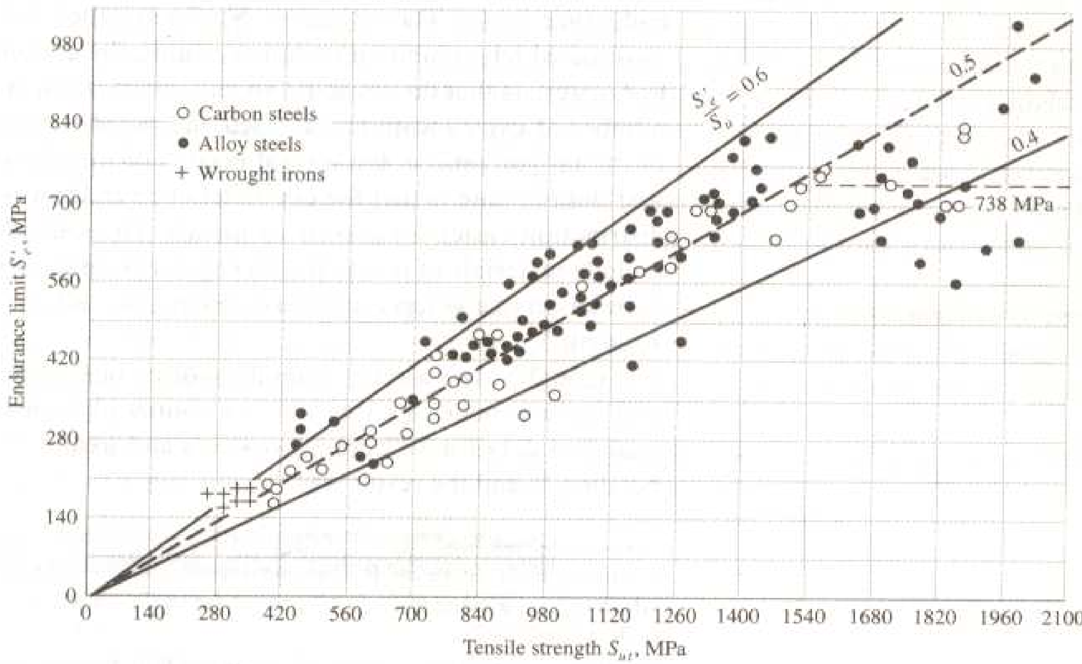


Figure1.14 Endurance limit vs. Tensile strength

The figure (1.14) shows endurance limit ranges from about 40-60 percent of the tensile strength for steels up to about 1400 M Pa Beginning at about S_{ut} = 1400 M Pa, the scatter appears to increase, but the trend seems to level off as shown by Dashed horizontal line at S'_e=700MPa.

1.11 Contact Ratio:- It is the number which indicates the average number of pairs of teeth in contact. Mathematically Contact ratio m_c is defined as :-

$$m_c = \frac{q_t}{p}$$

where q_t= arc of action = $q_a + q_r$

p= circular pitch

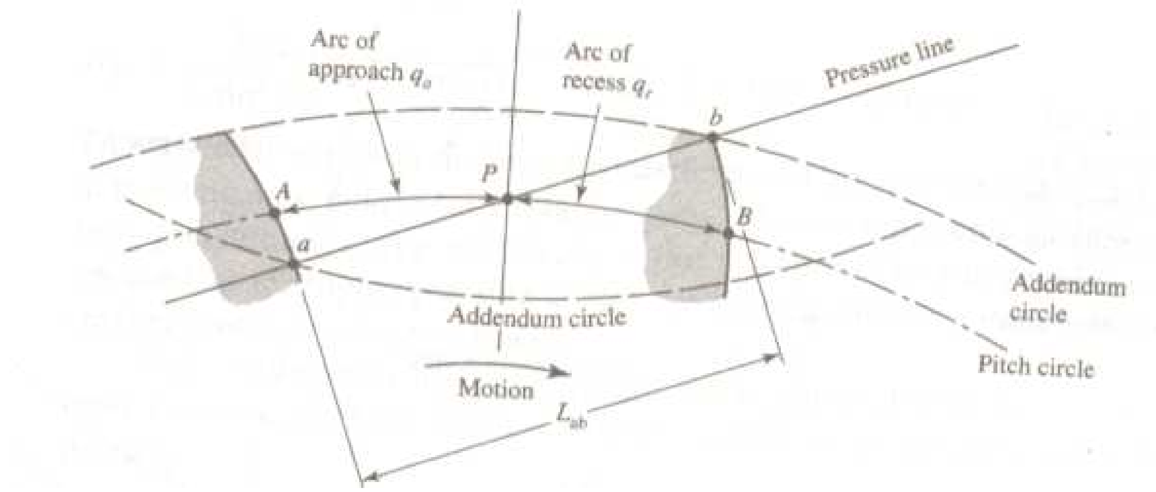


Figure 1.15 Calculation of q_a & q_r

This ratio is also equal to the length of path of contact divided by the base pitch. Gears should not generally be designed having contact ratios less than about 1.20, because inaccuracies in mounting might reduce the contact ratio even more, increasing the possibility of impact between the teeth as well as an increase in the noise level.

CHAPTER 2

LITERATURE REVIEW

The Lewis equation, as discussed before, is used for calculating bending stress for various tangential loads at different points on the tooth profile. The drawback in the stress analysis of the gear tooth using this equation doesn't give the desired results at desired points on the tooth profile. Therefore, investigators, analyzing the gear tooth for stresses, have done several studies.

Fredette and **Brown** [11] used holes drilled across the entire tooth as a function of size and location. The ultimate objective of this work was to find the overall effect of hole size and location on the critical stresses in the gear.

.Many of these researchers used the finite element analysis techniques for the analysis of the gear tooth. **Wilcox, L** [20] gave an analytical method for calculating stresses in bevel and hypoid gear teeth.

More recently number of authors had also done the analysis of the gears for the applied bending forces.

Litwin, Chen, Lu, [13] and **Handschuh** [16] again applied finite element analysis on a loaded tooth for the determination of load share, real contact ratio, precision of motion and the stress analysis. It was the work done in parts by these authors i.e. Litwin did the determination of the load share, real contact ratio was determined by Chen and Lu and Handschuh did the stress analysis.

A program was given by **Gosselin, Claude, Clautier** [12] predicting the motion error of spiral bevel gear sets under load, and explored some of the influences of the unloaded motion error curve shape and amplitude over the kinematical behavior under load. The effects of tooth composite deflection caused by bending and shearing, tooth contact deformation and initial profile separation due to profile mismatch were considered in the development.

Chen and **Tsai** [17] developed a finite element model applied to an involute gear considering friction effects. The loss of Torque transmission due to friction and effective friction coefficient were evaluated and computed.

Moriwaki [15] developed a technique named Global Local Finite Element Analysis (GLFEM) and applied it to a gear tooth for its stress analysis. He realized that for doing the stress analysis of the gear using the finite element analysis, a load acts at a point on the tooth profile and for that a fine subdivision is required at the applied load point. In GLFEM, no fine subdivision is required for the analysis. This method also guarantees an easy determination of the critical section. In fact, GLFEM is a numerical analysis technique that combines finite element solutions and the classical analytical ones on the basis of the energy principle. The application of finite element method for LTCA was also the subject of research by **Chao** and **Baxter** [19], **Drago** and **Uppaluri** [18].

Chen and **Tsay** [5] investigated the contact stress and bending stress of a helical gear set with localized bearing contact, by means of finite element analysis (FEA). They proposed a helical gear set comprised of an involute pinion and double crowned gear. Mathematical models of the complete tooth geometry of the pinion and the gear had been derived based on the theory of gearing. A mesh generation program was also developed for finite element analysis.

Chien-Hsing Li, Hong-Shun Chiou [6] established a batch module called, ”integration of finite element analysis and optimum design” by taking gear systems as testing examples. This batch module consisted of I-DEAS, ABAQUS/Standard and MOST software, which serve as the preprocessor, the numerical solver and the optimizer, respectively. A simple and practical method was developed, through which this module was enabled which would search for contact nodes and elements and also it would automatically define the contact surfaces for contact analysis. For the testing example, a simple gear-pair system and a complete planetary gear system were used for this integrated module. The module would automatically construct the geometrical model, analyze the contact stress and solve for the optimal solutions when gearing parameters are input.

Simon [9] performed stress analysis in hypoid gears, by using finite element method, in order to develop simple equations for the calculation of tooth deflections and stresses. He developed a method for the automatic finite element discretization of the pinion and the gear. The full theory of mismatched hypoid gears was applied for the determination of the nodal point coordinates on the teeth surfaces. He developed a corresponding computer program. With the help of this program the influence of design parameters and load position on tooth deflections and fillet stresses is investigated. On the basis of results, which were obtained by performing a big number of computer runs, equations for the calculation of tooth deflections and fillet stresses were derived.

Zhang and **Fang** [10] used an approach for the analysis of tooth contact and load distribution of helical gears with crossed axis. That approach was based on a tooth contact model that accommodates the influence of tooth profile modifications, gear manufacturing errors and tooth surface deformation on gear mesh quality. In this approach the tooth contact load was assumed distributed along the tooth surface line. As an example, the computer program analyzed the contact of a pair of helical gears with small crossing angle. It was found from analysis that helical gears with small crossing angles have meshing characteristics and load distribution similar to those of parallel axis gears.

Spitas and **Costopoulos** [2] studied the idea of spur gear teeth with circular instead of the standard trochoidal root fillet and it was investigated numerically using BEM. The strength of these new teeth was studied in comparison to the standard design by descretizing the tooth boundary using iso-parametric Boundary elements. In order to facilitate the analysis the teeth were treated as non-dimensional assuming unitary loading normal to the profile at their Highest Point of Single Tooth Contact. It was demonstrated that the novel circular design surpasses the existing trochoidal design of the spur gear tooth fillet in terms of fatigue endurance without affecting the pitting resistance. The proposed geometry does not produce undercut teeth even for small number of teeth.

Kapelevich and **Kleiss** [7] gave an alternative approach to traditional gear design. The approach was direct gear design. It allowed analysis of a wide range of parameters for all possible gear combinations in order to find the most suitable solution for a particular application. This optimum gear solution can exceed the limits of traditional rack generating methods of gear design. Direct gear design for asymmetric tooth profiles opens additional reserves for improvement of gear drives with unidirectional load cycles.

Guigand and **Icard** [1] gave a method for simulating loaded face gear meshing. It simulated the loaded behavior of face gear meshing and provides information such as the instantaneous pressure distribution across the entire width of the teeth in contact and loaded transmission error. Analytical simulations were used to define the geometry and study the unloaded kinematics. The aim was to obtain the best possible loading of face gear meshing in order to avoid line contact sensitivity due to misalignment.

Beghini and **Santus** [4] proposed a simple method to minimize the Peak-to-Peak Transmission Error (PPTE) for a spur gear set. A parametric analysis using advanced software was performed to obtain a general understanding of the problem. The main aim was to develop a simple method for profile optimization in terms of PPTE, which was the main cause of whining noise in spur gears. A combined semi-analytical and FEM software had been used for performing meshing simulations.

Parker and **Vijayakar** [8] investigated the dynamic response of a spur gear pair using a finite element/contact mechanics model. The gear pair was analyzed across a wide range of operating speeds. The non-linearity source was contact loss of the meshing teeth, which occurred for large torques despite the use of high precision gears. Using a detailed contact analysis at each time step as the gears rolled through the mesh, dynamic mesh forces were calculated. A semi-analytical model near the tooth surface was matched to a finite element solution away from the tooth surface.

Hiremagalur and **Ravani** [3] studied the effect of backup ratio in spur gear root stresses analysis and design. Backup ratio was considered important in understanding rim failures that start at the tooth root. Here an analytical approach, based on theory of elasticity, was used to provide a computational formulation for root stress calculations in spur gears.

Now a day's composite material finds increasing applications ranging from spacecraft to small instruments. Many types of gear pump use composite gears, however little literature is available on their use. **Vijayrangan** and **Ganesan** [14] obtained results by static stress analysis of composite gears using a three-dimensional finite element approach. Performance of two orthotropic material gears were presented and compared with mild steel gear. From the results it was concluded that composite material such as graphite/epoxy could be thought of as a material for power transmission gears.

Dally and **Riley** [23] tried to minimize the stress in a finite plate by drilling a hole in it. They optimized the hole profile for minimizing the photo elastic stress. For optimization they used the software ANSYS 5.3. They not only did their study on a hole profile but also on the square profile and got the results.

The idea of using holes to reduce stresses is not a new one. In 1990, Dippery experimented with the use of supplementary holes in a structure as a method of reducing the stress concentration that was already present. His result showed that stress concentration reductions are possible in a generic shape using holes as stress relief.

Yang [22] showed that similar reductions are possible in gears. His work was limited to hole placement in the region of relatively low stress of the bending gear tooth.

CHAPTER 3

PROBLEM DEFINITION

Gears which are very rigid develop high stress concentration at the root and contact point when subjected to loads. Due to these high stresses at the root and contact point there is a higher chance of fatigue failure at these locations. As the contact point shifts along the profile of the tooth a surface fatigue failure is more likely but the quick shifting of the load and enhanced material properties due to surface treatments means that this failure is not as critical. The repeated stress that occurs on the fillets is practically found to be the deciding factor in fatigue failure of the gear tooth. There is scope of improved design of gears by an introduction of stress relief features. These reduce stiffness which causes a more distributed lower stress at the fillets increases gear life. Stress relief features provide flexibility and help in reducing the points of stress concentration. It is proposed to investigate the effect of stress relief features of different size, shape, location and number. A study is done for spur gear with involute profile shapes for effects of stress relief feature of hole. The work done by Fredette L. and Brown M [11] is to be studied and furthered in this work.

3.1 Implementation

Hardware Used: - Pentium 4 CPU 3.00GHz 256 MB of RAM, Display 64 MB.

Software Used:-

All the modeling is done in the CAD/CAE software Pro-Engineer Wildfire 3.0 and the analysis is done in Pro-Mechanica Wildfire 3.0.

Experiments and Studies

Four different studies are done on spur gears having different specifications.

The following are the specifications of the gear that have been used for the analysis. All the gears have pressure angle (ø) 20°, they use the general formulas of gear design as follows:-

Pitch circle Dia. (PCD) =module(m) x no. of teeth,

Tooth thickness = (π x module) /2,

Root fillet = 0.2 x module,

Addendum Dia.(D_a)= PCD +2 x module,

Dedendum Dia.(D_d) = PCD – 2.5 x m,

Base dia. (D_b) = PCD x cosø

	Module	PCD	D_b / D_d
Gear 1	2	50	<1
Gear 2	2	100	>1
Gear 3	5	90	<1
Gear 4	5	350	>1
Gear 5	6	108	<1
Gear 6	6	360	>1
Gear 7	7	140	<1
Gear 8	7	350	>1

Study I: A gear was analysed for variation in stresses along the path of tooth contact. A contact ratio of one was taken and variation of tangential force couple on the tooth was studied for two cases of the gear design of the same module but having base circle larger than dedendum circle in one and base circle smaller than the dedendum circle in the other.

Stress analysis for the load contact point traveling along the involute curve is done for both gears.

Study II: Gears are analysed considering actual contact ratio greater than one which reduces the region in which a single tooth pair is in contact. The point of contact where maximum stress occurs is determined for all eight test gears and the study of variation of this H_dia. for contact ratio greater than one is done.

Study III: Comparison of stress analysis for highest load at H_dia. for all eight test gears is done. The representative weak profile which is suitable for stress relief studies is then selected.

Study IV: The fillets which are the most stressed regions in the tooth profile are made as the feature of focus, the selected gears are tested and the trailing fillet which has maximum stress is chosen for stress relief studies.

Stress relief features are made along the Dedendum, root-fillet, flank and involute profiles. Global sensitivity analysis is done taking the maximum stress at the fillets as a measure for location of hole, offset from the profile, diameter of the hole and multiple holes.

The studies described above are done in sequence in order to arrive at the suitable point of force application on a gear tooth which causes highest stresses at the fillets.

The gear which would practically be the most stressed is selected from the eight. Then the effect of stress reducing feature of hole is done. The aim of studies is to arrive at an optimized design of the hole as the stress relief feature of a gear tooth.

CHAPTER 4

RESULTS AND DISCUSSION

4.1 Experimentation and studies

STUDY 1

This study is done only on the gears having the specifications of module=2, T=25 & Module=2, T=50. Measure of tangential force couple on the tooth is taken. The distance of the point where load is applied is measured from the base circle. A study is done on the tangential force couple variation on the tooth along the entire involute curve.

Stress analysis is done to get the sensitivity of the stress at the fillets to traveling of the load contact point along the involute curve.

The graphs below show the variation of force couple.

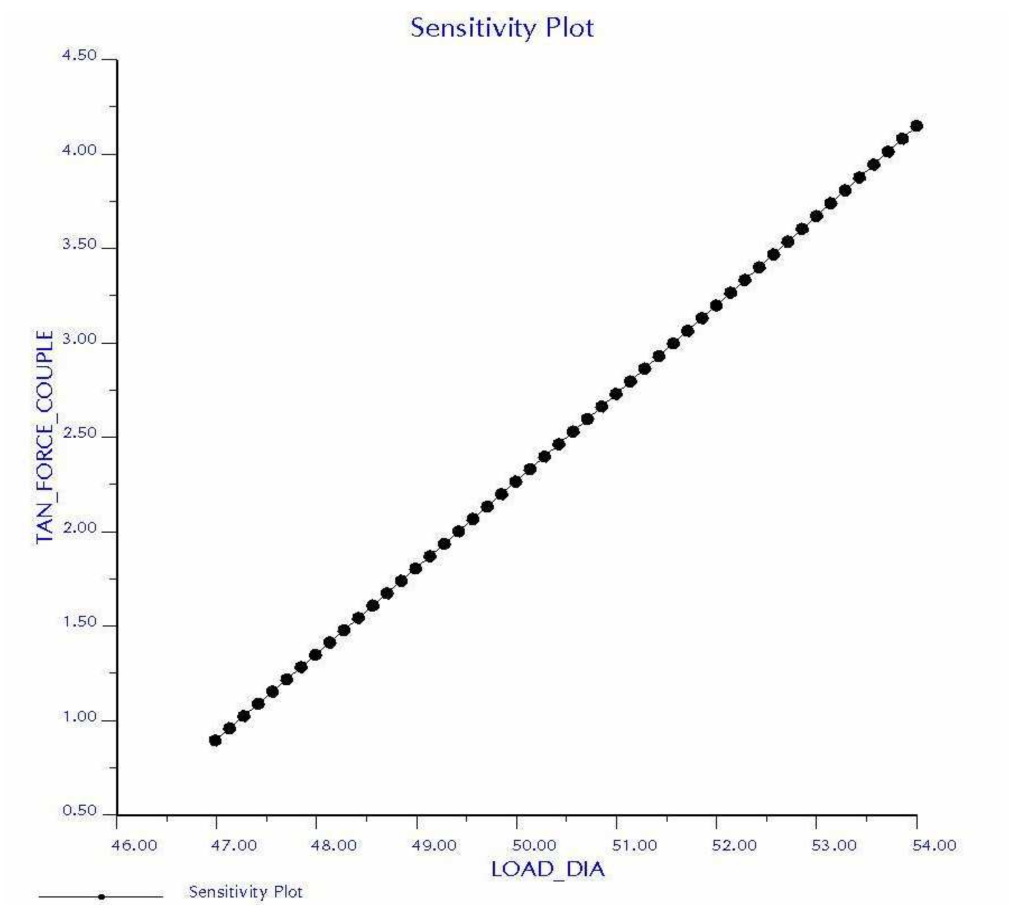


Figure 4.1 Couple variation graph for module =2, T=25

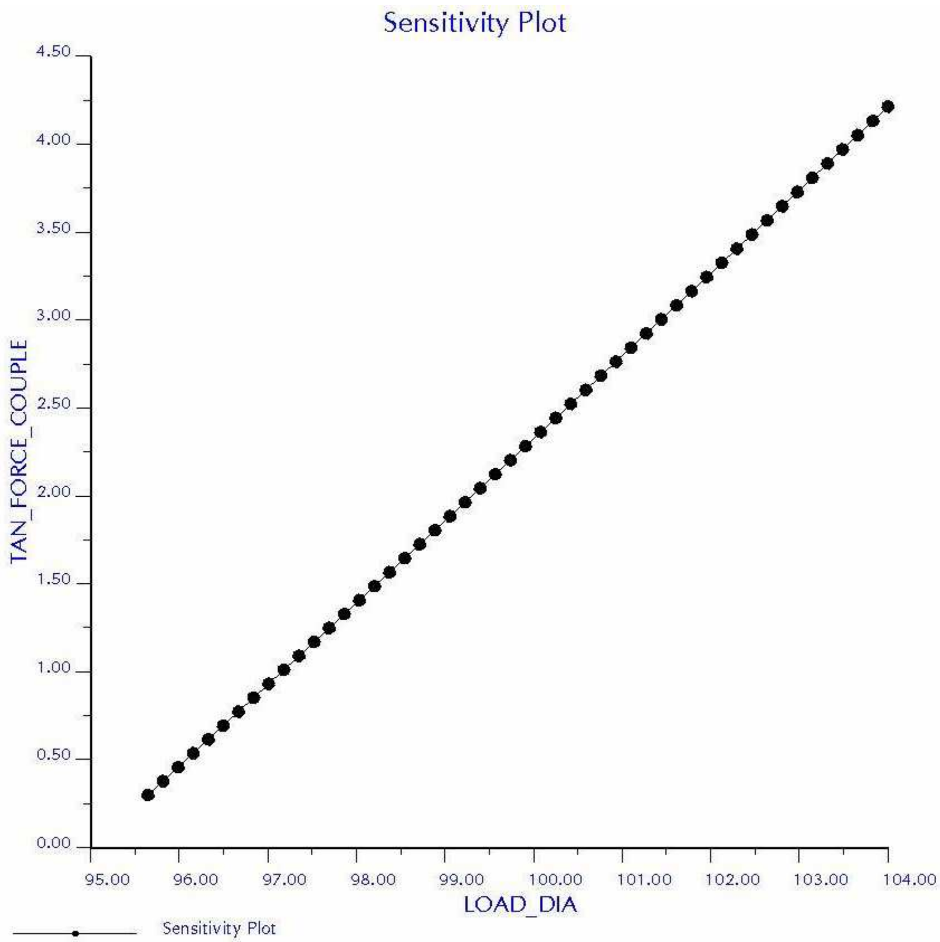


Figure 4.2 Couple variation graph for module =2, T=50

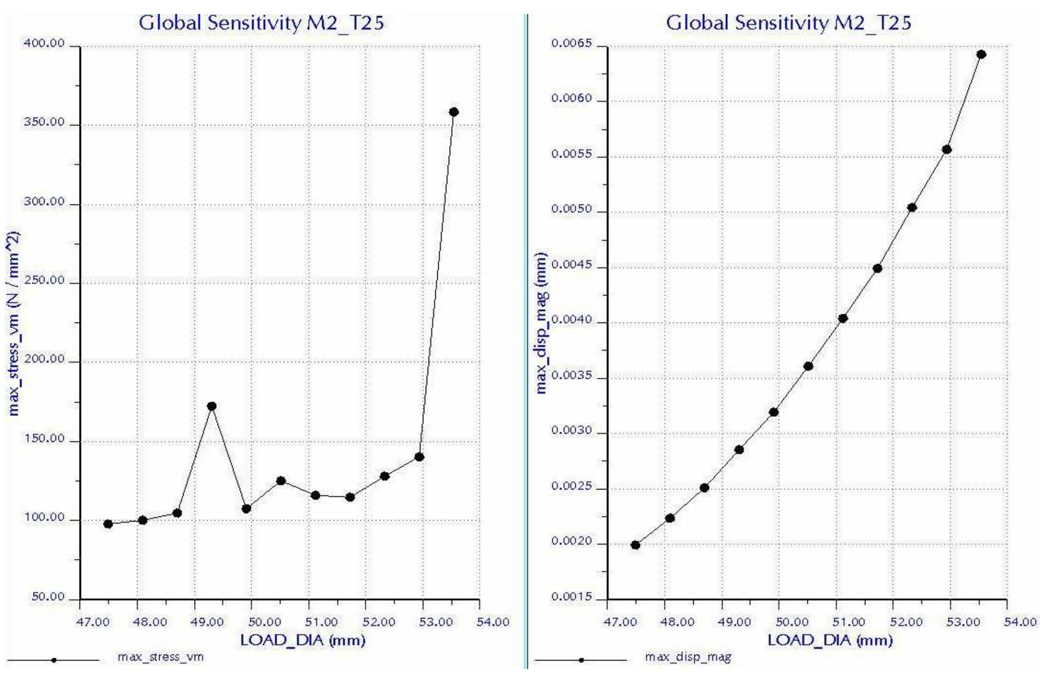


Figure 4.3 Global sensitivity analysis for m=2 T=25

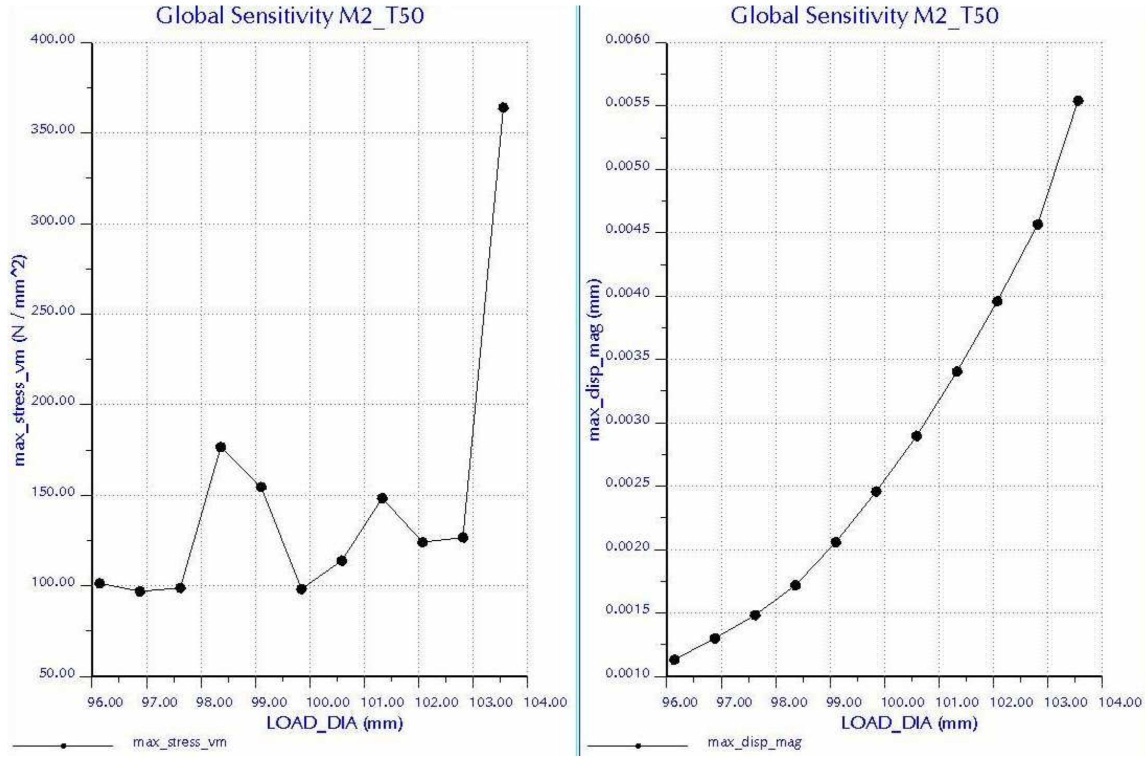


Figure 4.4 Global sensitivity analysis m=2 T=50

Conclusion: - From the graph shown above it is seen that the tangential force couple increases with the diameter of load application point. It is a nonlinear variation and the rate of increase of the couple increases till half way along the involute, beyond which it decreases. This is more noticeable in a smaller size gear with radial flanks i.e. base circle diameter is less than dedendum circle diameter.

