REDUCING STRESS CONCENTRATION AT ROOT FILLET IN GEAR TOOTH BY INTRODUCING HOLES THROUGHOUT FACEWIDTH

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In

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ABSTRACT

In Spur gear, the tooth are parallel to the axis. Gears are commonly used for transmitting power from one shaft to another shaft up to certain distance & also used to vary the speed & Torque. The cost of replacement of spur gear is very high. Failure of gear causes breakdown of system. When gear is subjected to load, They develop high stress concentration at the root fillet and the point of contact. So to avoid fatigue failure of the gear, the stresses should be minimized at maximum stress concentrated area. Design of spur gear can be improved by improving the quality of material, improving surface hardness by heat treatment, surface finishing methods. Apart from this stress also occurs during its actual working. Hence it is important to minimize the stresses. These stresses can be minimized by introducing stress relief features at stress zone. A finite element model of Spur gear with a segment of one teeth is considered for analysis and stress concentration reducing holes are introduced on gear tooth at various locations. The main objective of this study is to add holes to reduce stress concentration. Many simulation packages are available for checking the different values of stresses. Simulation is doesn't give exact results but gives a brief idea where stresses are induced. Experimental stress analysis is also used for studying stresses.

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CHAPTER - 1

1.INTRODUCTION

Spur gear is a cylindrical shaped gear in which the TOOTH are parallel to the axis. It is easy to manufacture and it is mostly used in transmitting power from one shaft to another shaft up to certain distance & it is also used to vary the speed & Torque. e.g. Watches, gearbox etc.

The cost of replacement of spur gear is very high and also the system down time is one of the effect in which these gears are part of system. Failure of gear causes breakdown of system which runs with help of gear. E.g. automobile vehicle.

When gear is subjected to load, high stresses developed at the root of the TOOTH, Due to these high Stresses, possibility of fatigue failure at the root of TOOTH of spur gear increases. There is higher chance of fatigue failure at these locations. So to avoid fatigue failure of the gear, the stresses should be minimized at maximum stress concentrated area. Design of spur gear can be improved by improving the quality of material, improving surface hardness by heat treatment, surface finishing methods. Apart from this stress also occurs during its actual working.

Hence it is important to minimize the stresses. These stresses can be minimized by introducing stress relief features at stress zone. Many simulation packages are available for checking the different values of stresses. Simulation is doesn't give exact results but gives a brief idea where stresses are induced. Hence experimental stress analysis method can also be adopted for studying stresses:

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 analyze the gear for stresses. Gears have been analyzed for different points of contact on the tooth profile and the corresponding points of contact on the pinion. In this study

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.

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.

Spur gear is a cylindrical shaped gear in which the TOOTH 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.

1.1.MECHANICAL DRIVES:

Belt, chain and gear drives are often called ‘mechanical drives’. A mechanical drive is defined as a mechanism, which is intended to transmit mechanical power over a distance, usually involving a change in speed and torque. Ingeneral, a mechanical drive is required between a prime mover, such as electric motor and the part of operating machine.

A mechanical drive is used on account of following reasons:

- The torque and speed of machines are always different than that of electric motor or engine. Machines usually run at low speeds and require high torque. For example, in case of an overload travelling crane, the motor runs at 1400 rpm while the speed of rope drum is as low as 20 rpm.

- In certain machines, variable speeds are required for operation, whereas the prime mover runs at constant speed. For example, in case of lathe, the motor runs at constant speed, while different speeds are required for the spindle of chuck to turn the jobs with different speeds and depth of cut.

Mechanical drives are classified into two groups according to their principal of operation. The two broad groups are as follows:

- Mechanical drives that transmit power by means of friction, e.g. belt and rope drives.

- Mechanical drives that transmit power by means of engagement, e.g. chain and gar drives.

  The selection of proper mechanical drive for given application depends on number of factors such as centre distance, velocity ratio, shifting arrangement, maintenance considerations and cost.

The guidelines for any given application are as follows:

- Flat belts and roller chains are suitable for long centre distances. V-belts have comparatively short centre distances. Gear drives have the smallest centre distance between two shafts.

- In flat belt drives, the belt slips over the pulley. Therefore, the driven pulley rotates at a speed which is less than that calculated by the ratio of diameters of the driving and driven pulleys. Due to slip, the velocity ratio is not constant. Therefore, flat belt drive is not recommended where constant speed is desirable. In case of chain drives, the velocity ratio is not constant during one revolution of the sprocket wheel due to 'polygonal' effect. Gear drives are preferred in applications which require constant speed.
- In some applications, shifting mechanism is required to obtain different speeds such as headstock of lathe or automotive gearbox. Flat belts with relatively long centre distances can be shifted from tight to loose pulleys. Spur gears can be shifted on

splined shaft. In case of V-belts or chain drives, it is not possible to use the shifting mechanism.

- Maintenance of belt drives is relatively simple. It usually consists of periodic adjustment of centre distance in order to compensate the stretch of the belt. In chain and gear drives, lubrication is an important consideration in maintenance.
- Flat belt drive is the cheapest, V-belt and chain drives are comparatively costly, and gear drives are costliest.