The two graphs of stress at the fillets when load application point moves along the involute show that there is a general tendency of stress levels increasing as the diameter of load application point increases.

STUDY 2

In this study gears are analysed considering that actual contact ratio is kept greater than one at a value which is suggested to be at least 1.20 [24, Pg 838] this reduces the region in which a single tooth pair is in contact. Considering the further reduction due to inaccuracies in mounting a contact ratio of 1.15 is taken. The point of contact where maximum stress occurs is determined for all eight test gears and the variation of this H dia for gear ratio greater than one is studied. The point in consideration is the highest point of single tooth contact. Then the gear ratio where it is maximum will be the taken for application of force for all other studies.

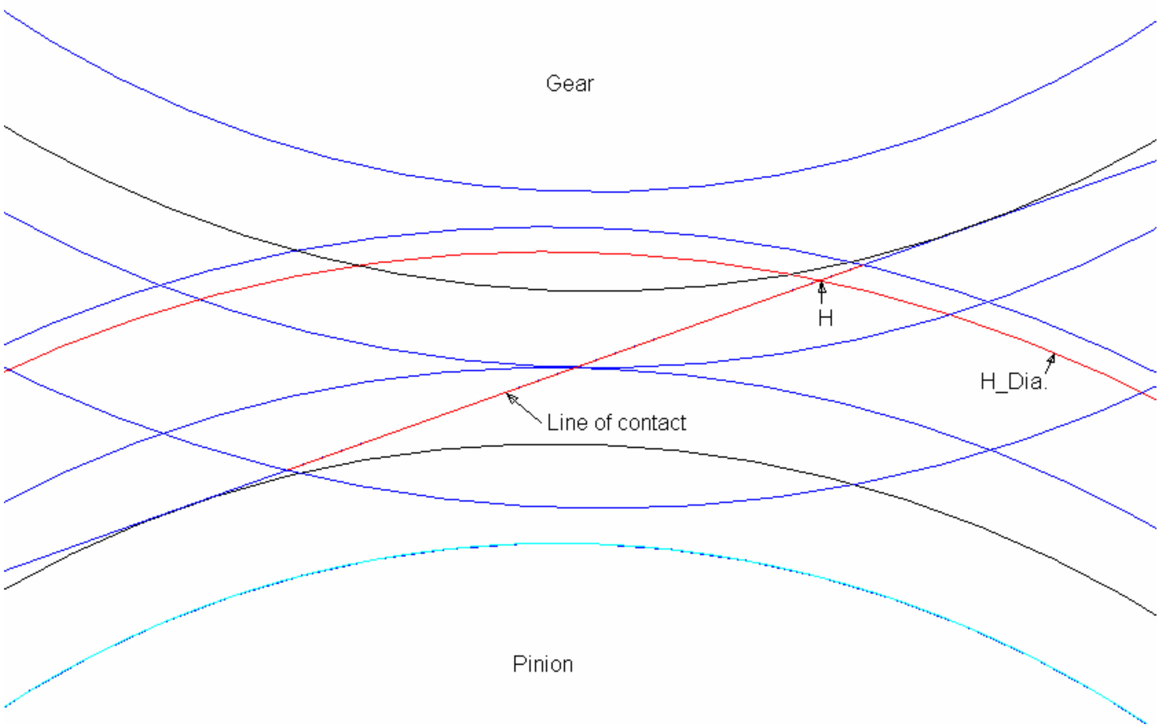


Figure 4.5 Method to find H

The above figure shows the position of H point, taken as a distance from the center of the gear. Line of the contact is the path on which single tooth contact occurs. The point H is the highest point of single tooth contact after which two teeth are in contact at the same time.

Taking contact ratio varying from one to fifty the variation of the H dia is plotted i.e the highest point of single tooth contact is found for all the contact ratios.

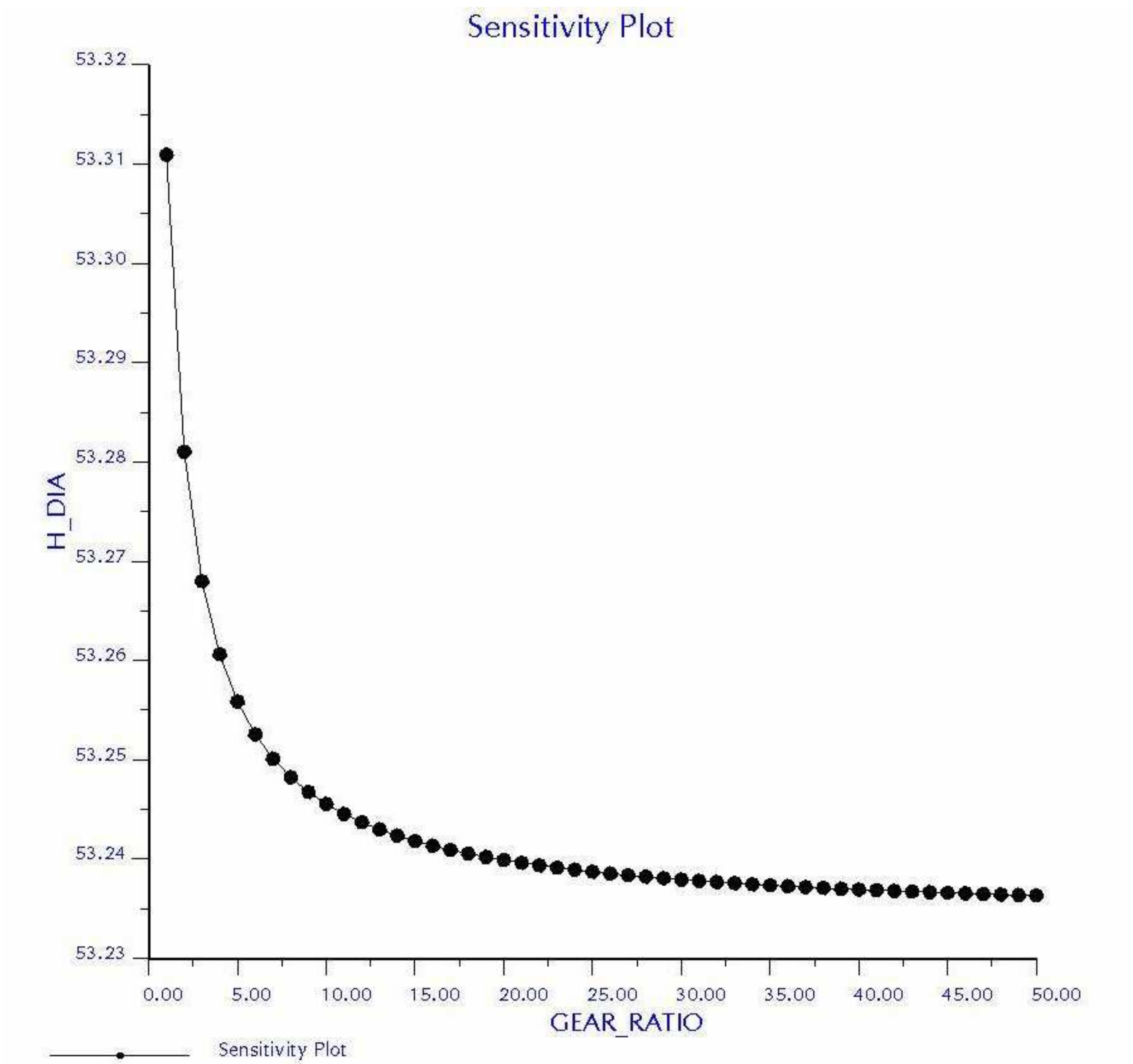


Figure 4.6 Variation of H dia., M=2, P=50

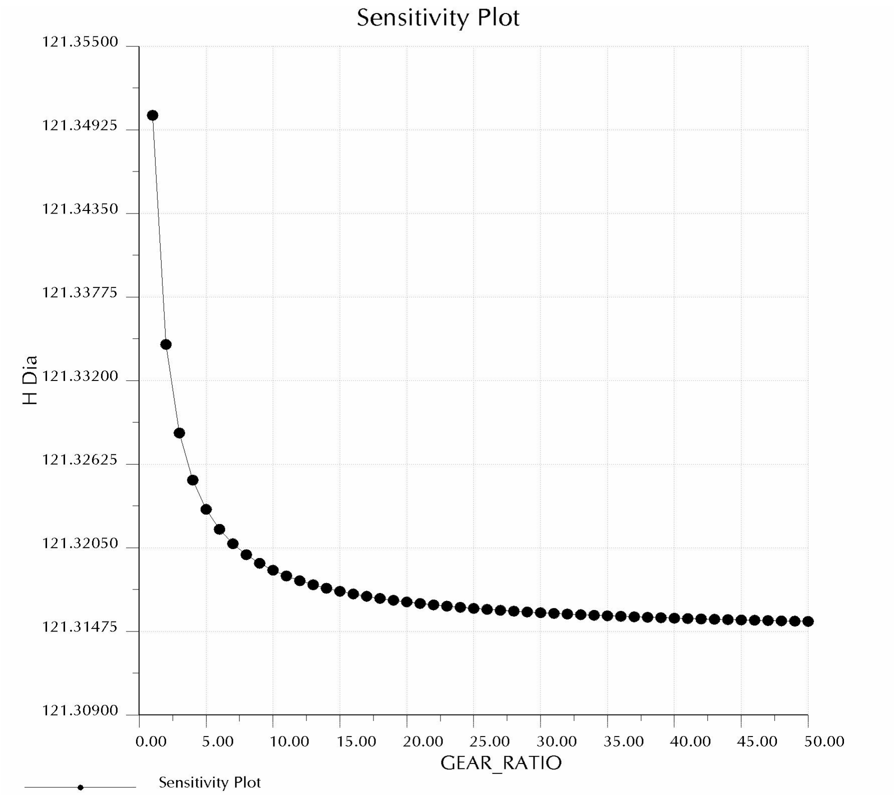


Figure 4.7 Variation of H dia., M=2, P=100

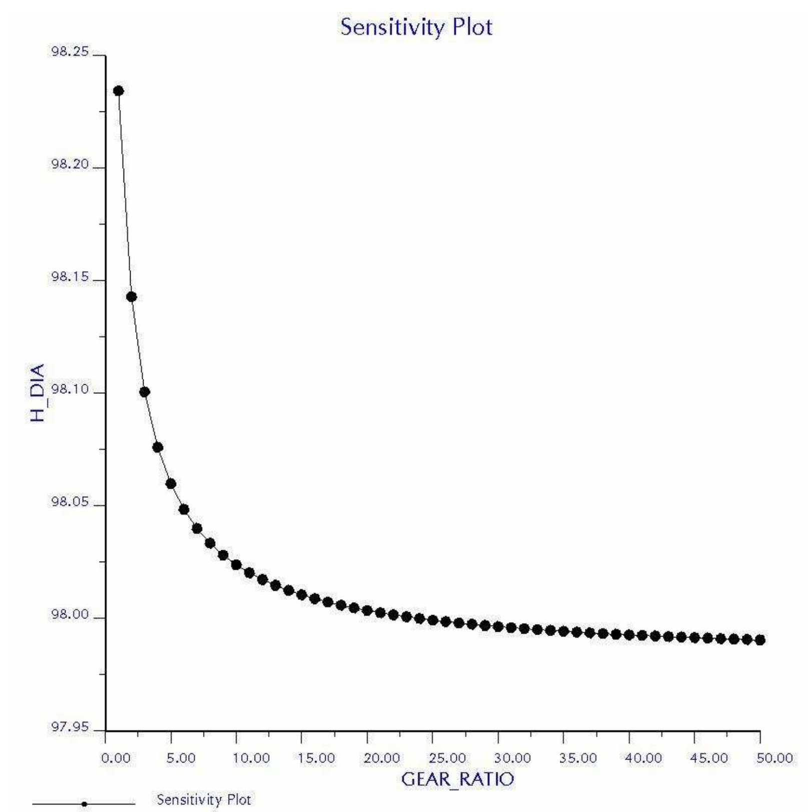


Figure 4.8 Variation of H dia., M=5, P=90

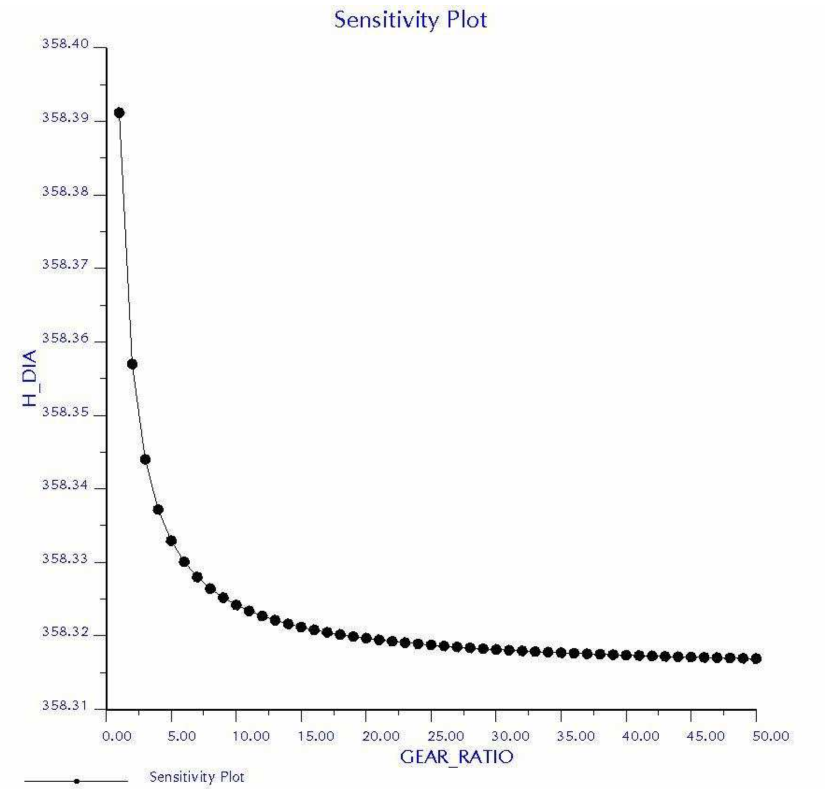


Figure 4.9 Variation of H dia., M=5, P=350

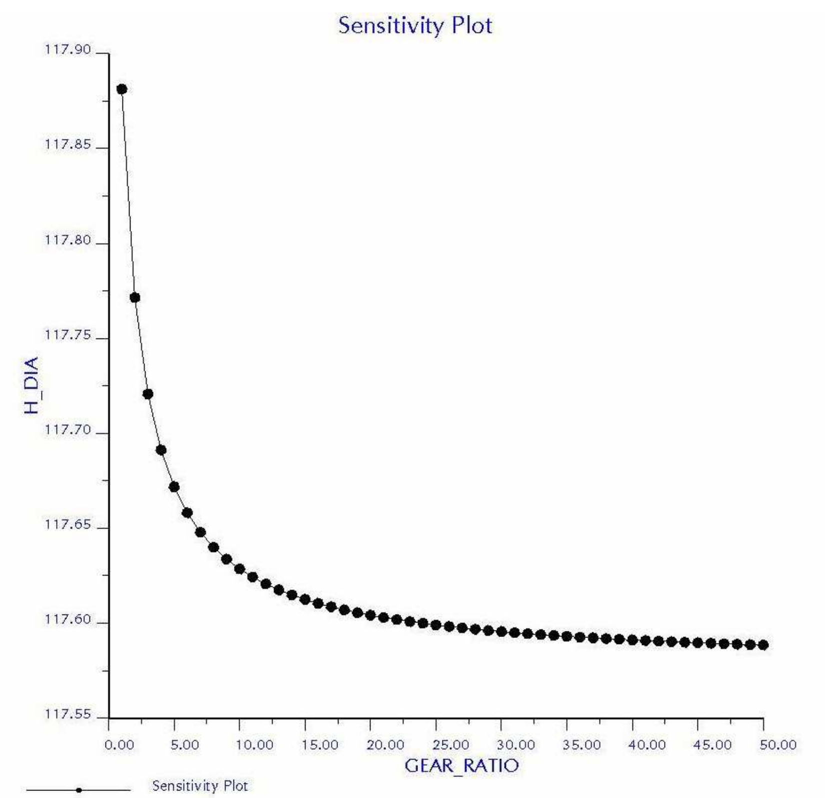


Figure 4.10 Variation of H dia, M=6, P=108

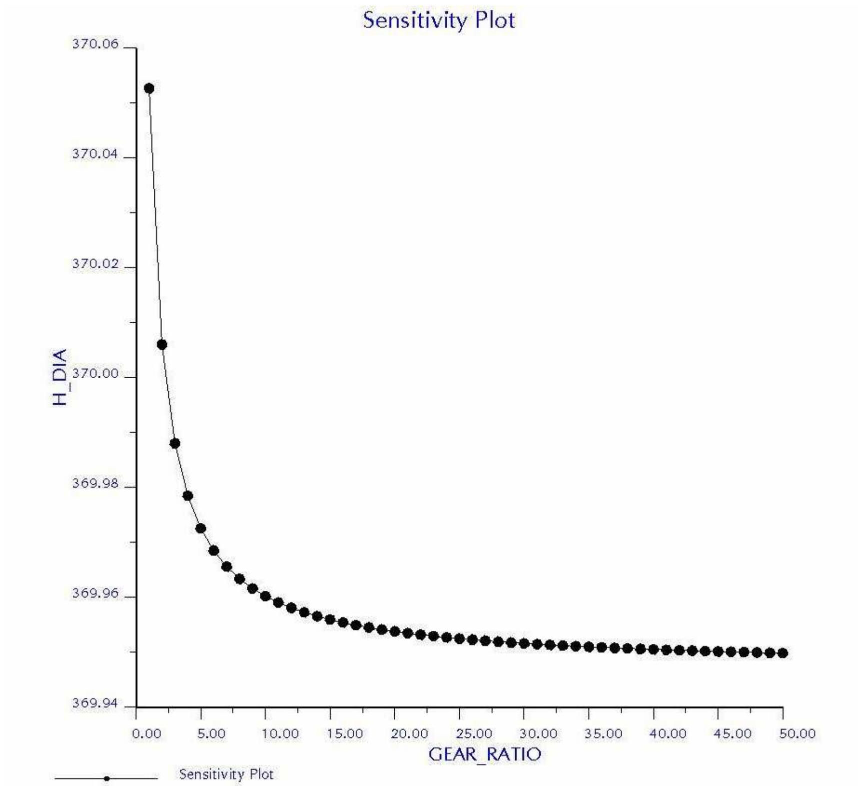


Figure 4.11 Variation of H dia., M=6, P=360

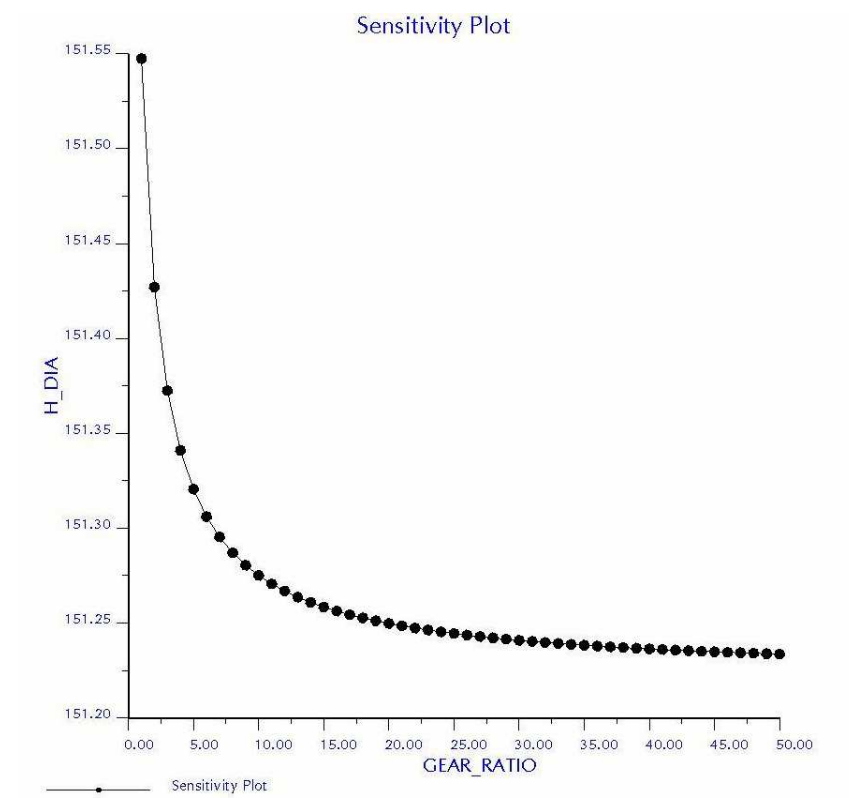


Figure 4.12Variation of H DIA M=7 P=140

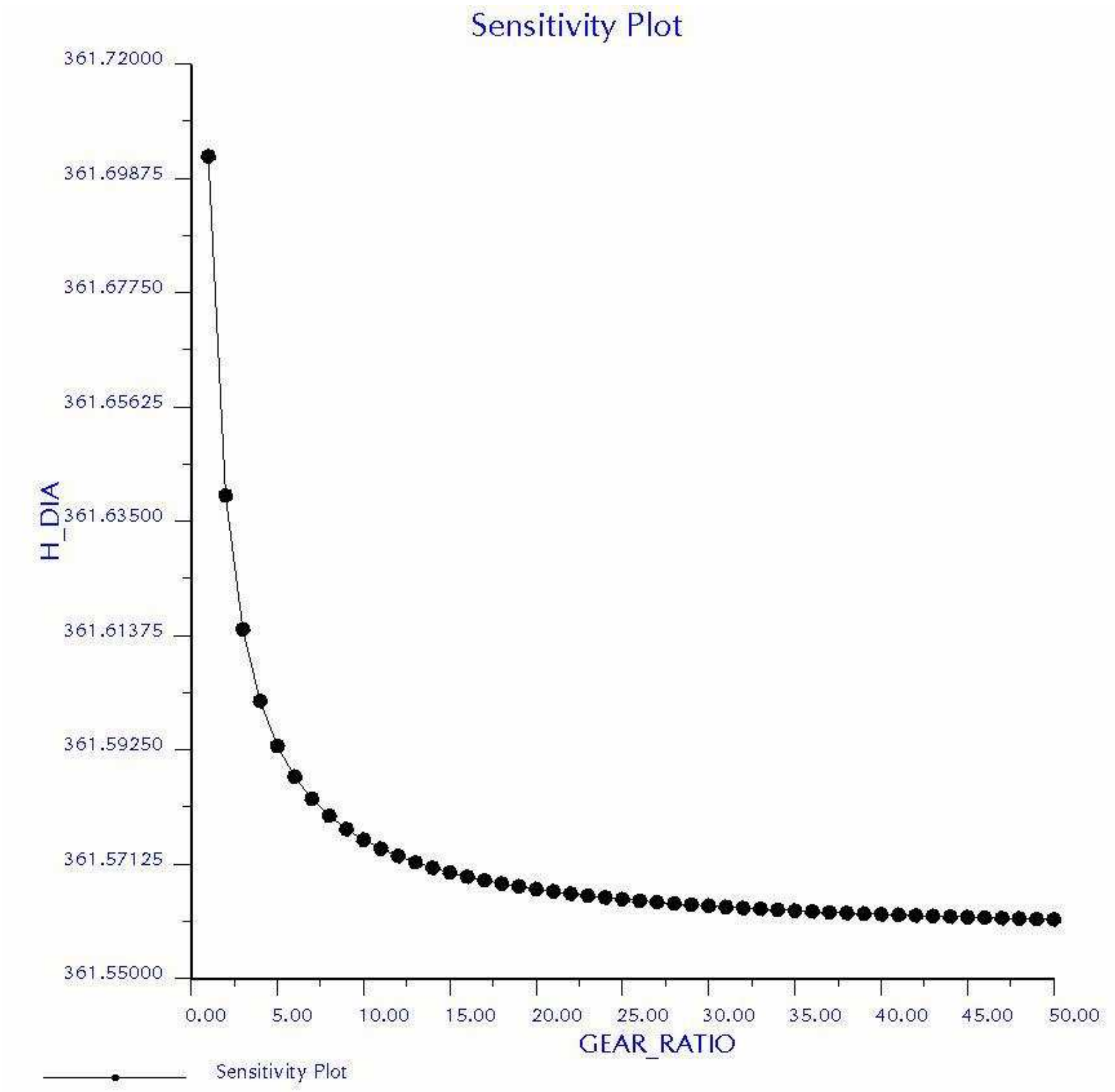


Figure 4.13 M=7 P=350

Conclusion: - For a given module, if the size of gear is increased then there is a fall in the value of H dia. The H dia. changes sharply for gear ratios less than 5. As the gear ratio increases, change in H dia gradually decreases till there is very less change for gear ratios greater than 20. The overall change in H dia. is noticed to be of a very small measure, with in a mm. The overall effect of gear ratio is such that the highest H_Dia. is for gear ratio equal to one and this is taken for the further studies.

As module increases the magnitude of change in H dia for different gear ratios is of a very small order.

STUDY3

Comparison of stress analysis for highest load at H diameter for all eight test gears is done. The representative weak profile which is suitable for stress relief studies is then selected.

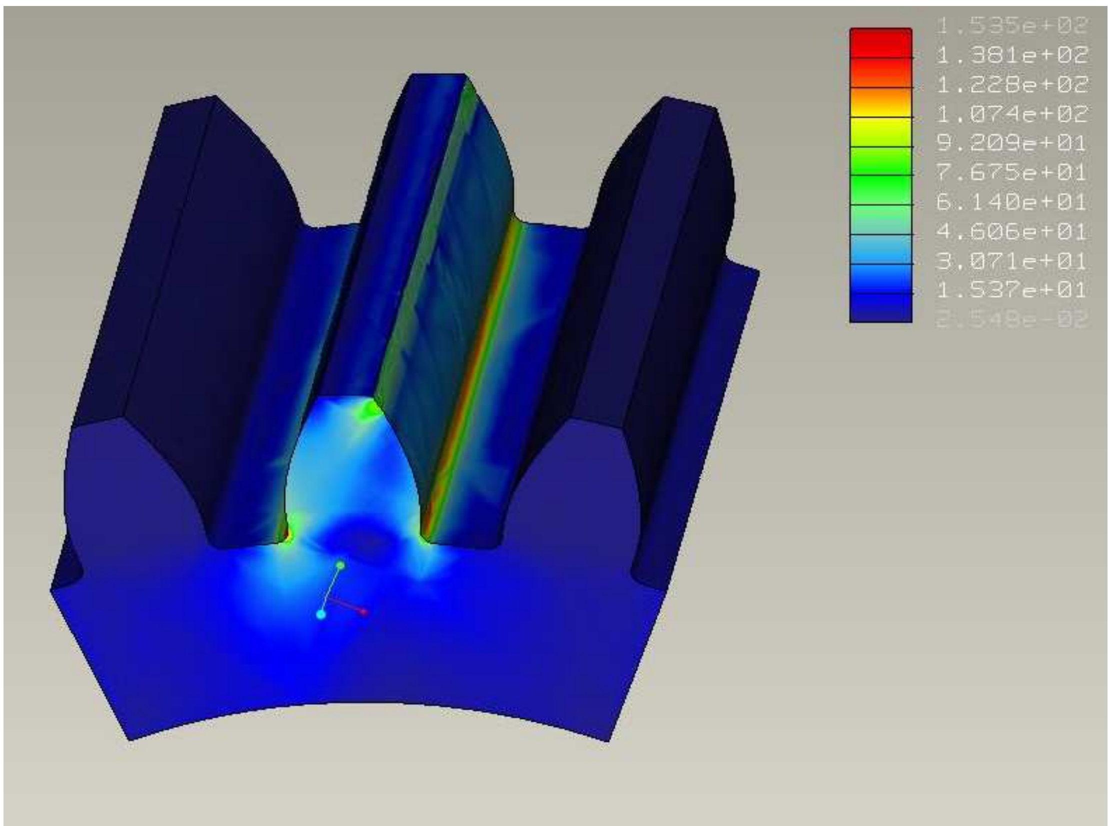


Figure 4.14 a) Stress variations in Gear 1

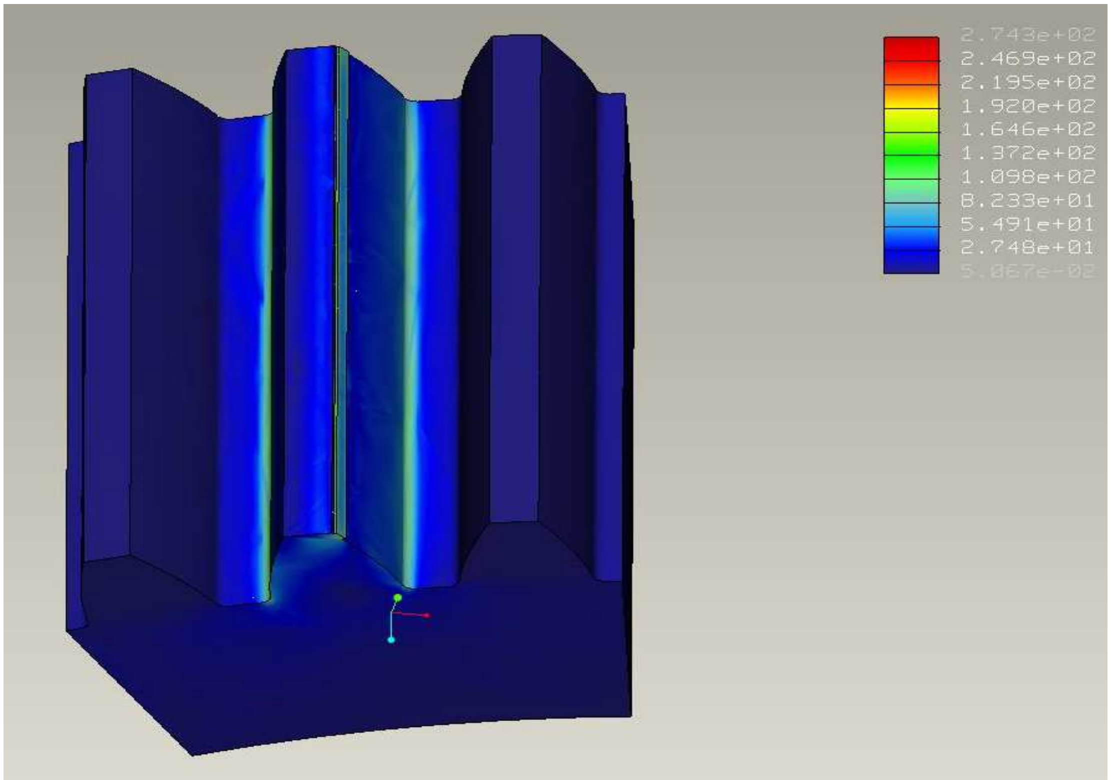


Figure4.15 Stress variations in Gear 2

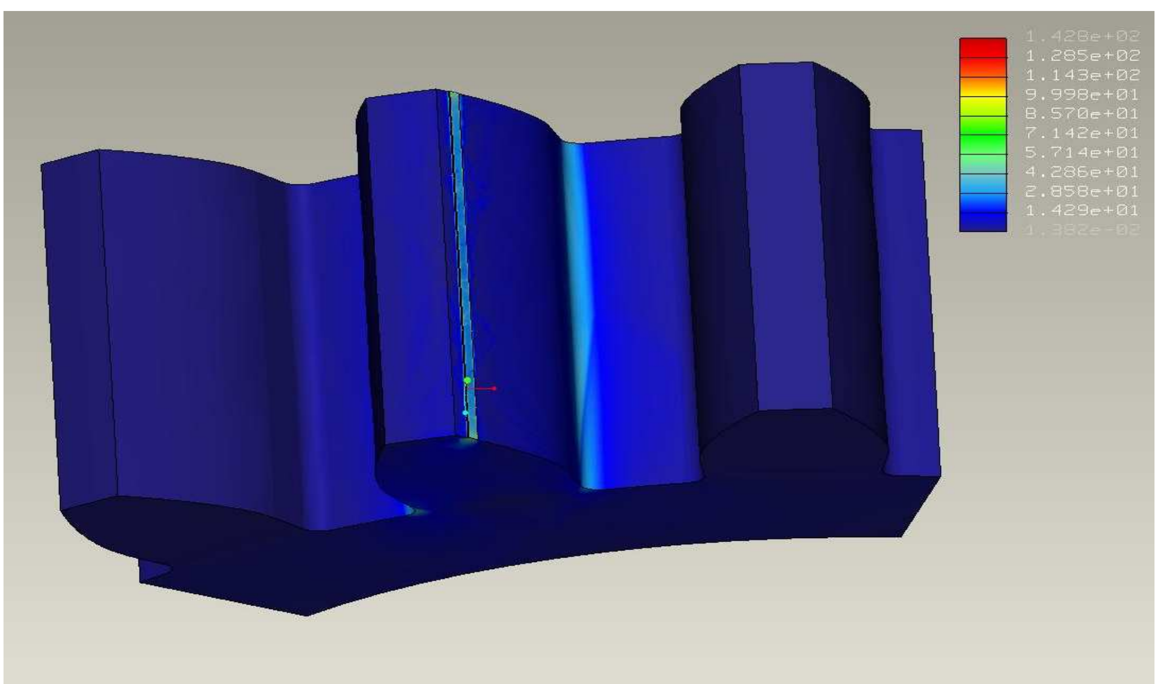


Figure fringe 4.16 Stress variations in Gear 3

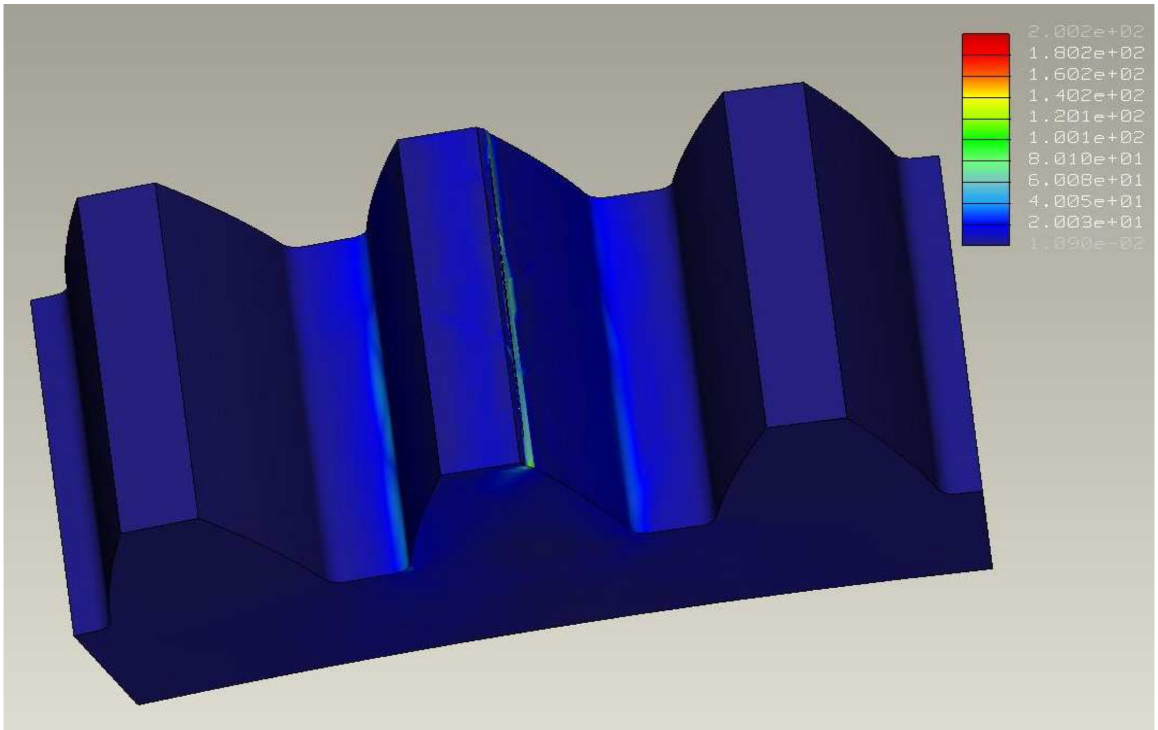


Figure 4.17 Stress variations in Gear 4

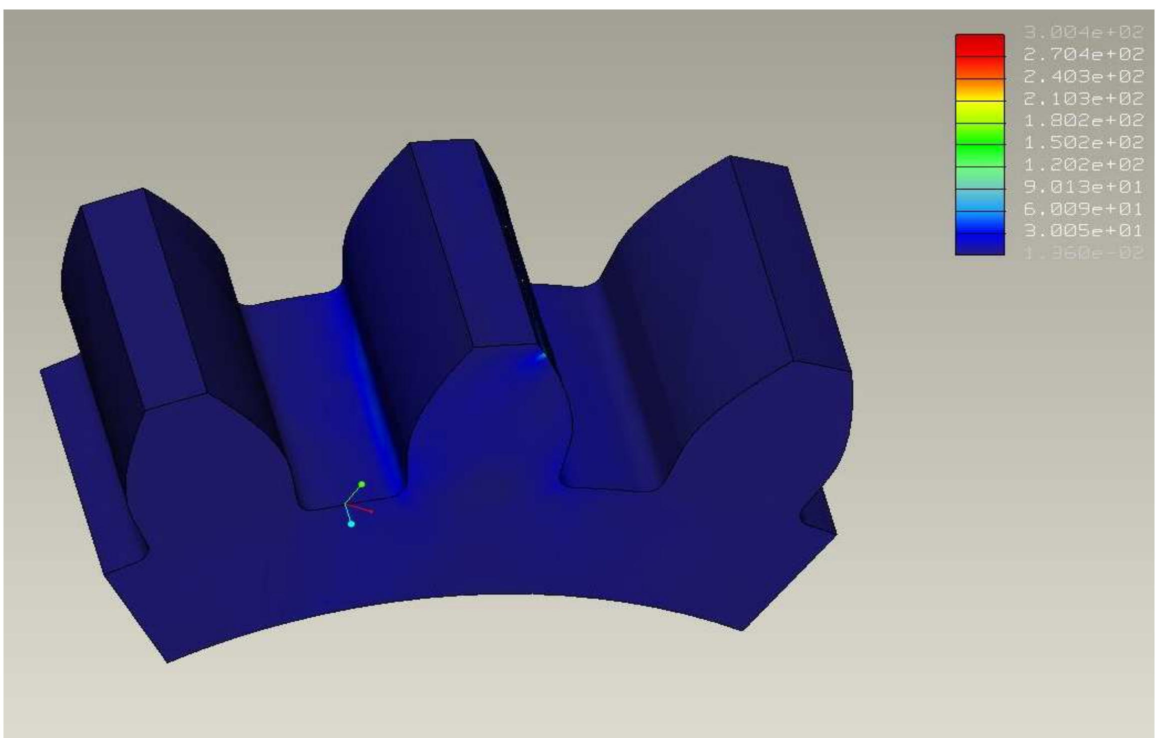


Figure 4.18 Stress variations in Gear 5

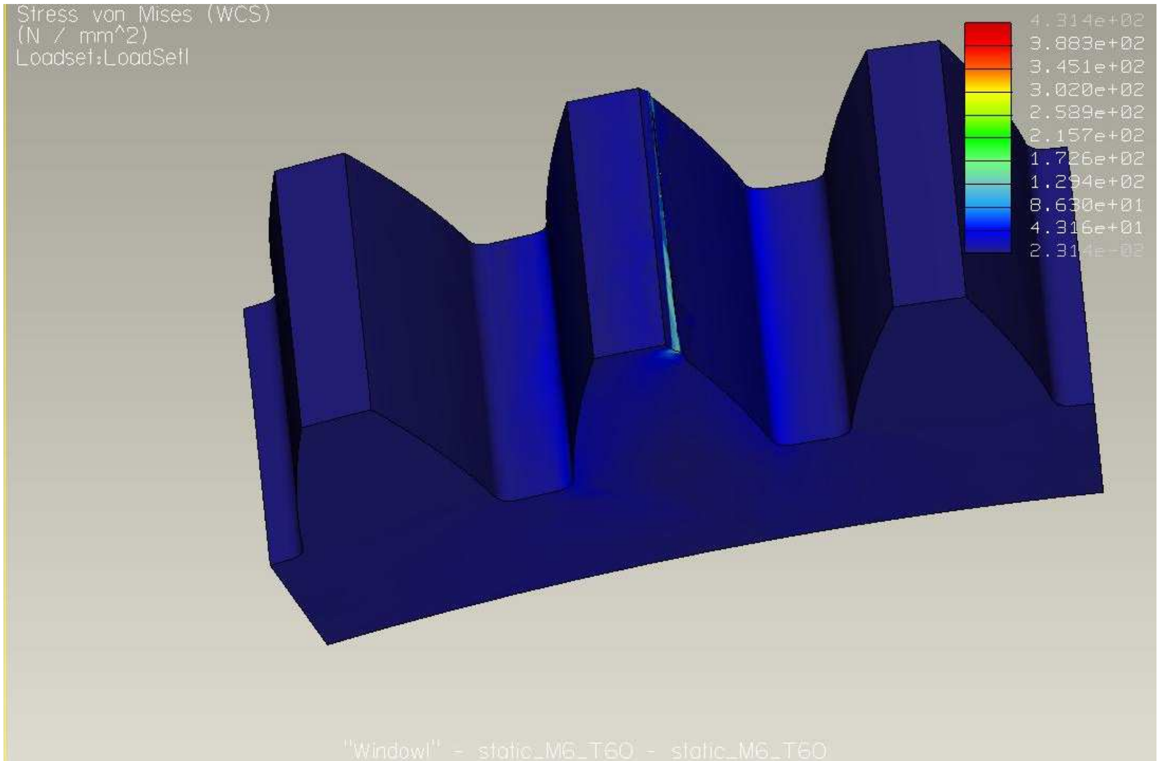


Figure 4.19 Stress variations in Gear6

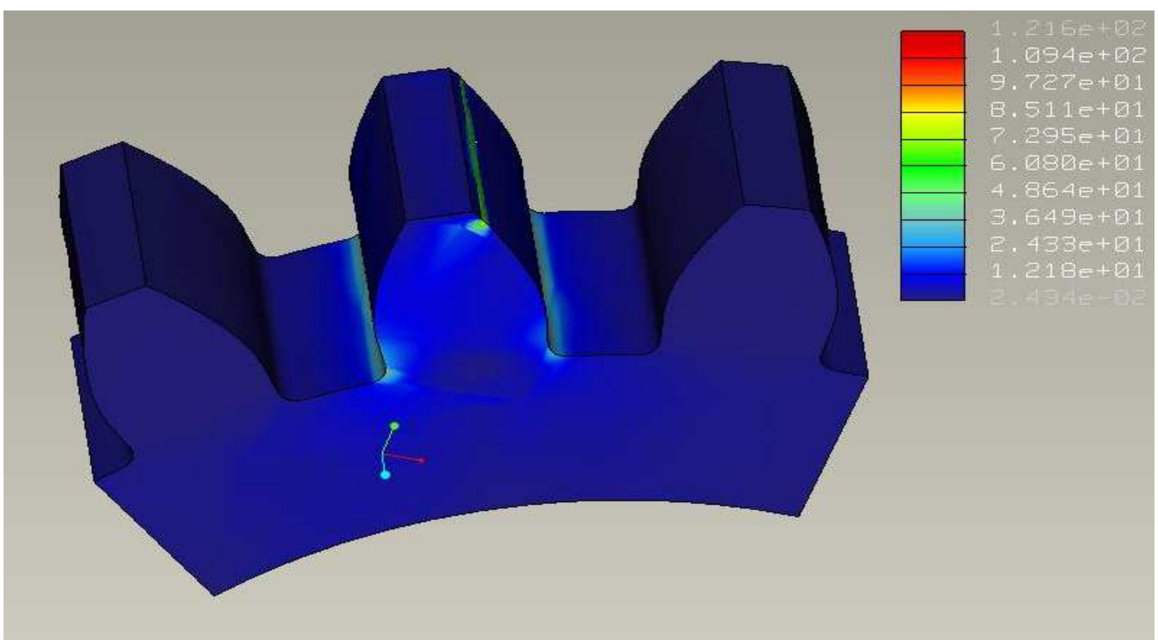


Figure 4.20 Stress variations in Gear 7

Conclusion: the maximum stress at the fillets for each gear is as below:

		Maximum stress at the fillets	Maximum deflection (mm)
Gear1	**M=2 T=25**	**168M Pa**	**0.00598**
Gear 2	M=2 T=50	146 M Pa	0.00552
Gear 3	M=5 T=18	135 M Pa	0.00394
Gear 4	M=5 T=70	126 M Pa	0.00461
Gear 5	M=6 T=18	146 M Pa	0.00634
Gear 6	M=6 T=60	137 M Pa	0.00466
Gear 7	M=7 T=20	127 M Pa	0.00632
Gear 8	M=7 T=50	116 M Pa	0.00464

Gear Number 1 m=2, T=25 with the maximum stress of 168 M Pa is chosen for stress relief studies. The aim is to reduce the stress and maintain stiffness.

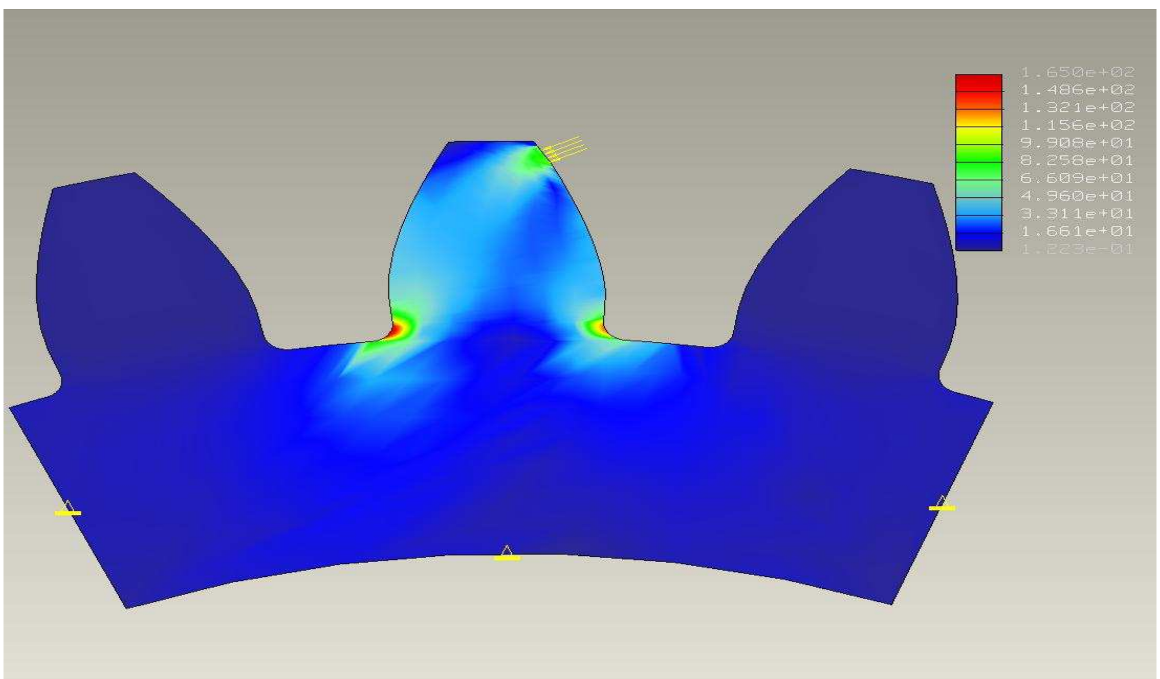


Fig 4.14 b Stress variation in fillets

As seen in Fig 4.14 b above the maximum stress occurs at the fillets. Of the two fillets the trailing fillet, on the left, is more stressed. The local stress at contact is not considered for further studies as the gear surface treatments make it resistant to these stresses and this point of contact constantly shifts.

STUDY 4

The fillets regions in the tooth profile, which are the most stressed, are made as the feature of focus. The selected gear is tested and the trailing fillet which has maximum stress is chosen for stress relief in this study.

The point of application of force is at H diameter. From study 3 the spur gear m=2,T=25 having base circle greater than dedendum circle is taken and stress relief hole feature is/are made along the dedendum, root-fillet, flank and involute profiles. Global sensitivity analysis is done taking the maximum stress at the fillets as a measure and, by variation location of hole, offset from the profile, diameter of the hole and multiple holes, these are also studied on either / both sides of the tooth. Finally an Optimization study is run on the basis of the results of the sensitivity analysis to find a optimized location and size of the hole(s).

a) Varying Location

In this study stress relief hole feature is made in the root fillet region. The study is done by varying the location of the hole along four regions, a. parallel to dedendum, b. along fillet, c. along radial flank and d. along involute profile. The following graphs show these locations. The diameter of the hole is 0.2 mm and it is at an offset distance of 0.5 mm from the edge of the gear tooth. The hole is placed parametrically with respect to the curve. In the curve 1 staring point of the curve is on the right side and ending point is on the left side of the curve. For the curve 2 starting point is the bottom right of the curve and ending point is top left of the curve. In the curve 3 bottom point is the starting point and top is the ending point. For the curve 4 the starting point is the bottom point and ending point is at the top end of the curve. In the curve 5 left end point is the starting point and right end point of the curve is the ending point. For the curve 6 bottom point of the curve is the starting point and the top point is the ending point. For the curve 7 bottom point is the starting point and top the curve is the end point. In the curve 8 bottom left point is the start point and top right is the end point.

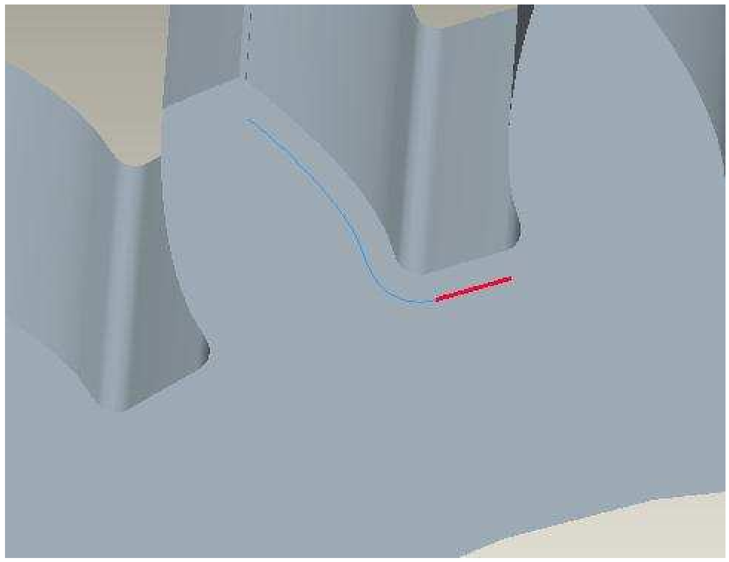


Fig 4.21 Curve 1

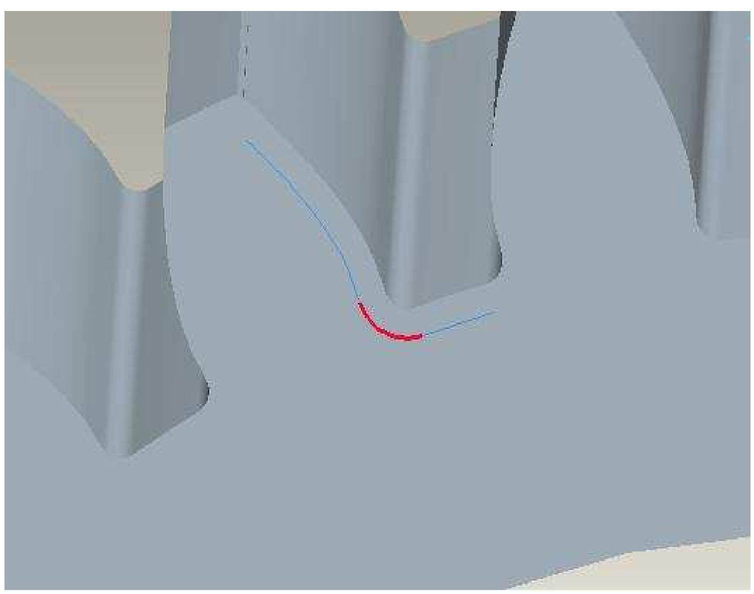


Fig 4.22 Curve2

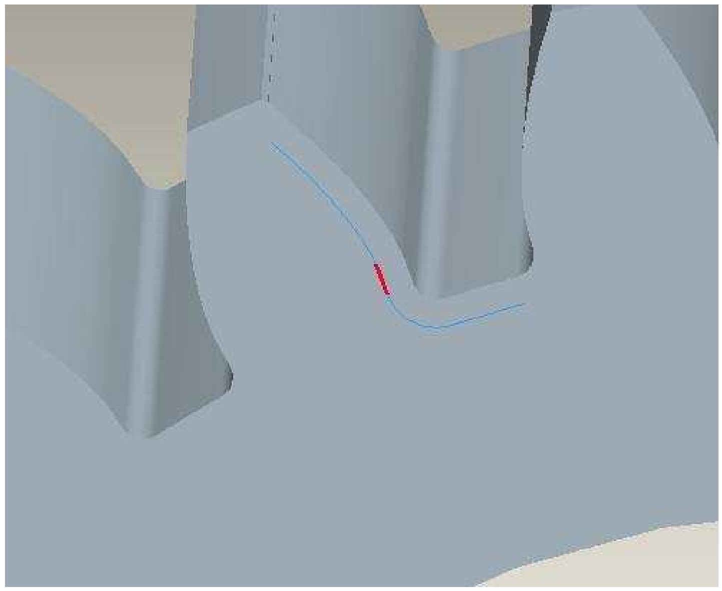


Fig 4.23 Curve 3

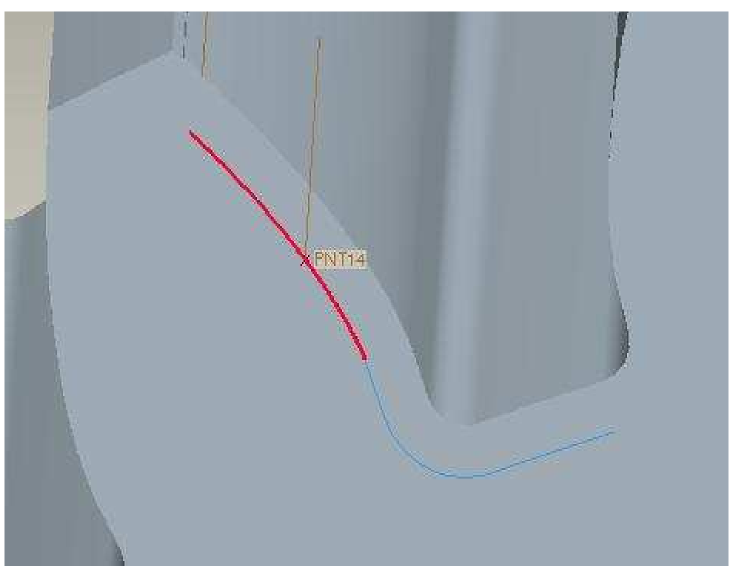


Figure 4.24 Curve 4

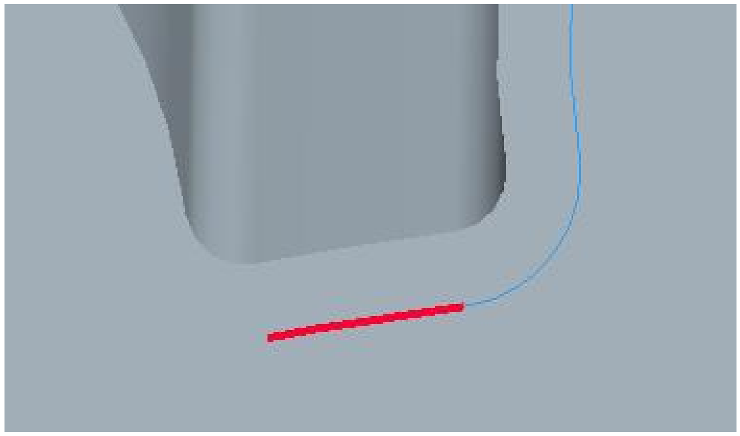


Figure 4.25 Curve 5

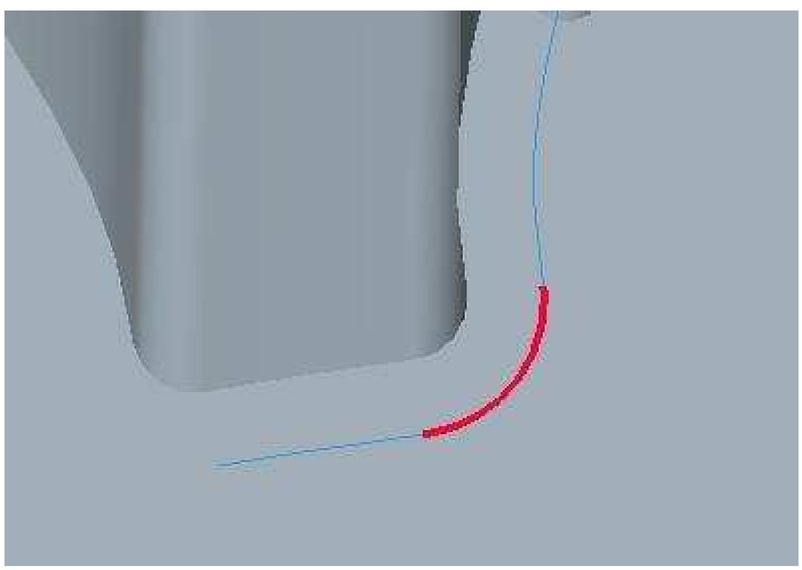


Figure 4.26 Curve 6

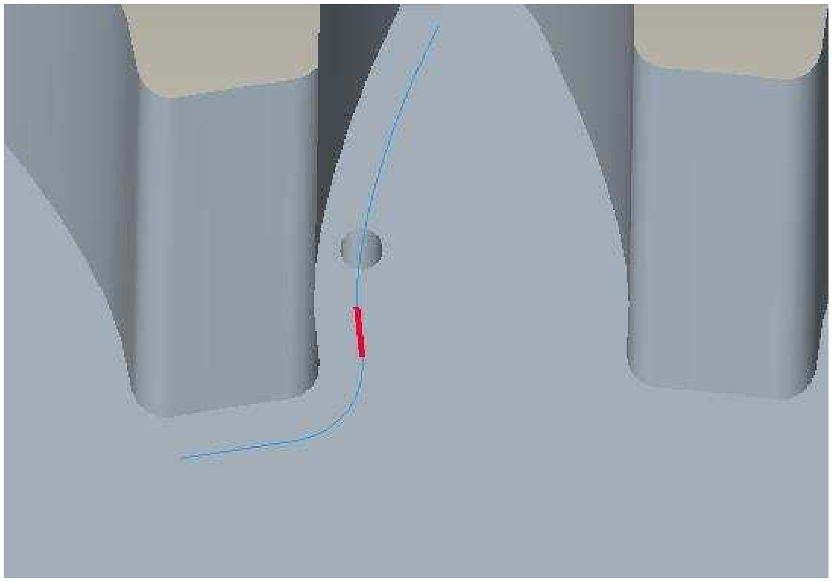


Figure 4.27 Curve 7

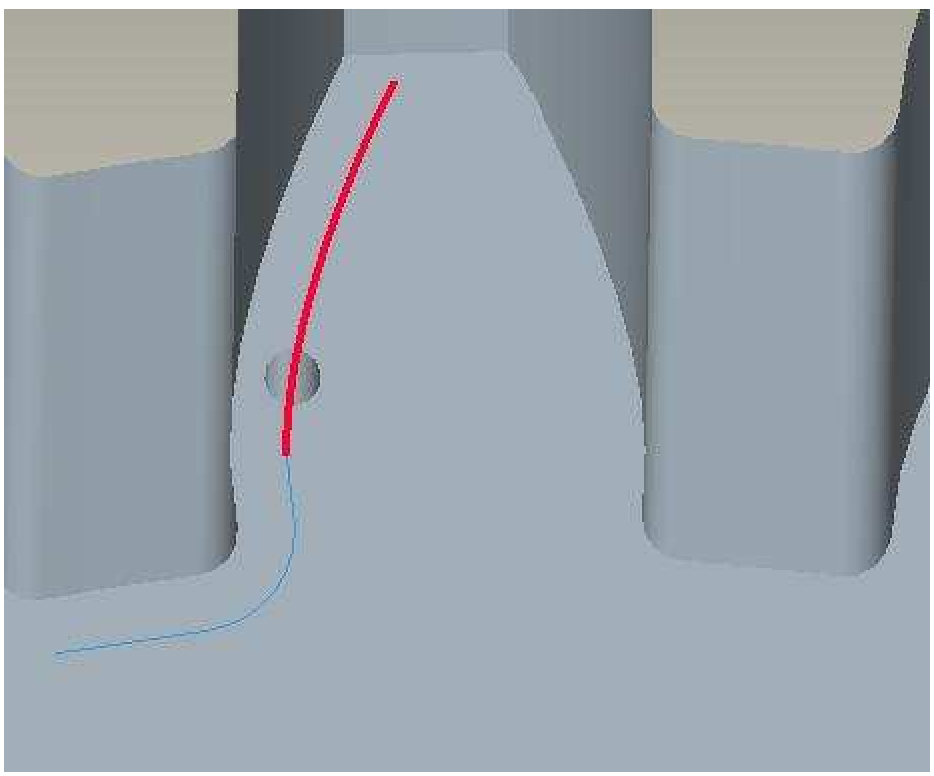


Figure 4.28 Curve 8

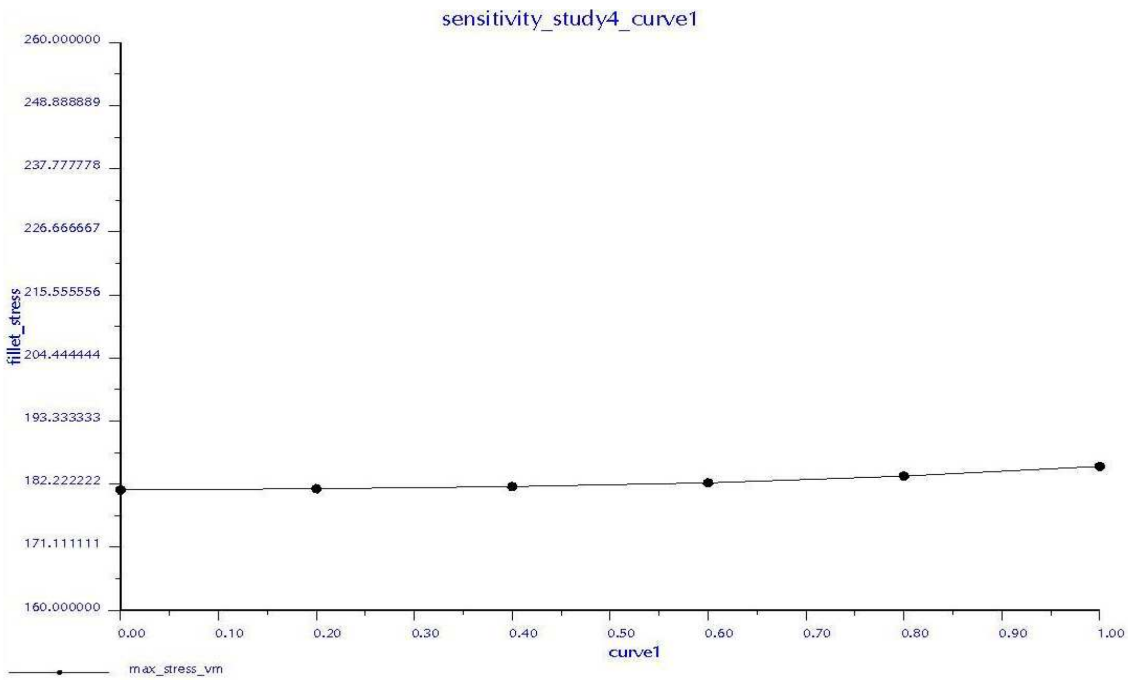


Figure 4.29 Graph showing sensitivity analysis of hole on curve 1

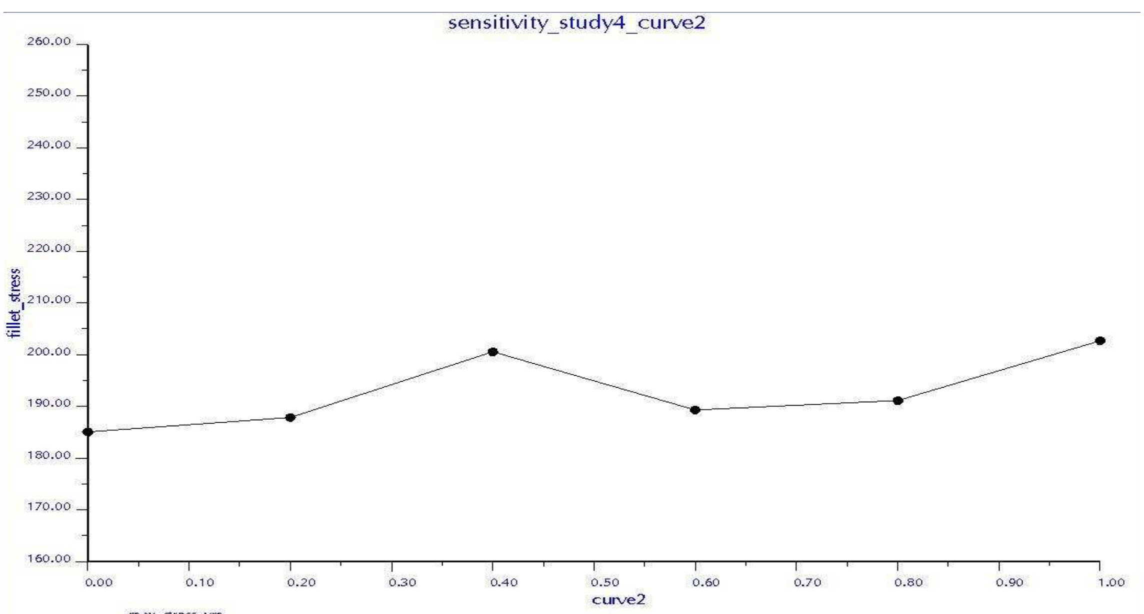


Figure 4.30 Graph showing sensitivity analysis of hole on curve 2

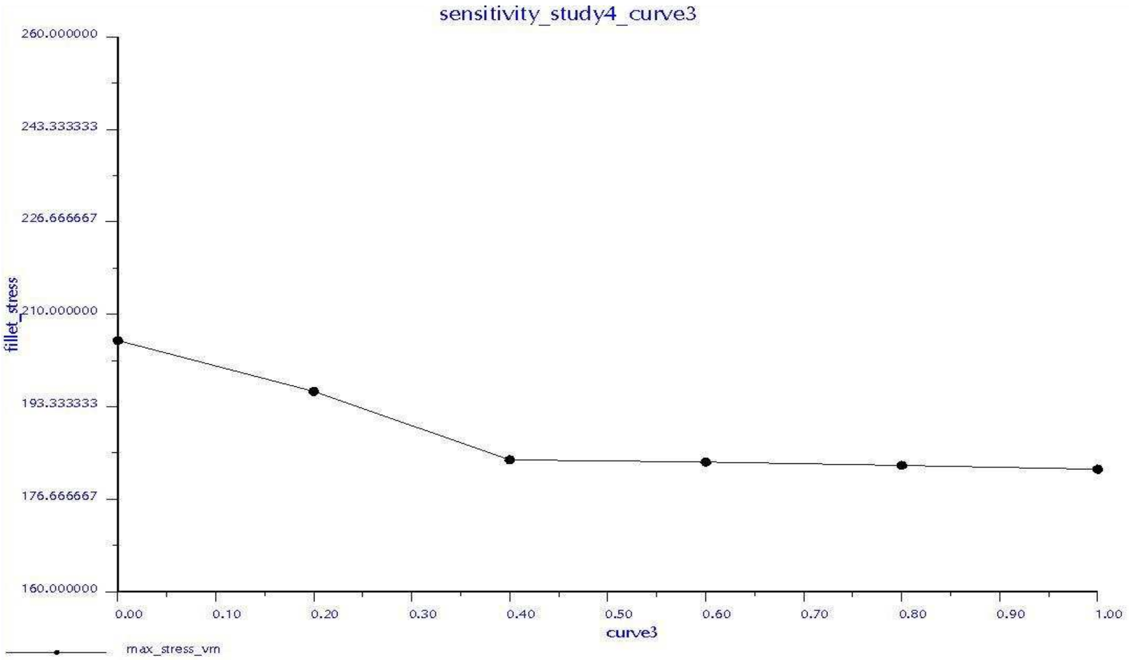


Figure 4.31 Graph showing sensitivity analysis of hole on curve 3

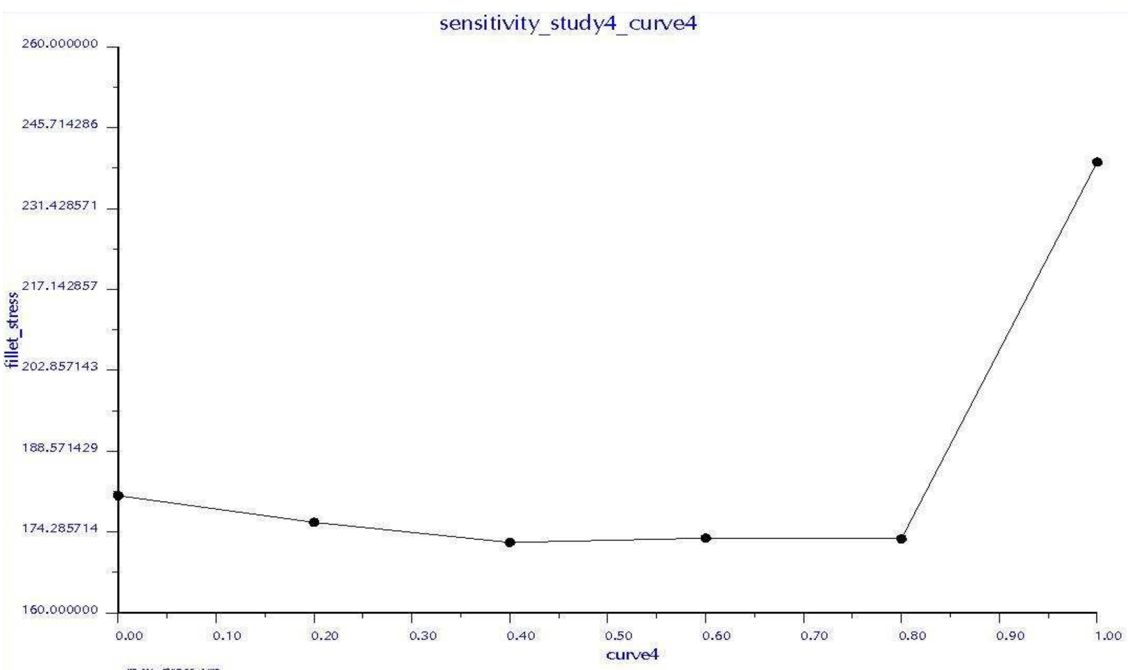


Figure 4.32 Graph showing sensitivity analysis of hole on curve 4

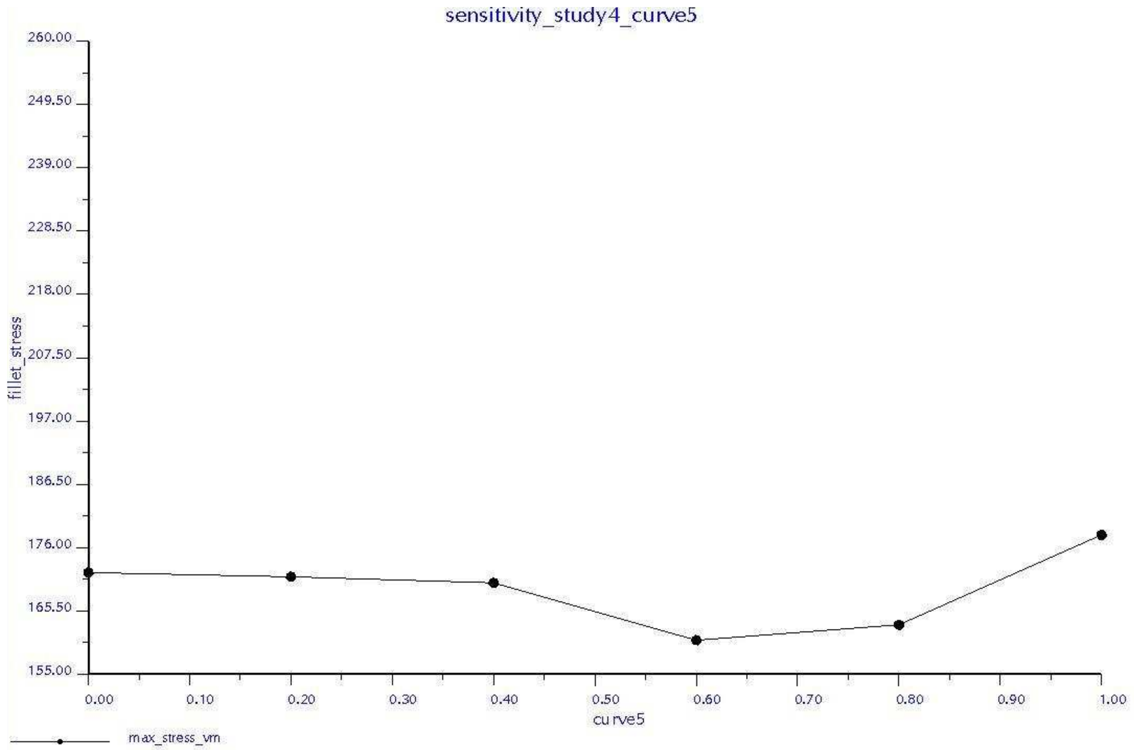


Figure 4.33 Graph showing sensitivity analysis of hole on curve 5

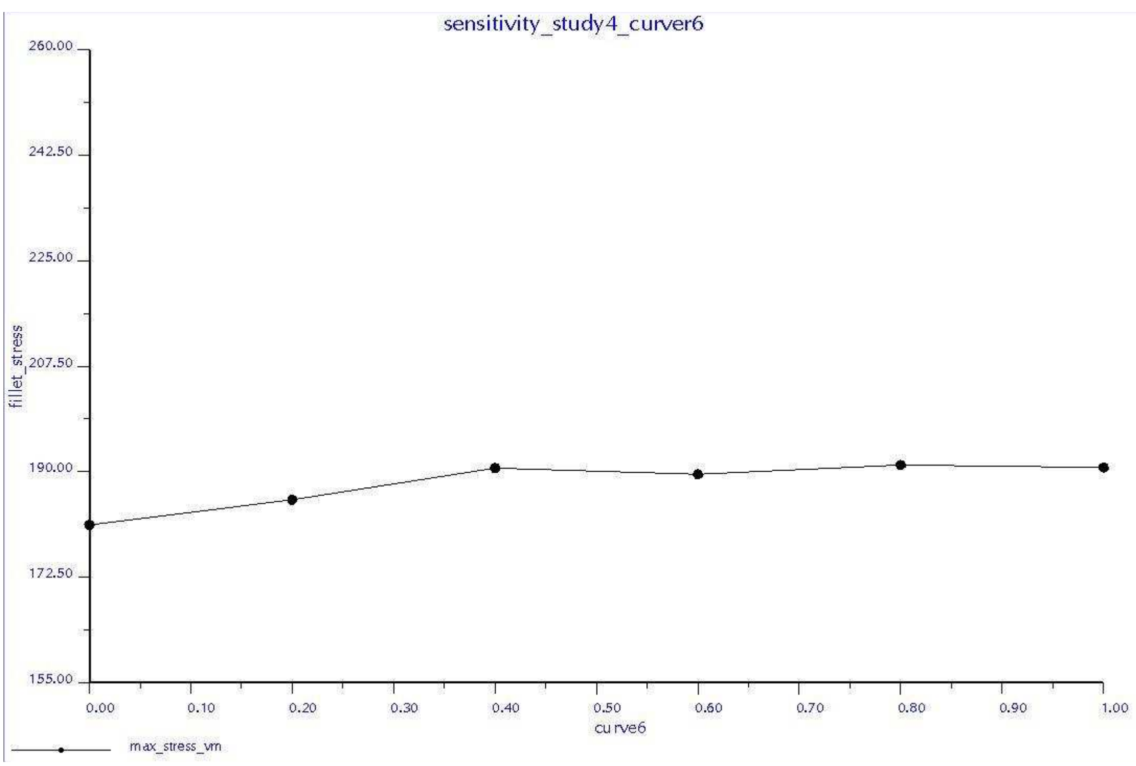


Figure 4.34 Graph showing sensitivity analysis of hole on curve 6

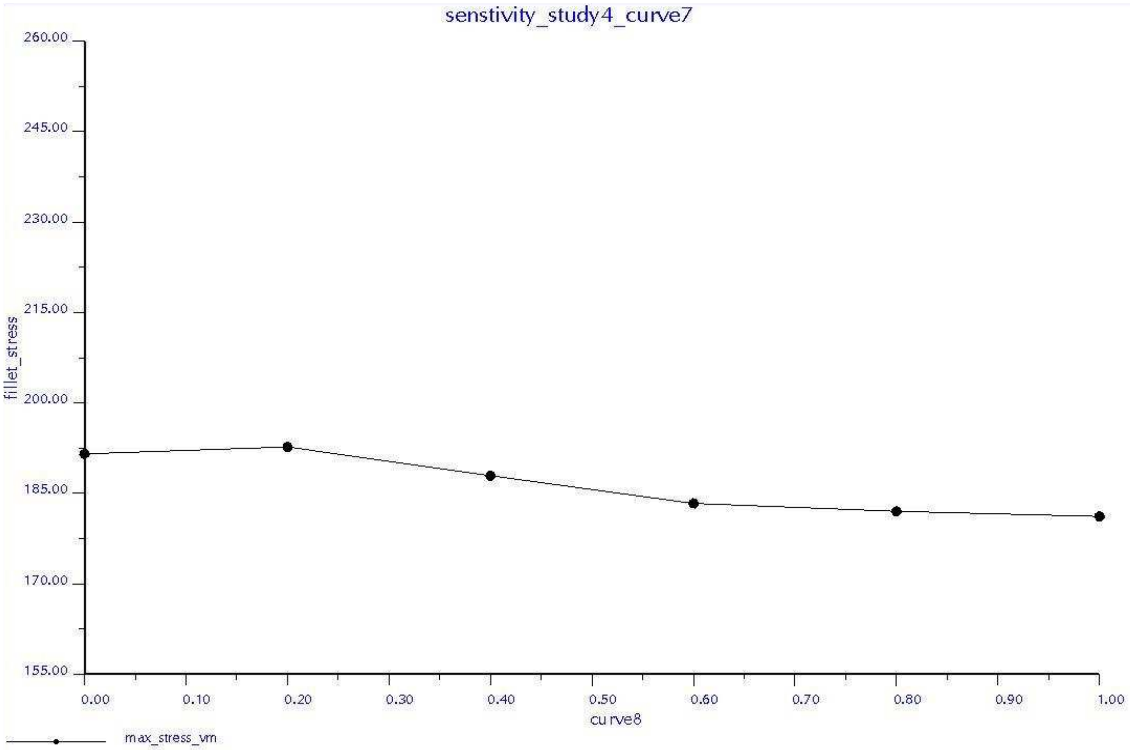


Figure 4.35 Graph showing sensitivity analysis of hole on curve 7

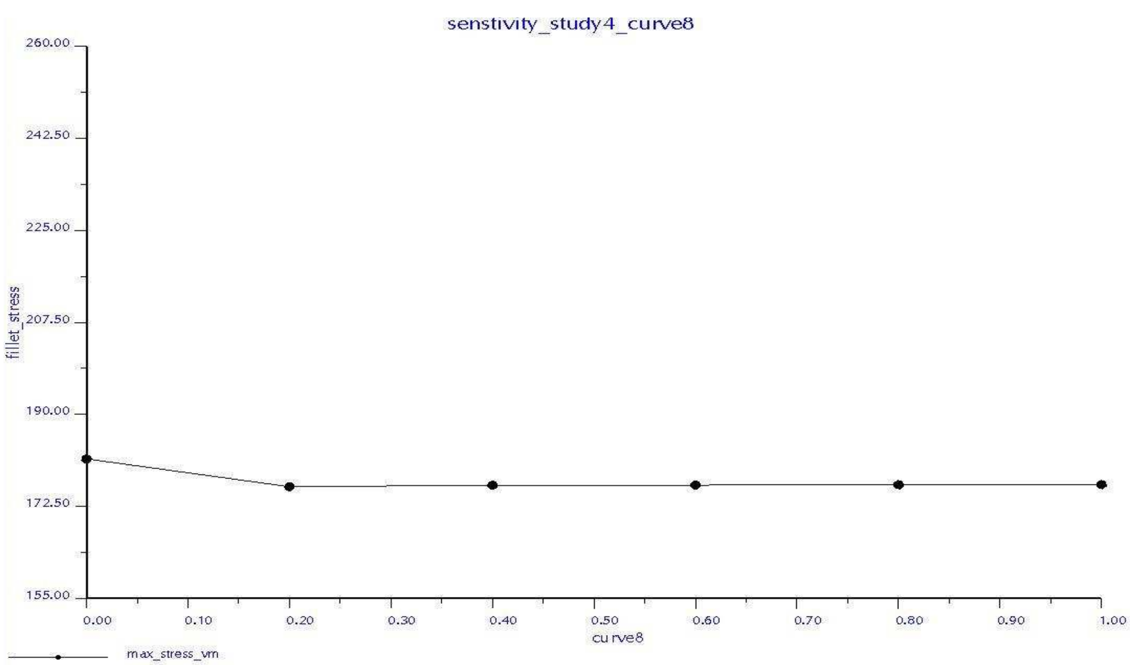


Figure 4.36 Graph showing sensitivity analysis of hole on curve 8

b) Varying Offset

As can be seen from the graphs of sensitivity above, the stress levels at the fillets are the least at roughly the middle of curves 1, and curve 5, while it is least at a parametric distance of 0.4 for curve 4 and curve 8. The stress due to force at contact point is not measured. The location of the hole at curve 1 and 5 has to be the same due to symmetry in manufacturing the gear teeth.

The sensitivity studies for effect of offset / distance of hole center from the profile edge are now done for curves 1, 4, 5 and 8. The size of the hole is taken as 0.2 mm.

The next graph in Fig 4.37 is for curve 1 with hole in the middle of the curve.

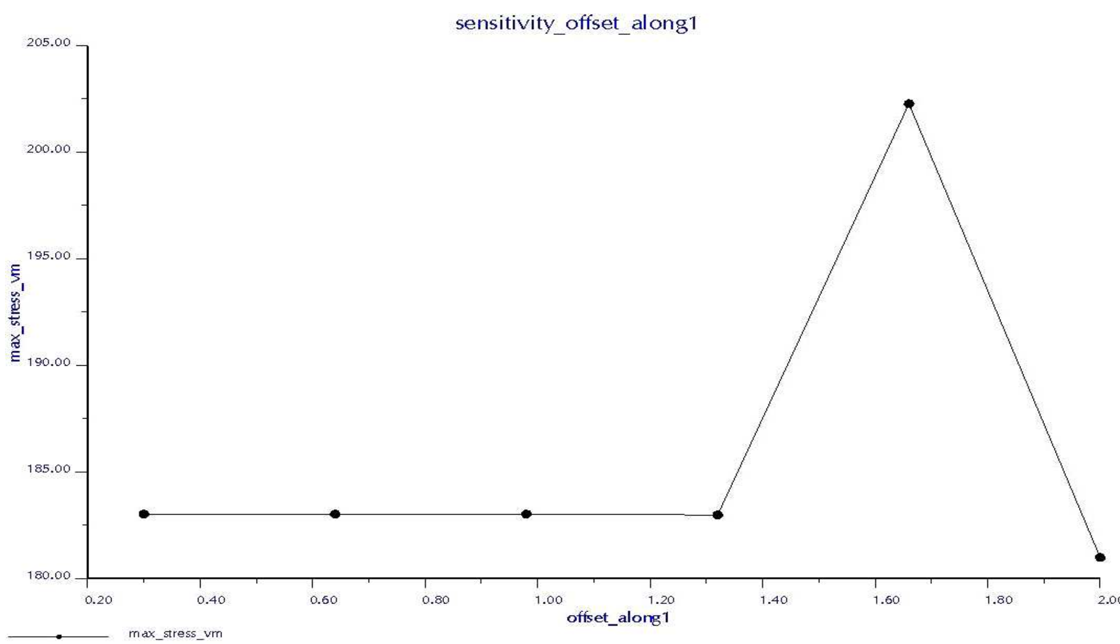


Figure 4.37 Variation of offset on curve 1

As is seen above the stress levels are least for offset up to 1.3 approximately. Thus, taking a material of at least 0.4 mm from the hole to the profile edge the offset for the next study is kept at least at 0.5mm away from the profile edge.

The next graph Fig 4.38 is showing sensitivity of stress at fillets to offset of hole at 0.4 parametric distance on curve 4. The offset is varied from 0.5 to the center of the tooth, an offset of circular pitch / 4.

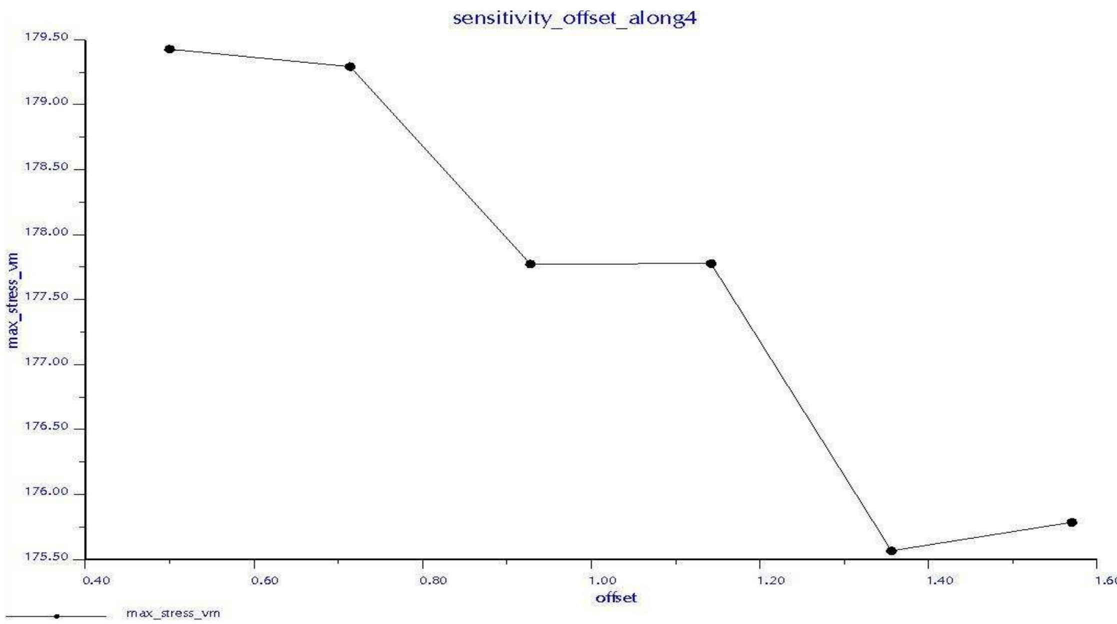


Figure 4.38 Variation of offset on curve 4

The graph in Fig 4.38 shows an offset of about 1.35mm away from the profile edge results in the least stress at the fillets. As this value is very close to the offset of the tooth center from the profile edge it is advisable to take the offset for further studies at 1.57mm which places the hole on the central axis of the tooth. The next graph in Fig 4.39 is for curve 5.

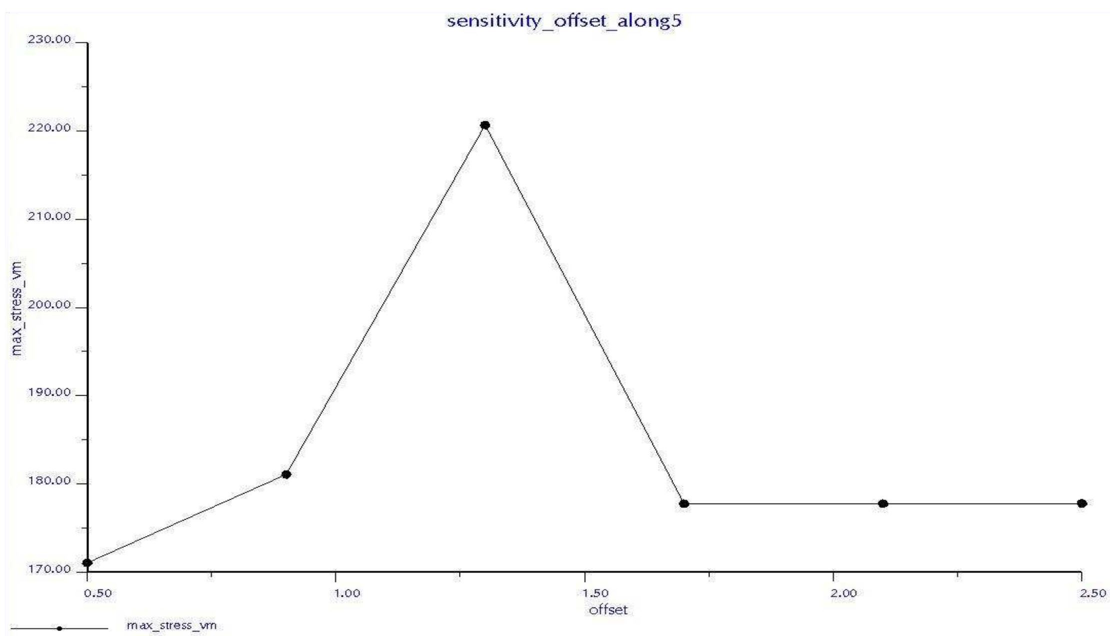


Figure 4.39 (a) Variation of offset on curve 5

The size of the hole is 0.2 mm diameter and is placed at mid point of the curve. As there is a sudden increase in the stresses for offset increasing above 0.5mm another study is done for offset less than 0.5mm as shown below in Fig 4.39 b.

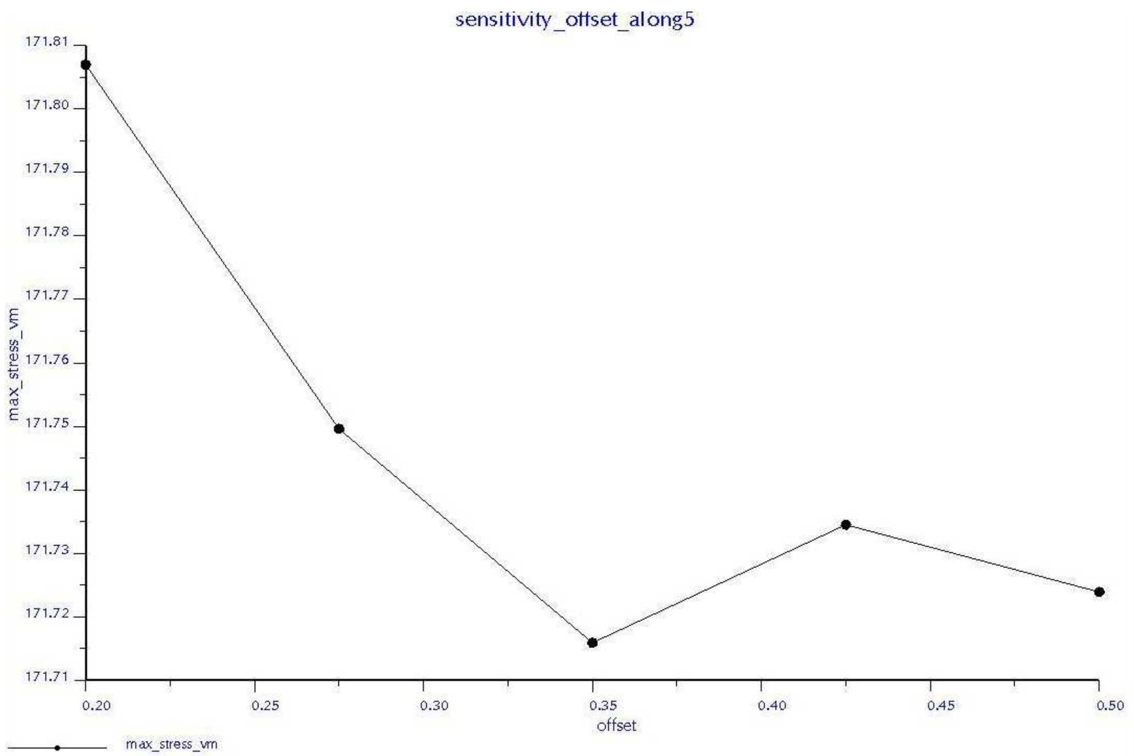


Figure 4.39 (b) Variation of offset < 0.5mm on curve 5

As the stress is not sensitive to offset less than 0.5mm, and taking a material of at least 0.4 mm from the hole to the profile edge the offset of hole center for the next study is kept at least at 0.5mm away from the profile edge.

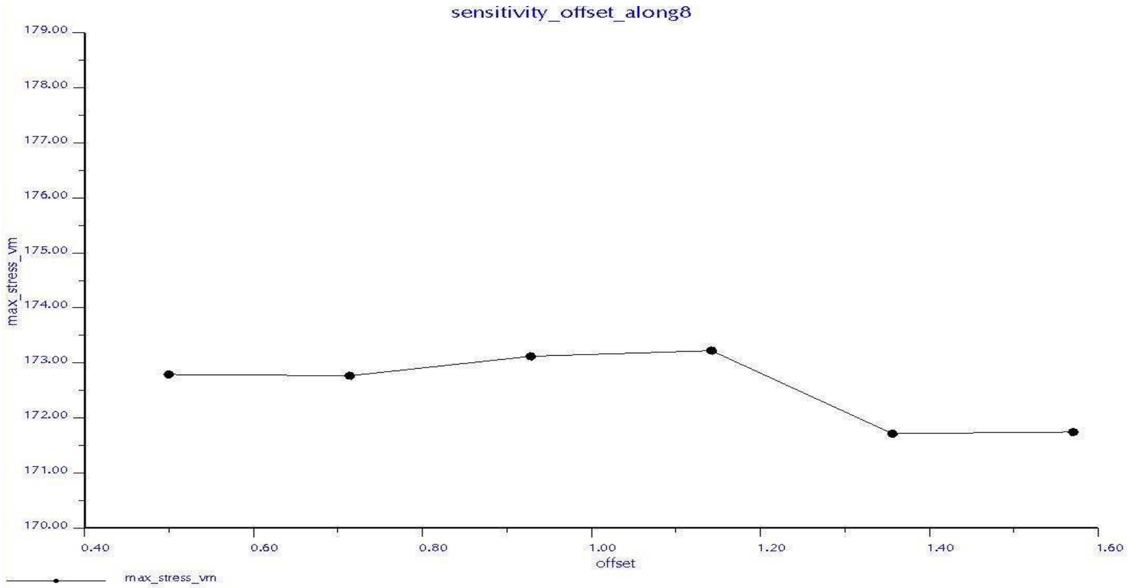


Figure 4.40 Variation of offset on curve 8

In the Fig 4.40 above for the curve 8, size of hole is taken as 0.2mm diameter and is placed 0.4 from start of the curve. As the trend is similar to that for curve 4 from Fig 4.38 the same result and conclusion holds good.

c) Varying Size

The dia of the holes is varied, and is placed at 0.5 distance from the start of the curve 1 & 5. The offset of the hole center is also varied so that the hole is always leaving a material of 0.4mm from the edge of the profile (dedendum circle).

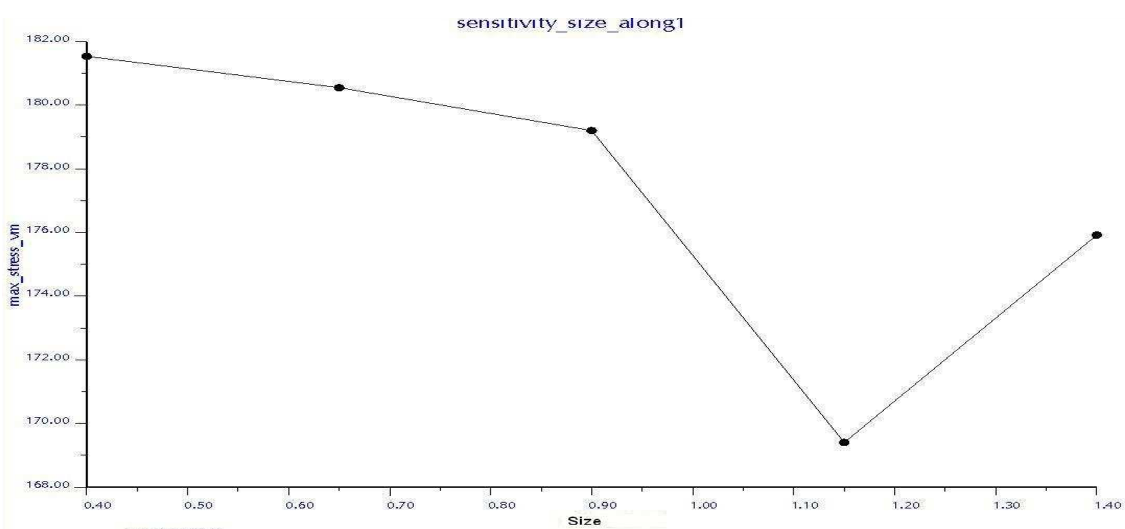


Figure 4.41 Variation of diameter on curve 1&5

From the above graph we can conclude that the hole size increases the stresses at the fillets decreases till a minima around a diameter of 1.15mm.

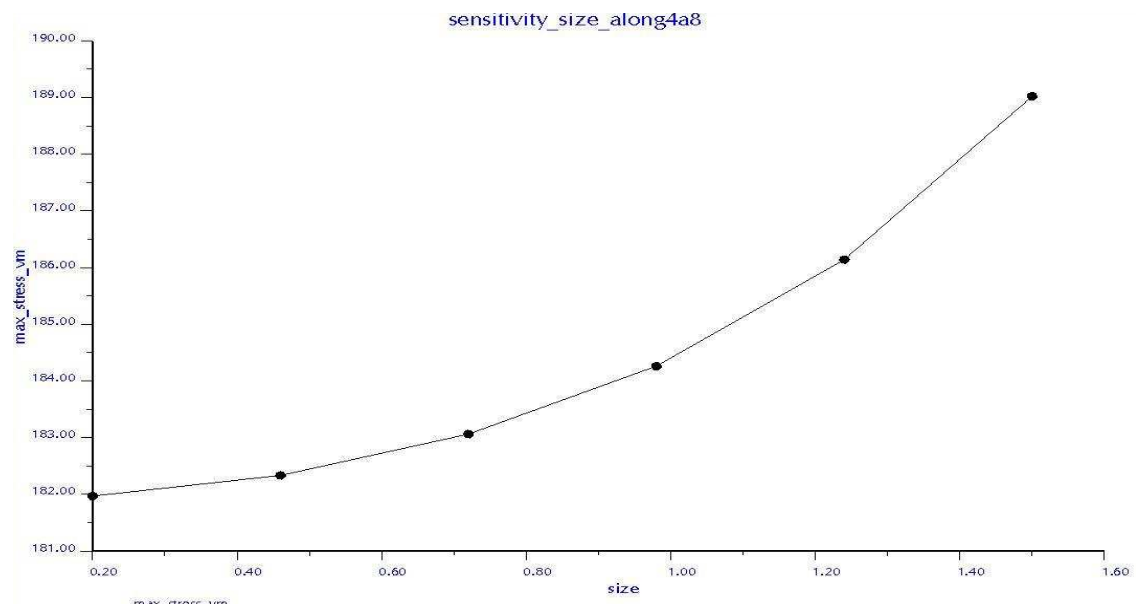


Figure 4.42 Graph showing effect of variation of hole size at tooth center

Here the hole is placed on the center of the tooth, and only diameter is varied from 0.2 to 1.5 mm. It is seen that there is a steady increase in the stresses in the fillets when the diameter of the hole is increased.

It can be concluded from this that the presence of a hole feature in the tooth above the dedendum does not help in any stress reduction in the fillets, rather it increases the stresses beyond the levels noted for the static analysis of the gear without any hole in study 3. Thus for further Optimization studies we would not take any hole within the tooth.

d) Variation of number of holes

In this the single large hole of study 4 (c) is replace by a pattern of smaller holes of dia 0.3mm, axis of pattern is at an offset of 0.67mm, from study 4 (c).

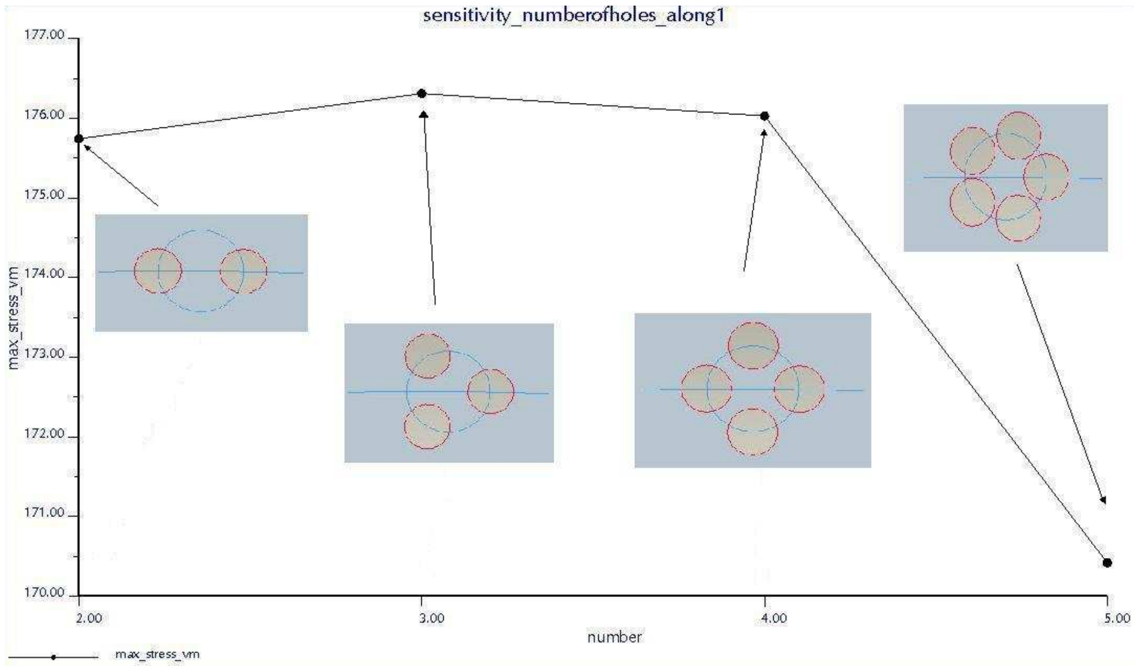


Figure 4.43 Variation along curve 1 &5

In the above plot there is a variation of number of holes 2, 3, 4 and 5 filling a circle of dia 1.15 taken from Study 4 (c), where the min stress occurred. It is observed that the stress levels increased for 3 and 4 holes due to the orientation of the holes in the pattern and the max effect of the 5 holes to reduce stress was due to the similarity of the metal removal to the single hole of 1.15 dia case.

Thus, it is more beneficial to make a single large hole for stress reduction and ease in manufacturing.

Conclusions:-

From all these studies we find that stress is still higher than the stresses in the representative gear. Now finally we do an optimization study to find the possibility of reducing the stress below the Gear 1 without hole.

e) Optimization study

Two equal holes are taken on curve 1 & 5, as the holes would be manufactured at the same location relative to every tooth. The study of effect of hole placement along the curve concentric to dedendum, size and hole axis offset from dedendum for Gear 1 is done.

Parametric placement along curve 1 & 5.

A sensitivity study of the fillet stress to the parametric placement of holes on the curves 1&5 is done. Taking the dia of hole as 1.15mm from Figure 4.41 and keeping 0.4mm of uncut material from dedendum to the hole edge.

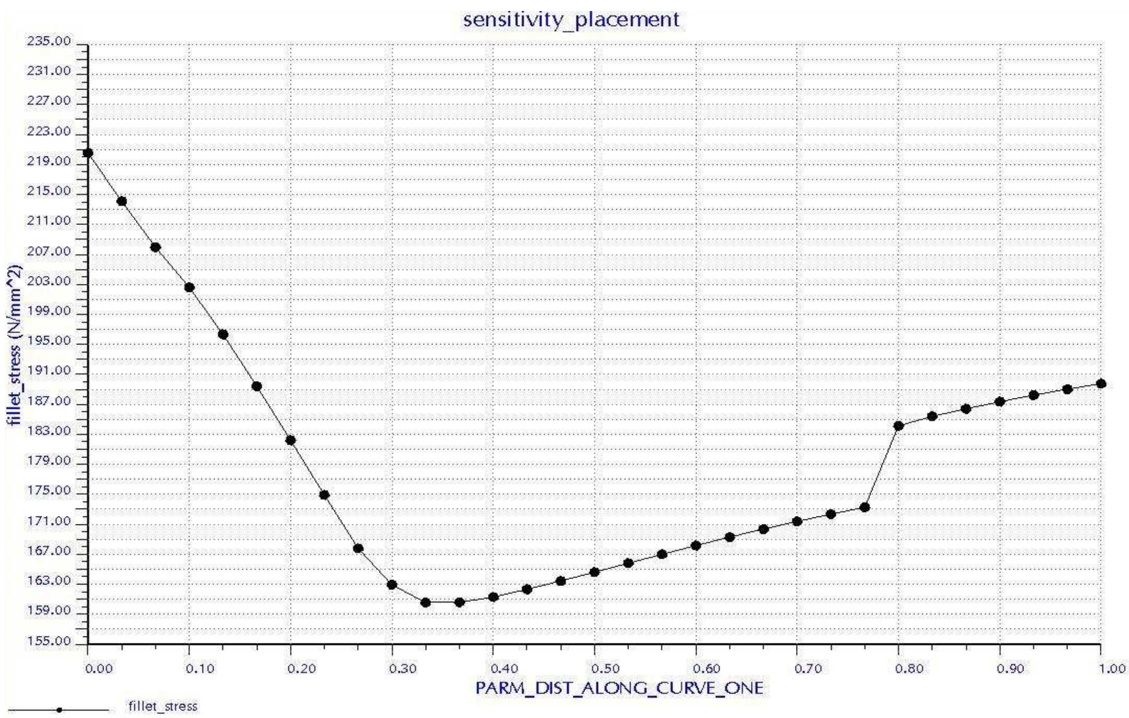


Figure 4.44 Fillet stress vs. parametric placement of holes on curves 1 / 5

From the above sensitivity analysis it is found that the minimum stress occurs at around 0.355 parametric distance from the start point of curve 1 / 5. This parametric distance is now taken for all further sensitivity analysis. From Study 3 the max von mises stress at fillets of 168 MPa and deflection of 0.00599mm are marked in the graphs for reference.

Hole axis offset from dedendum for a hole diameter

Following graphs show the sensitivity of tooth displacement and fillet stress to hole axis offset from dedendum variation. The starting is from min -ve offset when there is no cutting of the material by the hole, till a max offset of 2mm from dedendum when the hole is completely inside the gear material. Separate results of studies for hole dia of 0.2, 0.5, 0.8, 1.0, 1.2, 1.4, and 1.6 are presented.

The parametric placement of hole on curve was increased to avoid cutting the fillet. Using P_dist (mm) = sqrt(hole rad^2 - offset^2) +.02. when required.

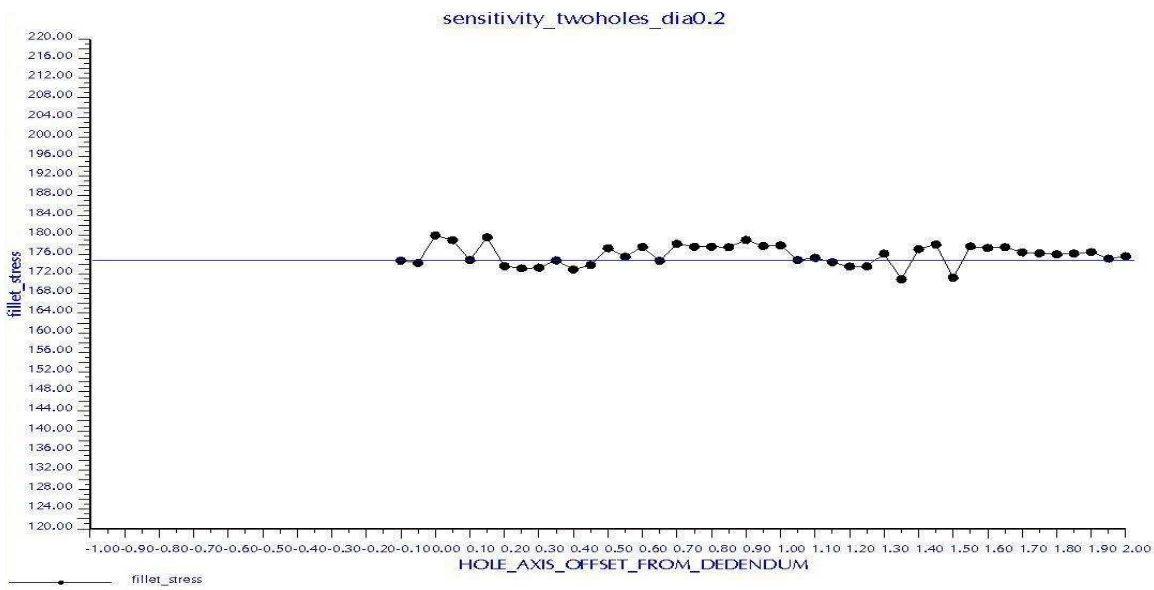


Fig 4.45 Graph of fillet stress vs. hole axis offset for dia 0.2 mm

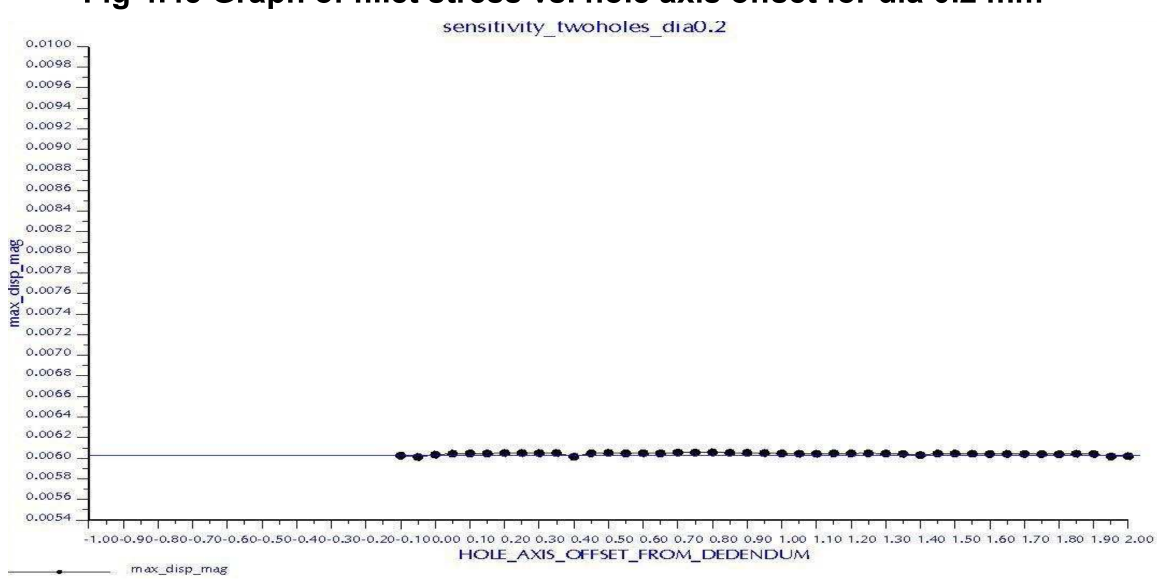


Fig 4.46 Graph of deflection vs. hole axis offset for dia 0.2 mm

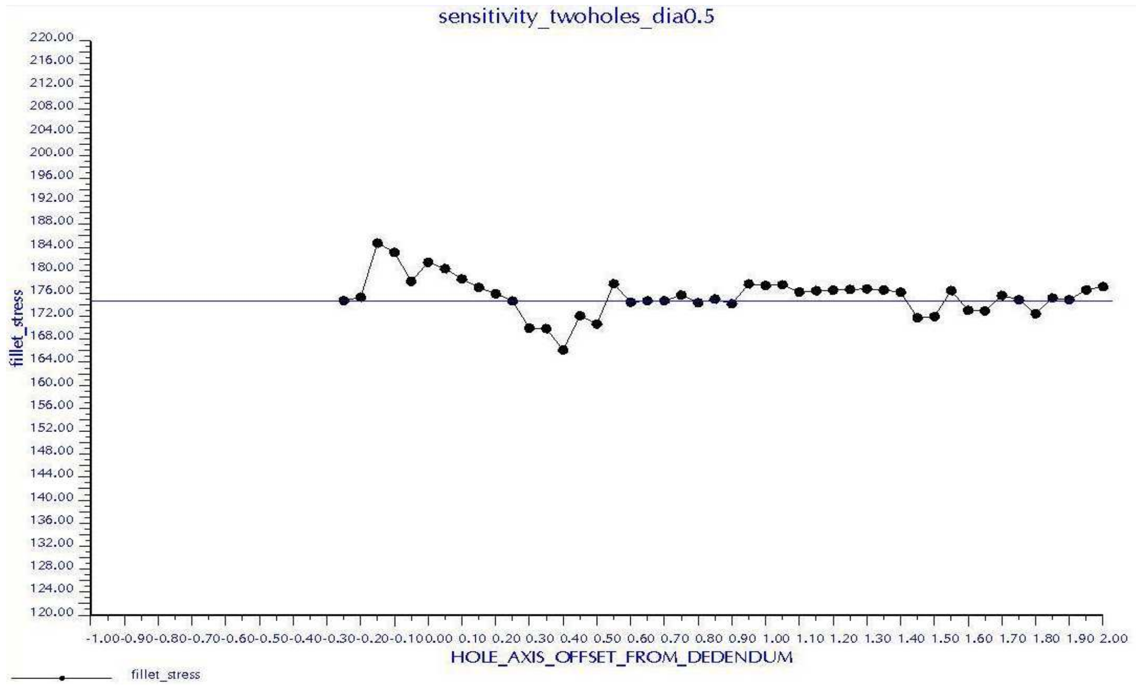


Fig 4.47 Graph of fillet stress vs. hole axis offset for dia 0.5

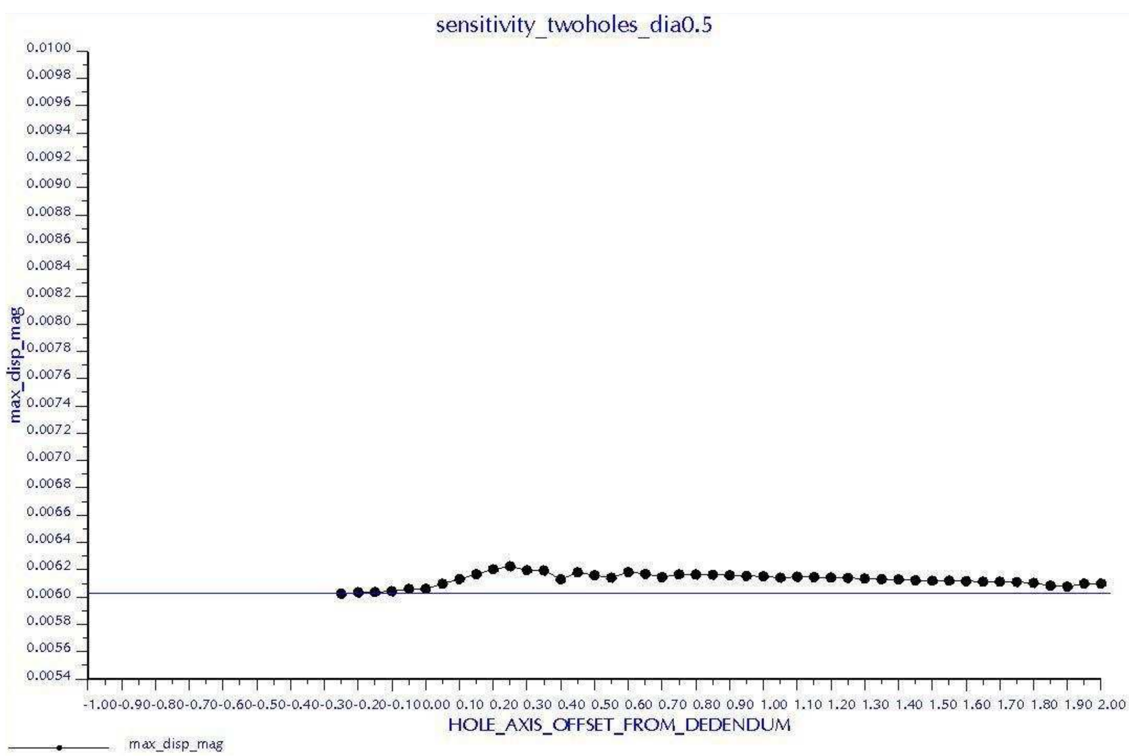


Fig 4.48 Graph of deflection vs. hole axis offset for dia 0.5 mm

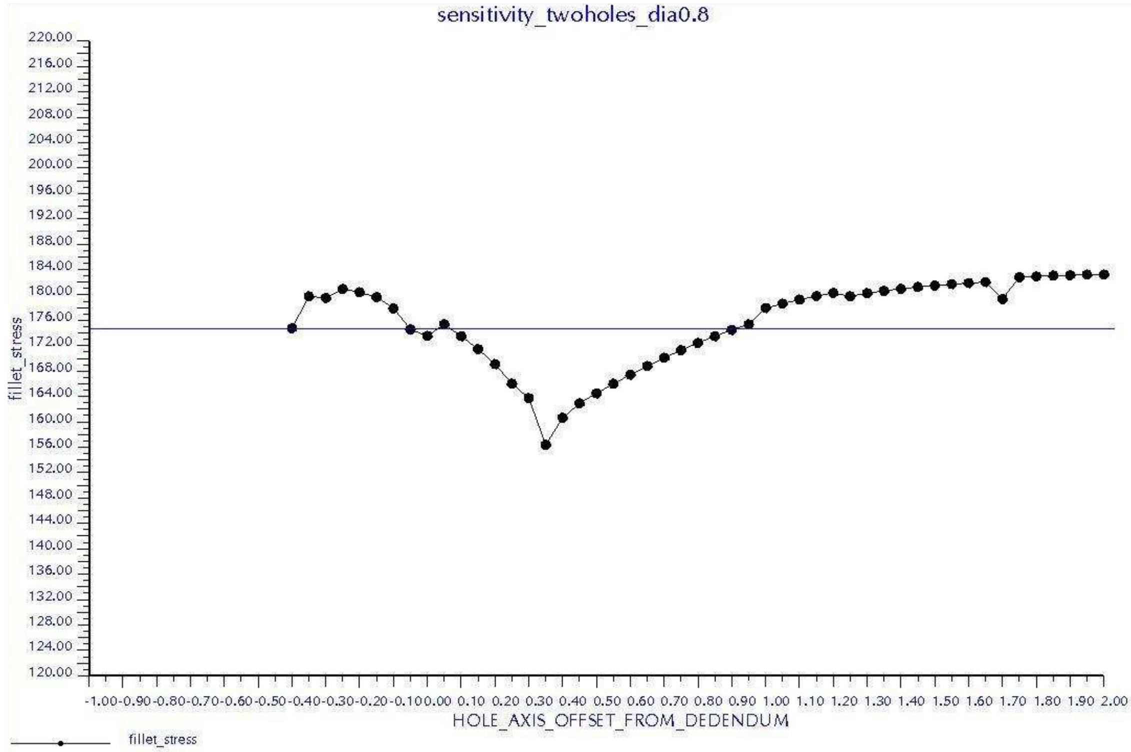


Fig 4.49 Graph of fillet stress vs. hole axis offset for dia 0.8 mm

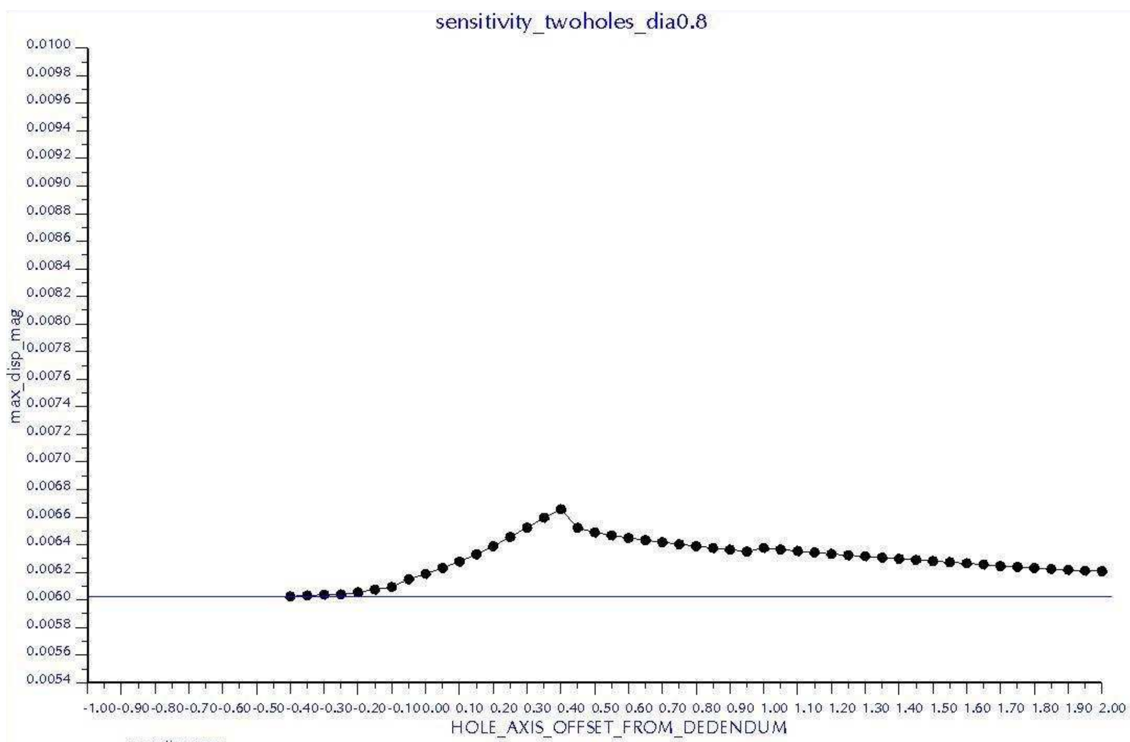


Fig 4.50 Graph of deflection vs. hole axis offset for dia 0.8 mm

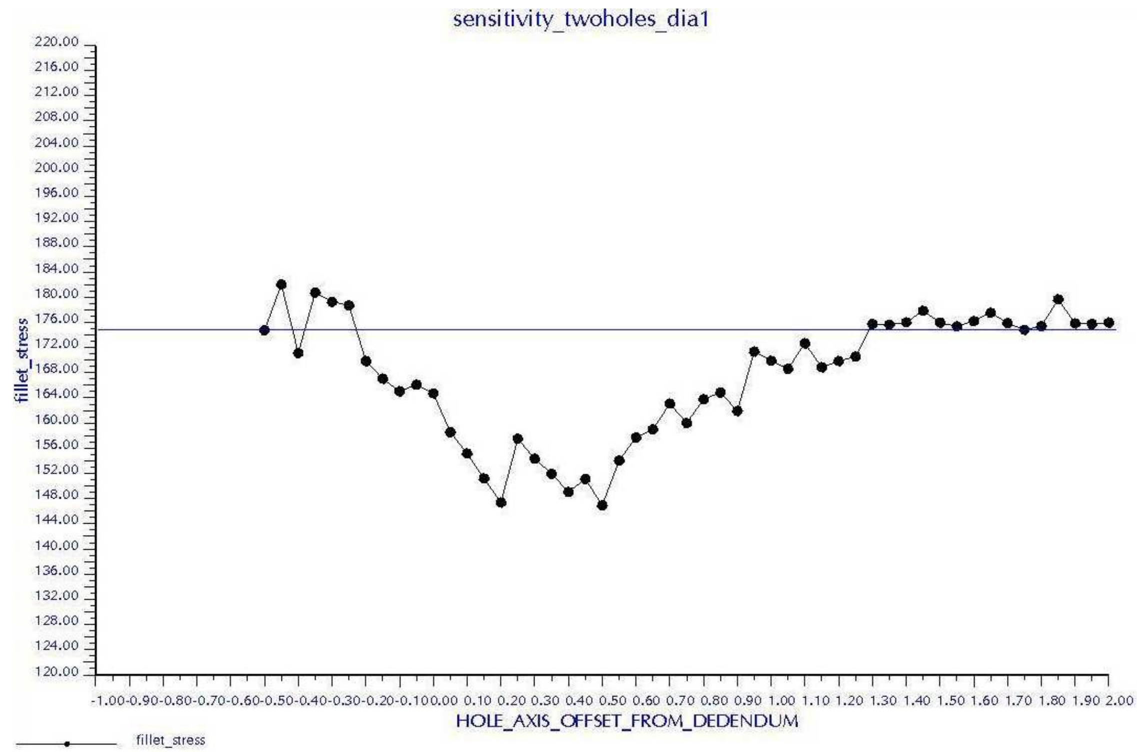


Fig 4.51 Graph of fillet stress vs. hole axis offset for dia 1 mm

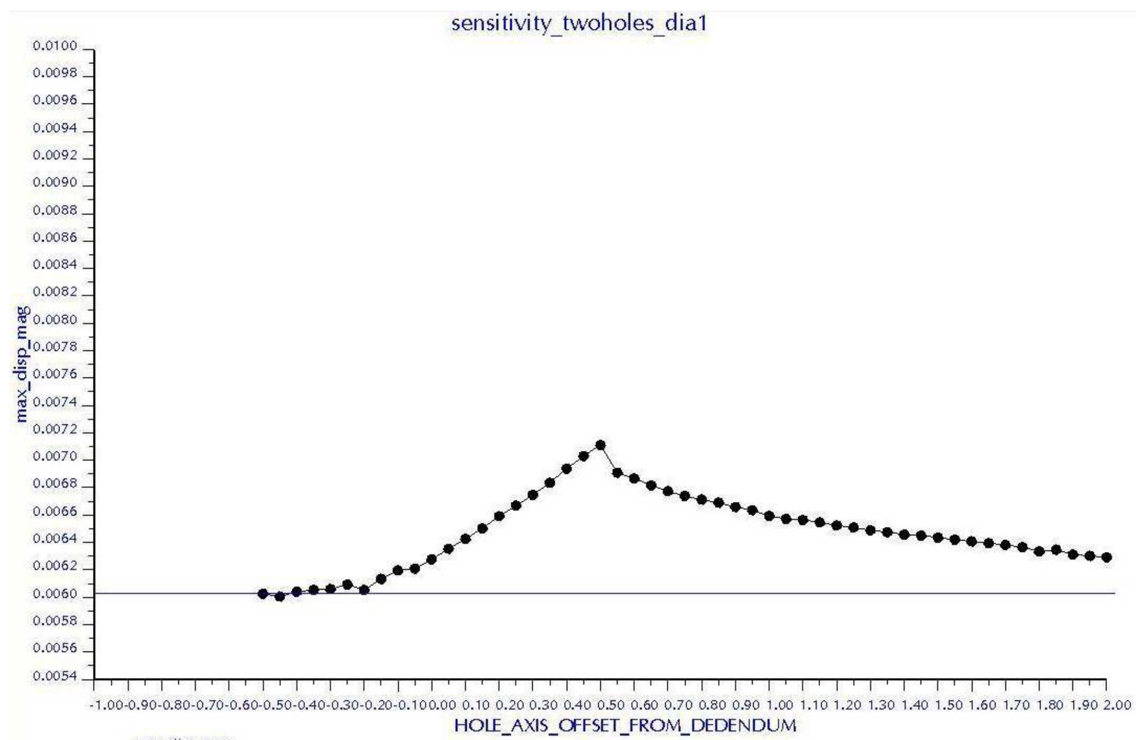


Fig 4.52 Graph of deflection vs. hole axis offset for dia 1 mm

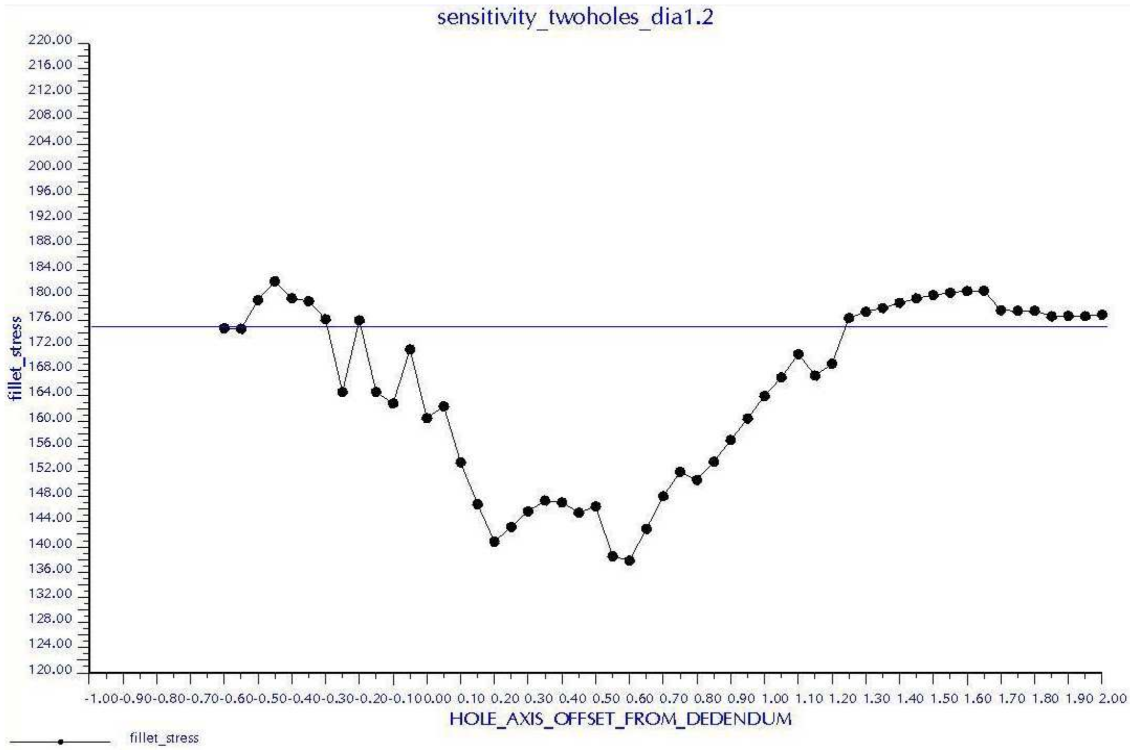


Fig 4.53 Graph of fillet stress vs. hole axis offset for dia 1.2 mm

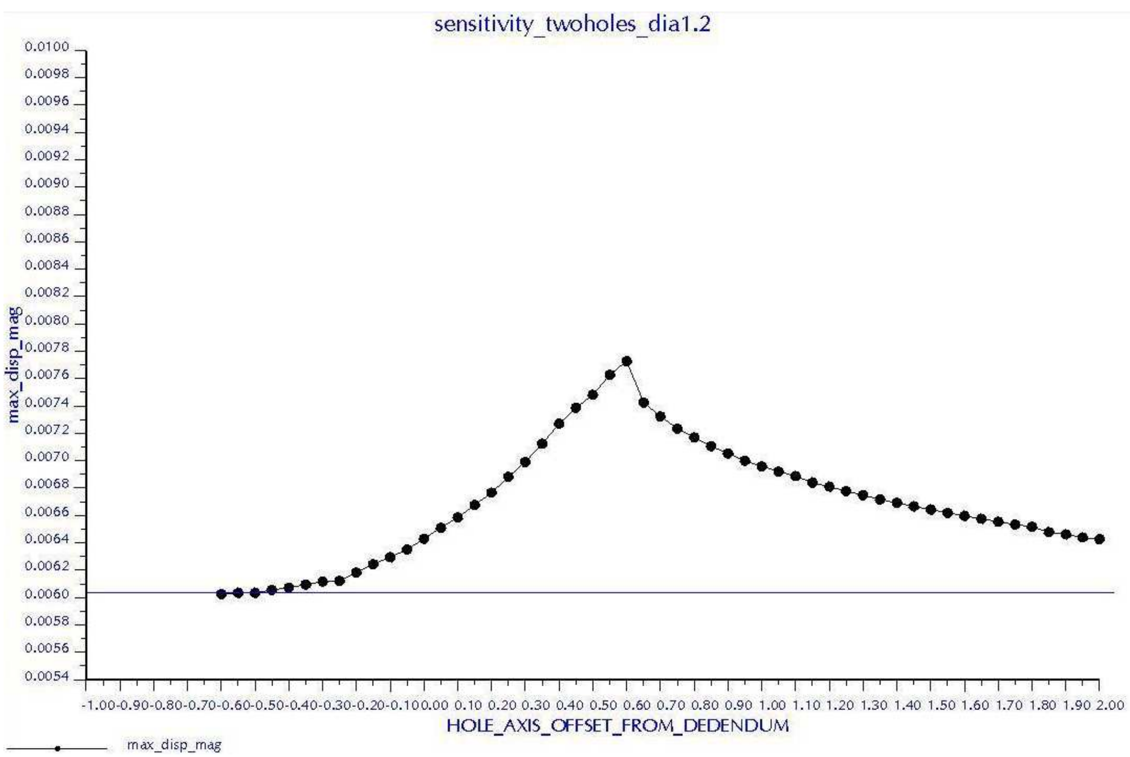


Fig 4.54 Graph of deflection vs. hole axis offset for dia 1.2 mm

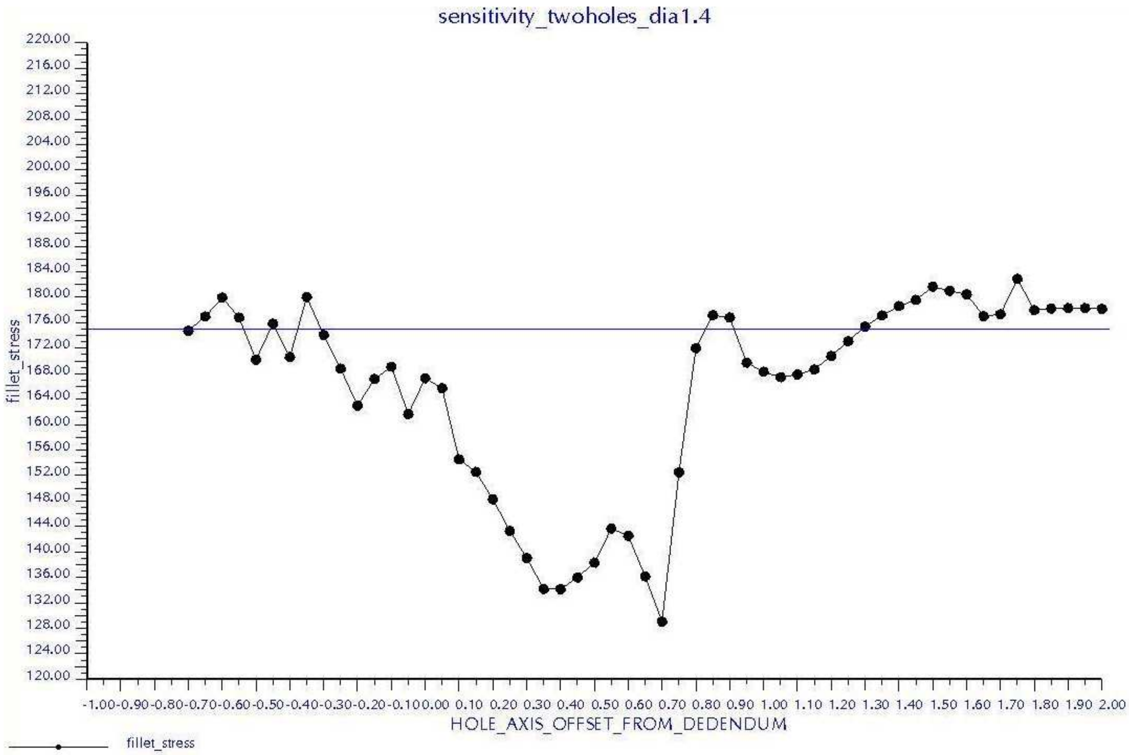


Fig 4.55 Graph of fillet stress vs. hole axis offset for dia 1.4 mm

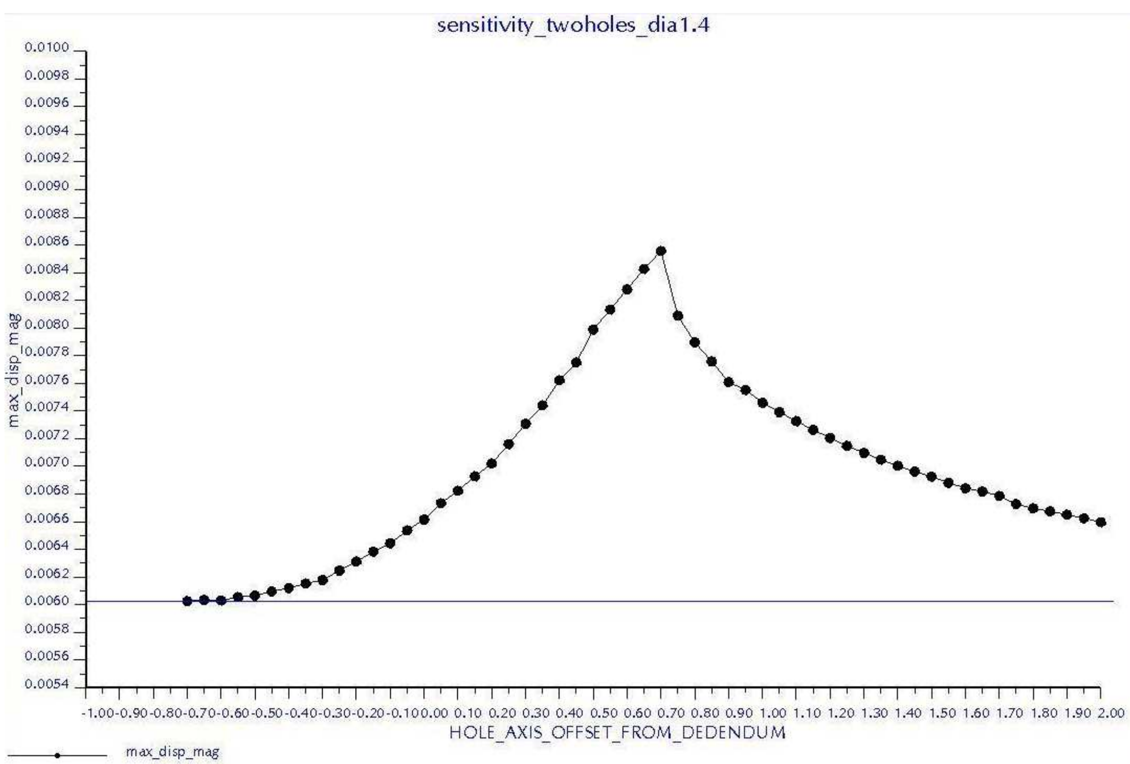


Fig 4.56 Graph of deflection vs. hole axis offset for dia 1.4 mm

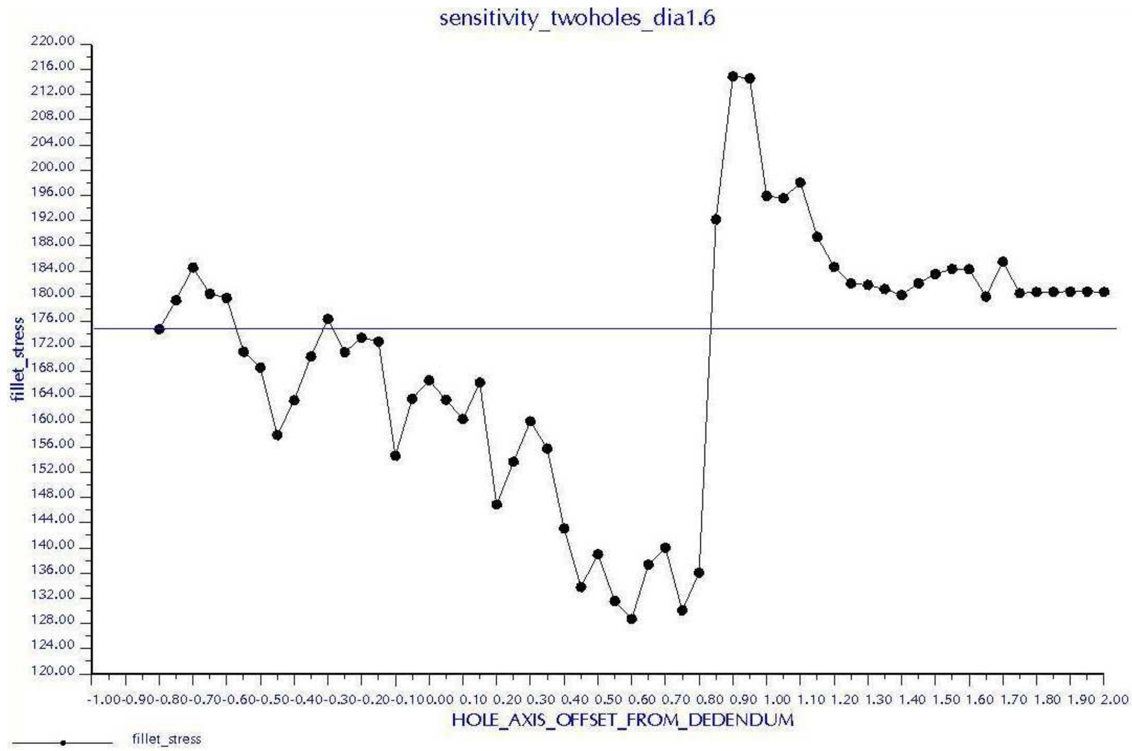


Fig 4.57 Graph of fillet stress vs. hole axis offset for dia 1.6 mm

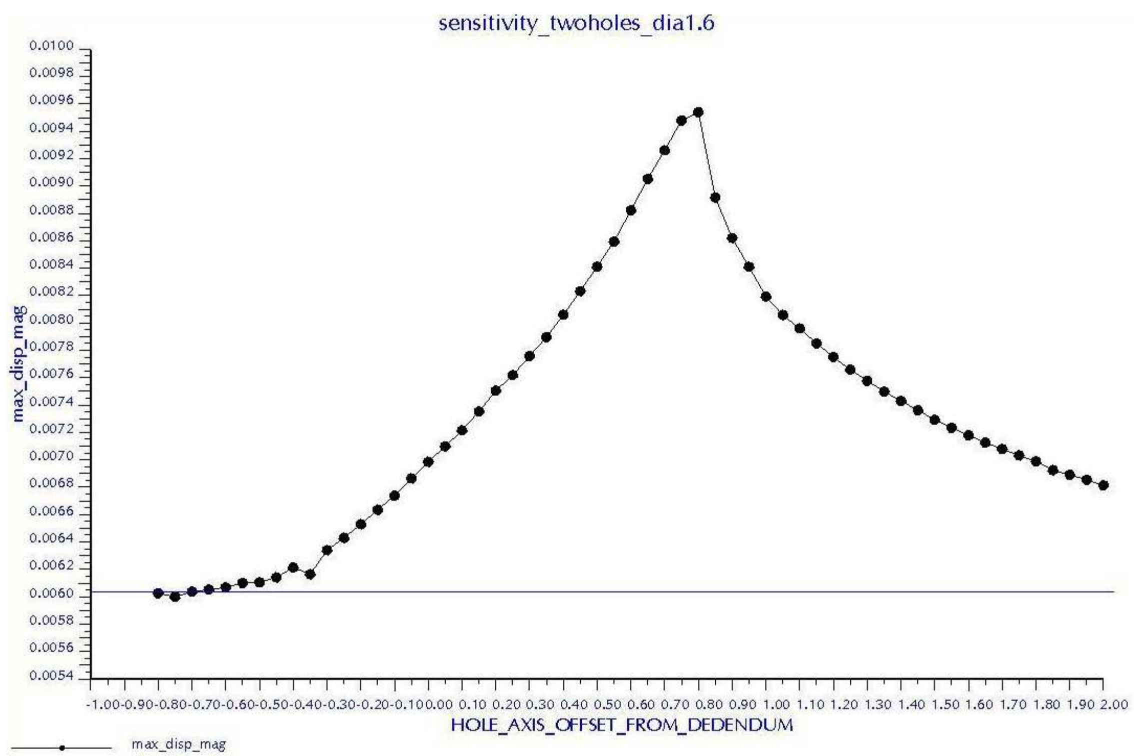


Fig 4.58 Graph of deflection vs. hole axis offset for dia 1.6 mm

Observations: It is observed from the deflection graph that, as the diameter of hole is increased the maximum deflection of the gear tooth increases upto a point where the hole is tangent to the inside of dedendum circle. After this point at an initial high rate of decrease, the deflection decreases but stabilizes at a value at around 20% of the maximum value.

Stress graphs show that as the diameter of the hole is increased, the minimum stress point is experienced around the same point where max deflection occurred. For hole dia less than 0.8 the stress levels do not change appreciably. For hole dia between 0.8 and 1.2 the stress at the fillets tend to gradually decrease to a minimum and then gradually increase to levels beyond the reference stress 168 MPa. For dia greater than 1.2 the graphs show a decrease in stresses with number of local minimas in the stress levels till the inside tangency with the dedendum circle, then a sudden increase beyond 168 MPa.

Selection of optimal design for hole / circular groove size and placement

For this selection offsets of 0.1, 0.15, 0.2, 0.25, 0.3, 0.35, 0.4, 0.45 and 0.5 times the diameter are taken and studies are done for various diameters. This is done based on the previous results so that the circular groove depth varies from a small depth of 10% the hole dia to a maximum of when the point of max deflection, i.e.: the hole forms an inner tangent to the dedendum. The max stress von mises any where on the tooth, groove or fillet and max deflection of tooth are plotted in the following graphs.

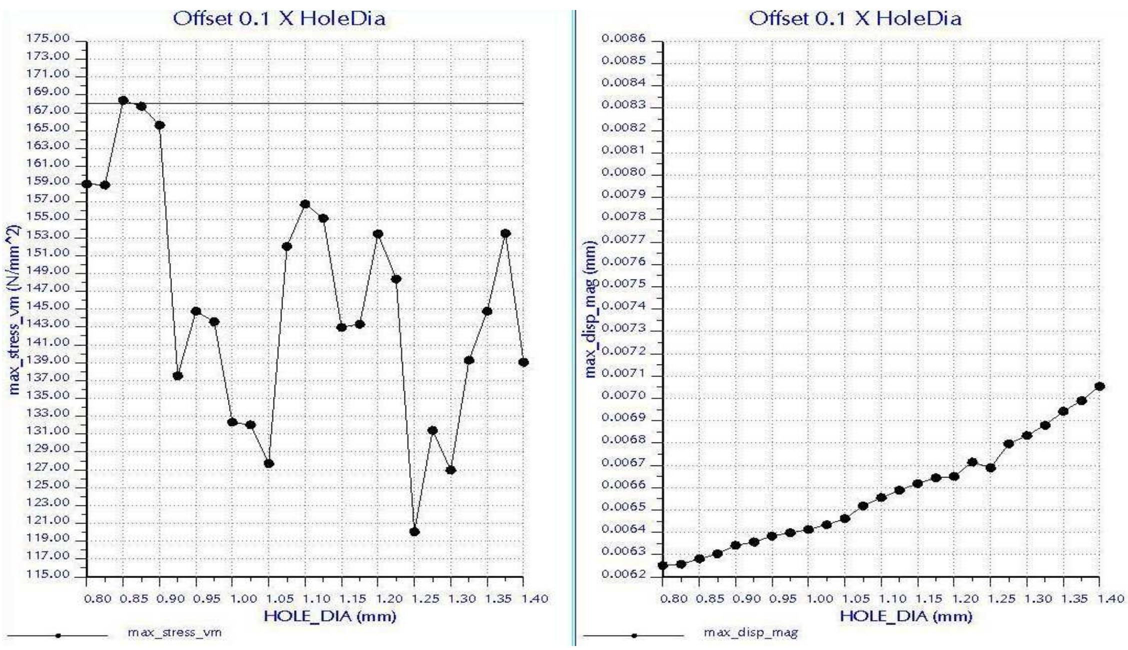


Fig 4.59 Graphs of fillet stress and deflection for offset = 0.1 X Hole dia

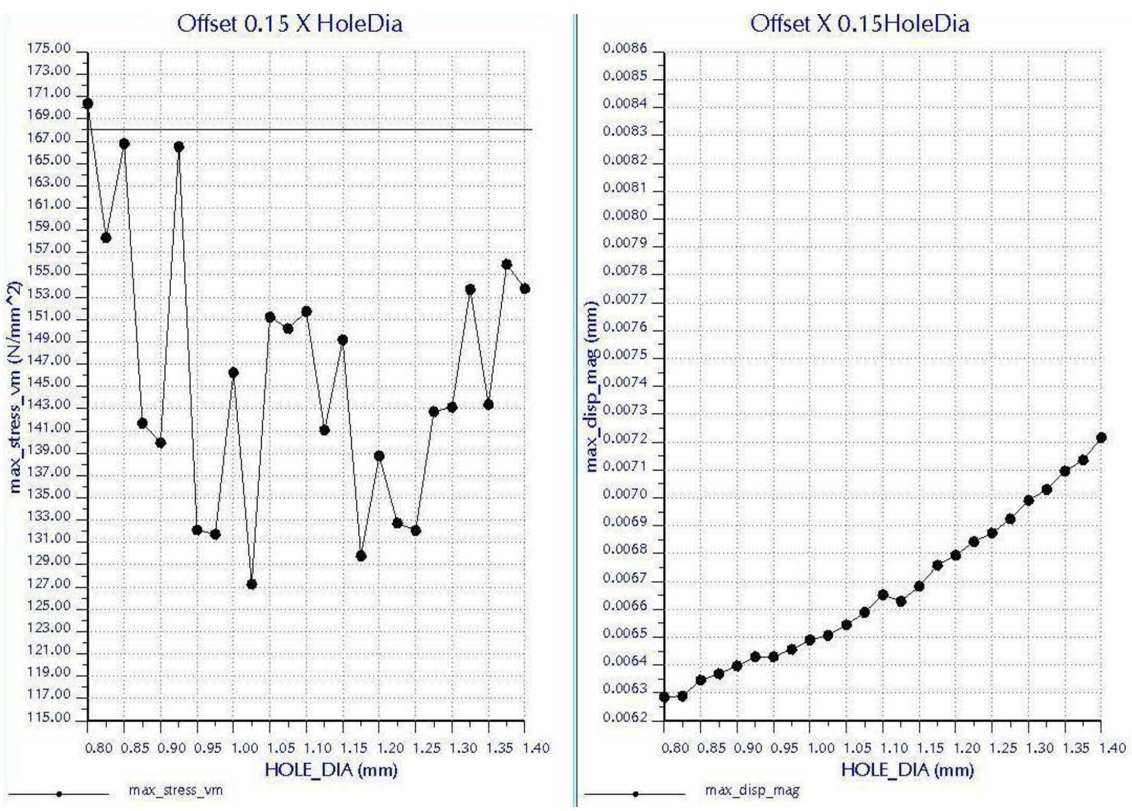


Fig 4.60 Graphs of fillet stress and deflection for offset = 0.15 X Hole dia

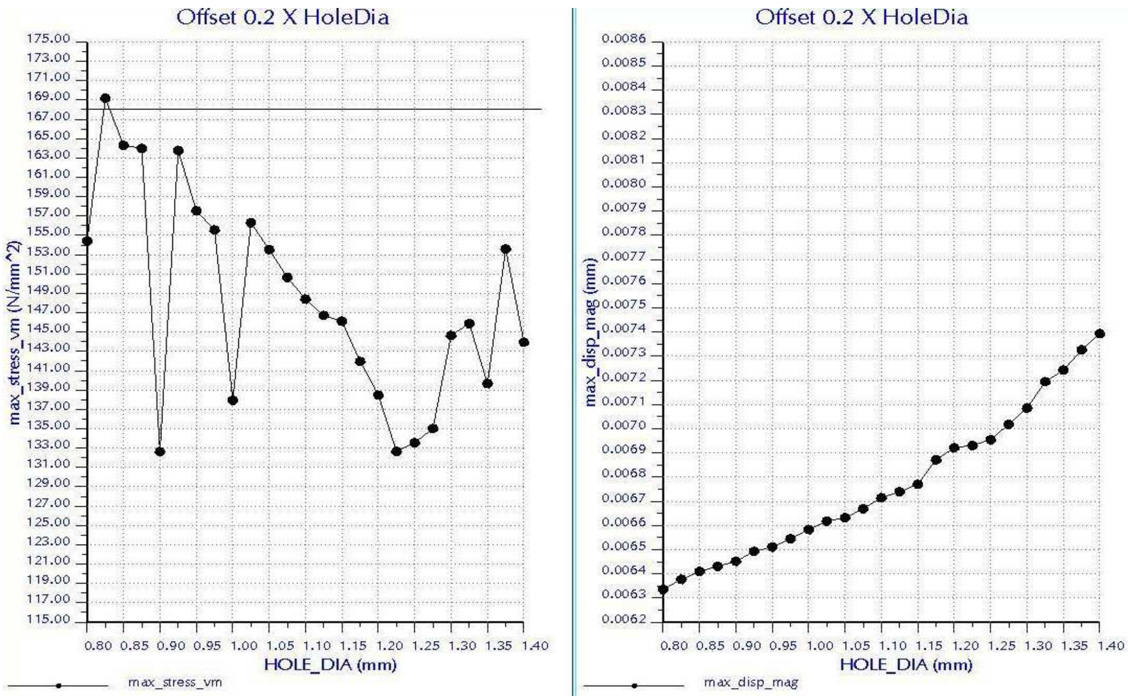


Fig 4.61 Graphs of fillet stress and deflection for offset = 0.2 X Hole dia

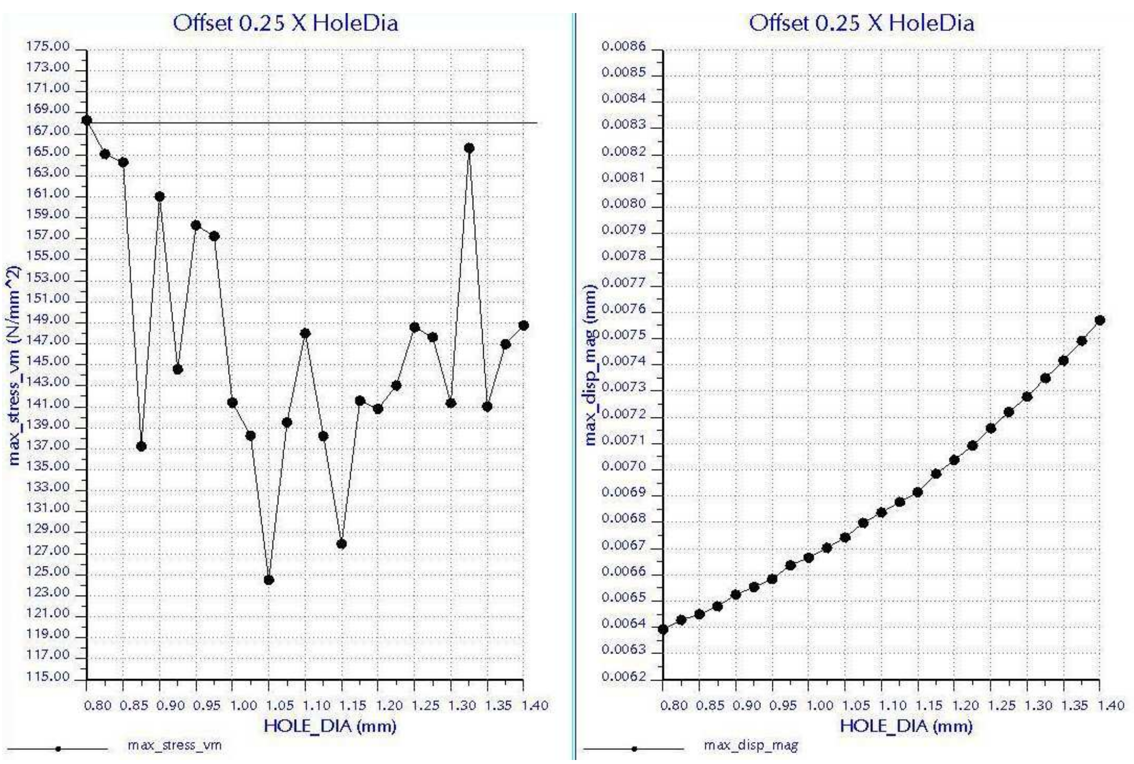


Fig 4.62 Graphs of fillet stress and deflection for offset = 0.25 X Hole dia

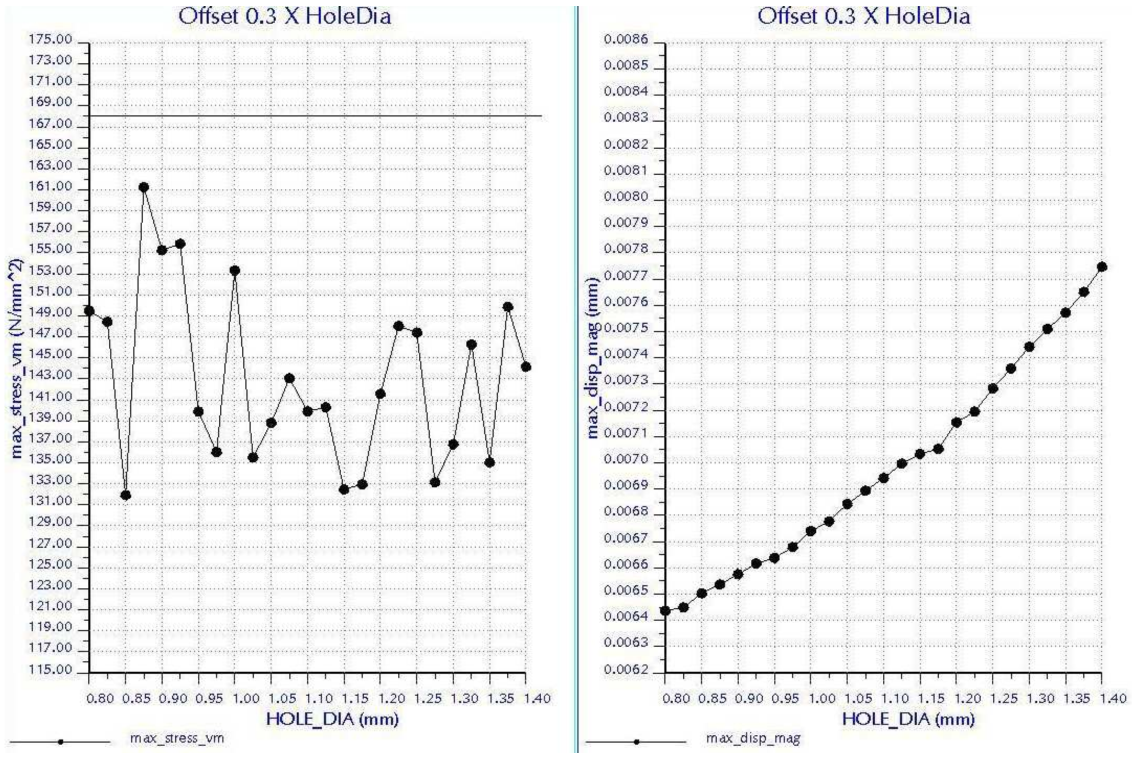


Fig 4.63 Graphs of fillet stress and deflection for offset = 0.3 X Hole dia

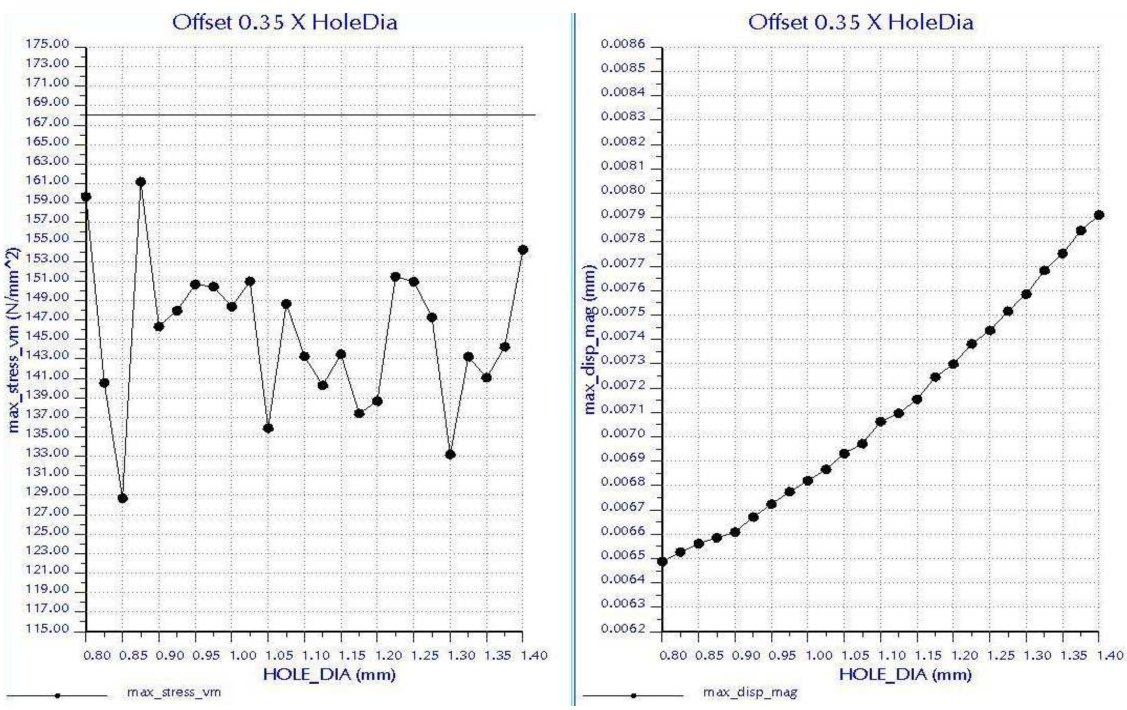


Fig 4.64 Graphs of fillet stress and deflection for offset = 0.35 X Hole dia

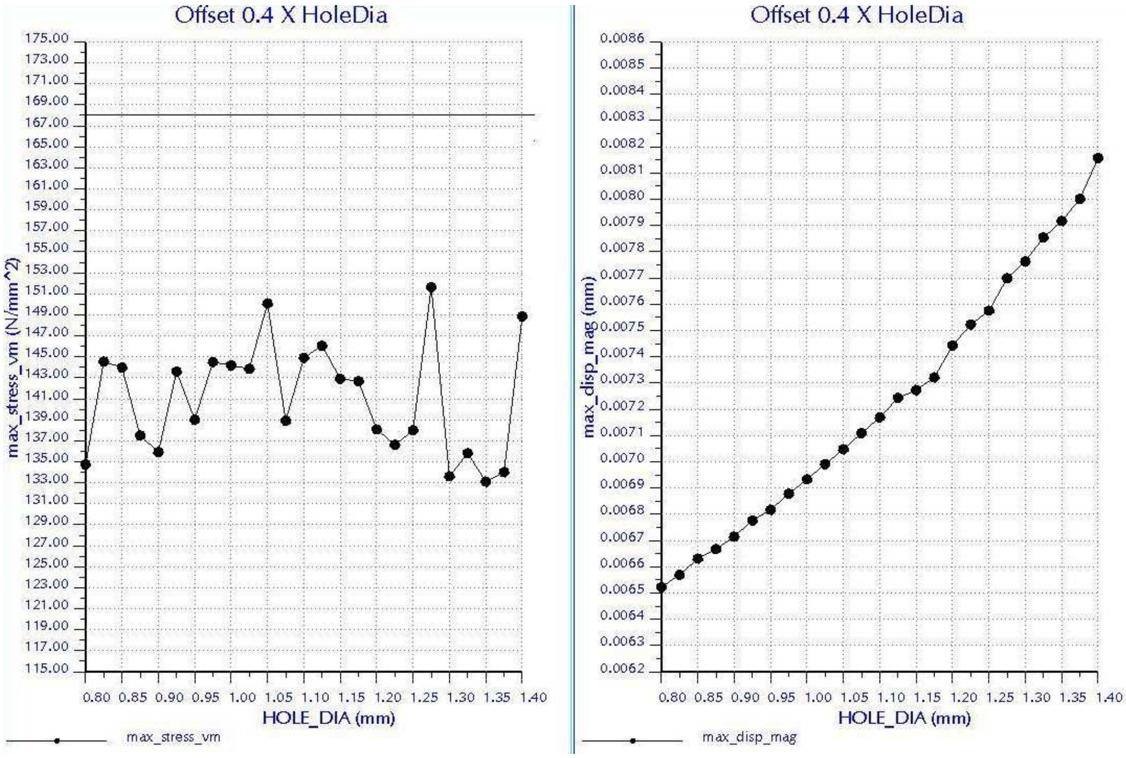


Fig 4.65 Graphs of fillet stress and deflection for offset = 0.4 X Hole dia

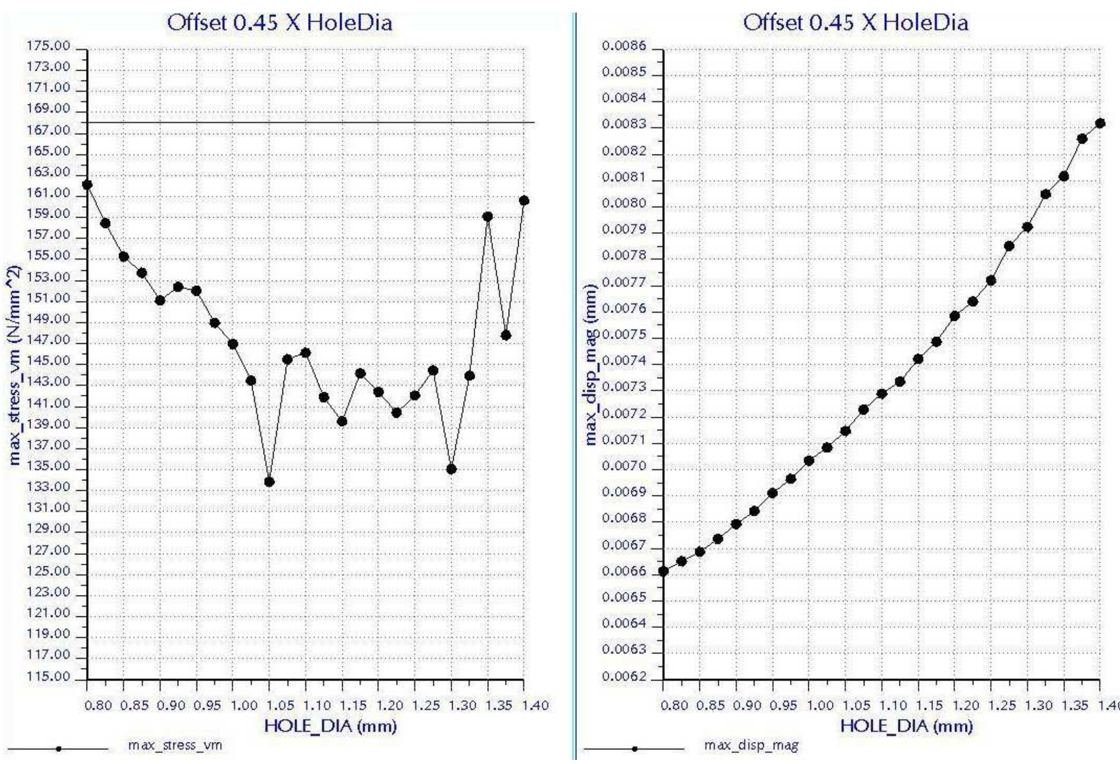


Fig 4.66 Graphs of fillet stress and deflection for offset = 0.45 X Hole dia

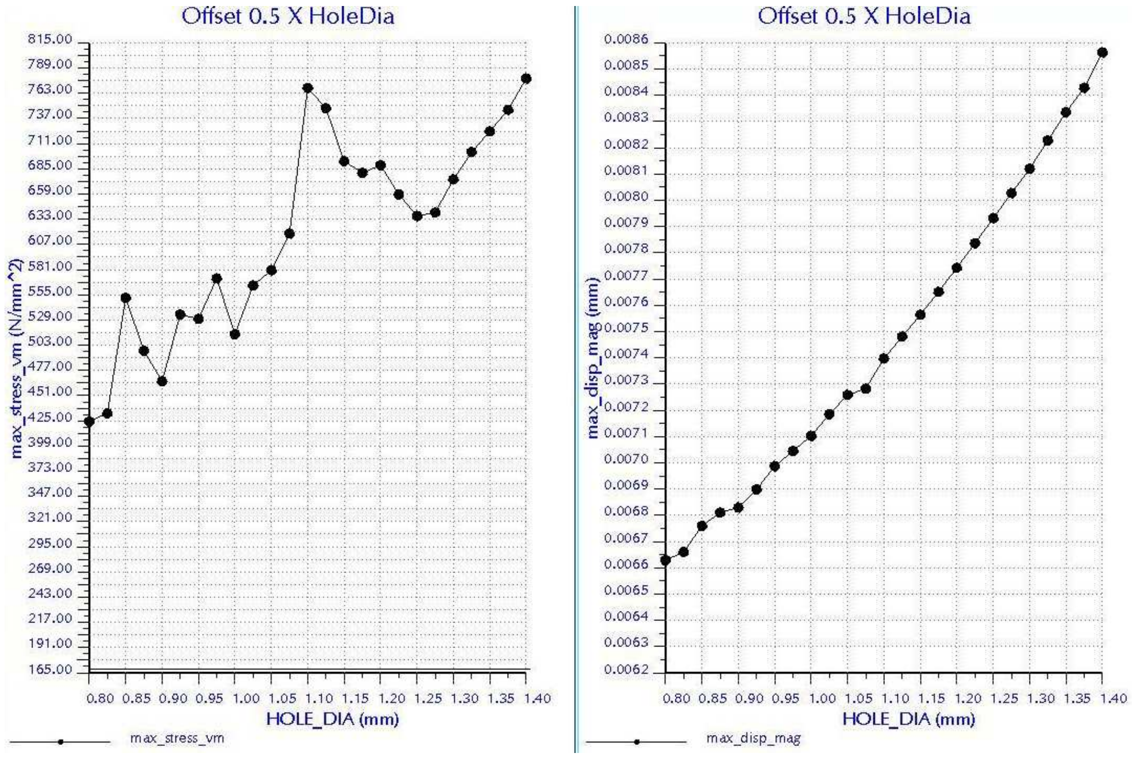


Fig 4.67 Graphs of fillet stress and deflection for offset = 0.5 X Hole dia

From the above graphs it is found that overall stresses can be reduced by a large amount while deflection increased due to loss of rigidity. Large reduction in the stress at fillets has been achieved for large holes but at a cost of rigidity and the maximum stress point also shifts from the fillet to the Hole. From these graphs representative holes are chosen which give a stress reduction but the increase in deflection is within 1 micron.

The best results are listed below:-

Sr No	Dia (mm)	Offset (mm)	Stress (N\mm^2)	Stress Reduction	% Reduction	Deflection (μm)	Deflection Increase (μm)	% Increase
1.	1.025	0.1537	127.41	40.39	24.07	6.5	0.51	8.51
2.	1.05	0.105	127.63	40.17	23.94	6.46	0.47	7.85
3.	0.85	0.2975	129.85	37.95	22.62	6.56	0.57	9.52
4.	0.975	0.1462	131.93	35.87	21.38	6.455	0.46	7.76

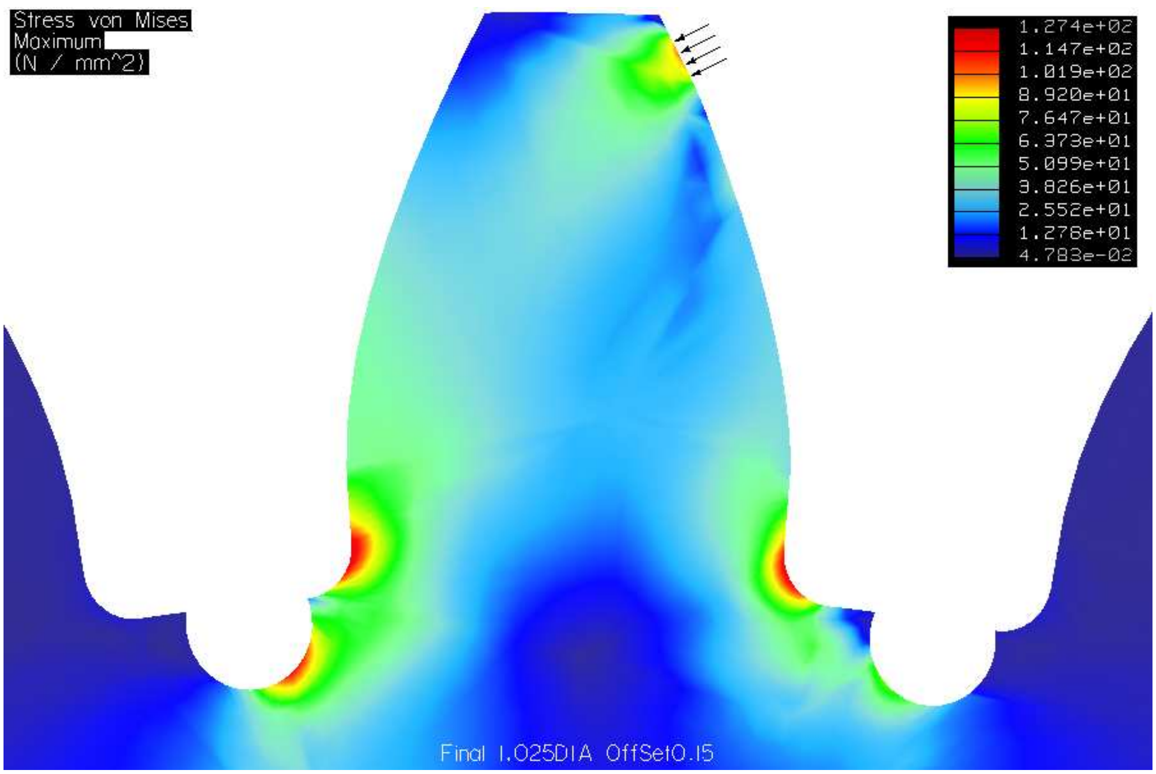
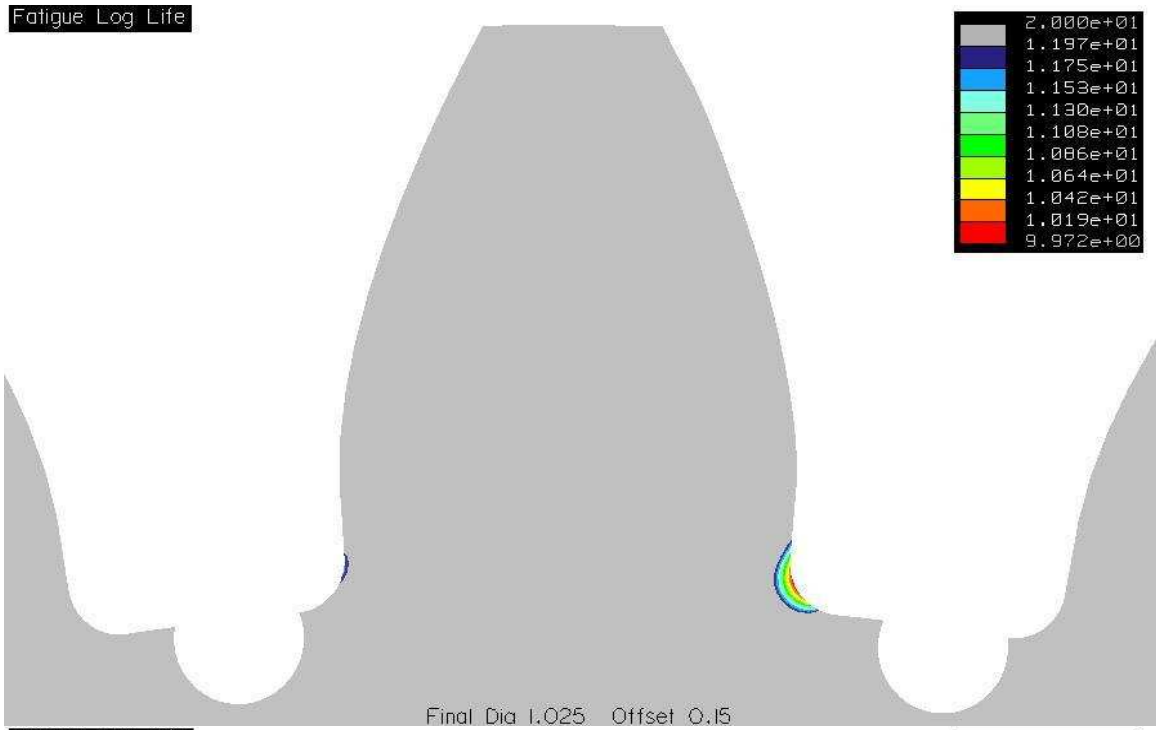


Figure 4.68 Optimized Model of gear tooth

The figure shows how the effective base of the tooth has increased leading to a decrease in the stress levels due to the circular groove of partial hole.

Verification of design improvement using Fatigue Analysis

The best design from stress relief feature studies is taken and compared with the gear without any holes.



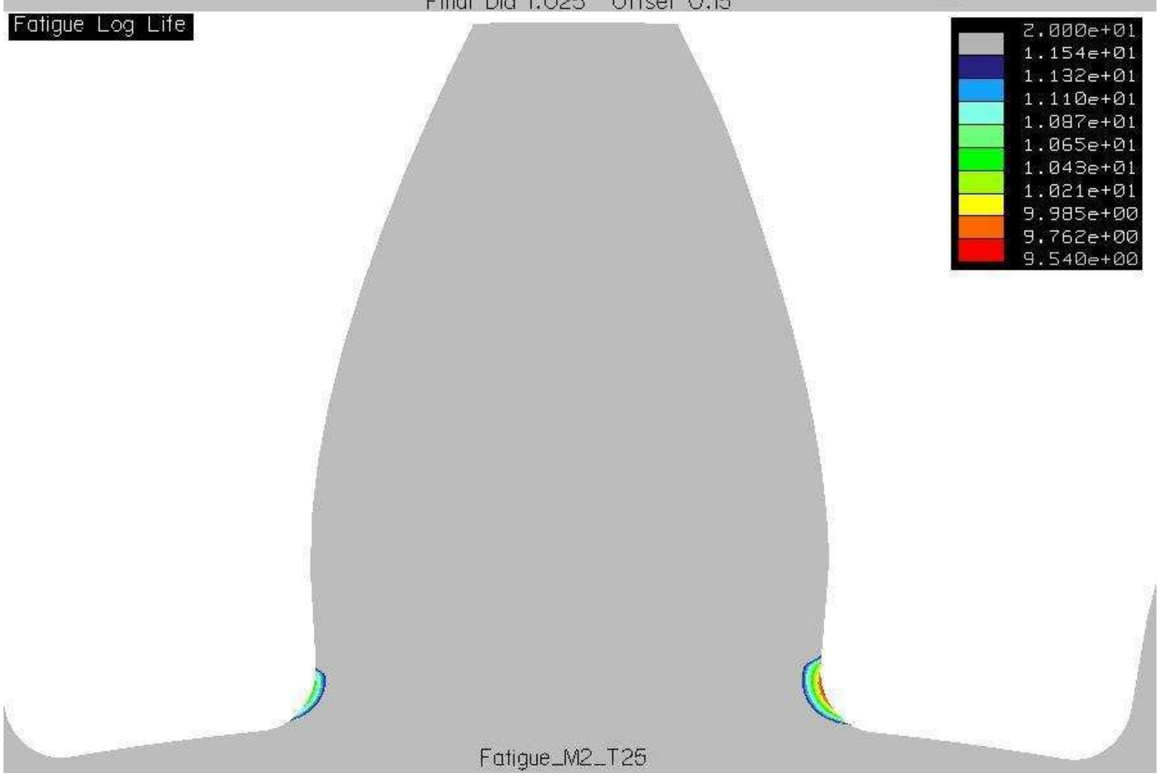


Figure 4.69 & Figure 4.70 Showing fatigue analysis on the two gears

The above figures show the log life of the gears. The life of the gear without hole is about $10^{9.54}$ and the one having diameter 1.025 mm is $10^{9.972}$ an increase of 1.7 times or 170% in the life.

4.2 Conclusion

The maximum load on the pinion is at the point of largest single tooth contact point “H”. The most critical design is for a gear ratio of 1:1.

The stress analysis of a tooth shows that the trailing fillet which has compressive stresses has higher stress levels than the leading fillet which has tensile stresses.

It is concluded from the sensitivity studies of keeping the hole along the profile of the tooth that the effect of any feature like a hole any where above the dedendum and in the tooth leads to an increase in the stresses in the fillets. The choice of the size and location of the circular groove is not a simple process, due to the non linear variations in a complex geometry, as the studies have shown. These studies results show a general tendency to have a stress reduction by adding the groove but the best results have to be chosen only after a detailes sensitivity analysis is done as demonstrated.

The introduction of a hole or circular groove on the dedendum circle reduces the stress levels by a very high percentage with a small loss of rigidity of the tooth. This translates into a exponential increase in the life of the gear due to a better location on the S-N curve for fatigue loading.

4. 3 Scope of further work

4 Other types of gears like helical, bevel, worm etc. can also be studied.

5 Different shapes of hole can also be studied for relieving stress.

6 Study can be carried out by varying the depth of a groove feature only.

7 Other failure criteria can be used for the study other than fatigue.

8 The study can be done using other CAE software like Pro/Mechanica and physical verification of stress relief by prototype / actual testing.

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