1.2.GEAR DRIVES:

Gears are defined as toothed wheels or multi lobed cams, which transmit power and motion from one shaft to another by means of successive engagement of TOOTH.

Gear drives offer the following advantages compared with chain or belt drives:

- It is a positive drive and the velocity ratio remains constant.
- The centre distance between the shafts is relatively small, which results in compact construction.
- It can transmit very large power, which is beyond the range of belt or chain drives.
- It can transmit motion at very low velocity which is not possible with the belt drives.
- The efficiency of gear drives is very high, even up to 99 per cent in case of spur gears.
- A provision can be made in the gearbox for gear shifting, thus changing the velocity ratio over a wide range.

Gear drives are, however, costly and their maintenance cost is also higher. The manufacturing processes for gears are complicated and highly specialized. Gear drives require careful attention for lubrication and cleanliness. They also require precise alignment of the shafts.

1.2.1.CLASSIFICATION OF GEARS:

Gears are broadly classified into four groups-

1. Spur gear   2. Helical gear   3.Bevel gear4.Worm gear.

1. Spur gears:

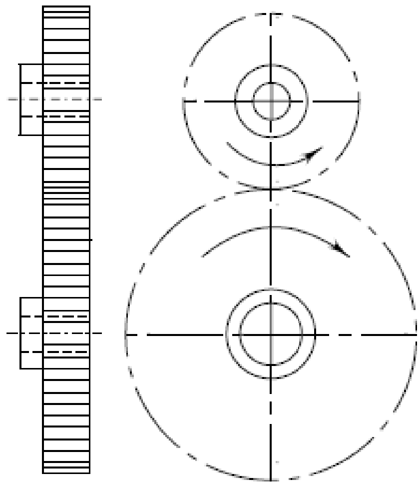

Fig.1.1. Spur gears

The two parallel and co-planar shafts connected by the gears is shown in Fig.1.1. These gears are called spur gears and the arrangement is called as spur gearing. These gears have TOOTH parallel to the axis of wheel. Spur gears are used only when shafts are parallel. The profile tooth is in the shape of involute curve. Spur gears impose radial loads on shafts.

2. Helical gears:

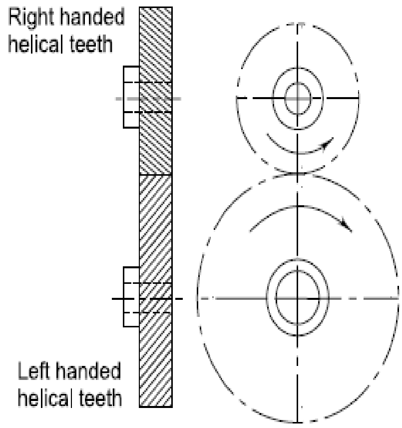

Fig. 1.2.Helical Gears

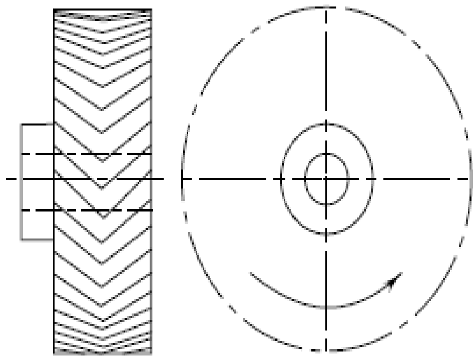

Fig. 1.3. Herringbone Gear

A pair of helical gears is shown in Fig. 1.2. The TOOTH of these gears are cut at an angle with the axis of the shaft. Helical gears have an involute profile similar to that of spur gears. However, this involuteprofile is in a plane, which is perpendicular to the tooth element. The magnitude of the helix angle of pinion and gear is same; however, the hand of the helix is opposite. A right-hand pinion meshes with a left-hand gear and vice versa. Helical gears impose radial and thrust loads on shafts. There is a special type of helical gear, consisting of two helical gears with the opposite hand of helix, as shown in Fig. 1.3. It is called herringbone gear. The construction results in equal and opposite thrust reactions, balancing each other and imposing no thrust load on the shaft. Herringbone gears are used only for parallel shafts.

3. Bevel gears:

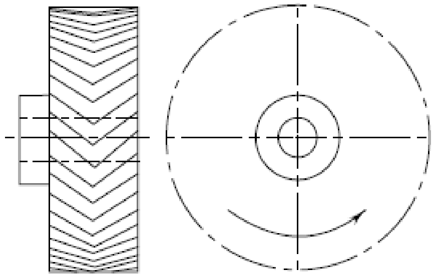

Fig.1.4. Bevel Gears

Bevel gears, as shown in Fig. 1.4, have the shape of a truncated cone. The size of the gear tooth, including the thickness and height, decreases towards the apex of the cone. Bevel gears are normally used for shafts, which are at right angles to each other. This, however, is not a rigid condition and the angle can be slightly more or less than 90 degrees. The tooth of t the bevel gears can be cut straight or spiral. Bevel gears impose radial and thrust loads on the shafts.

4. Worm Gears:

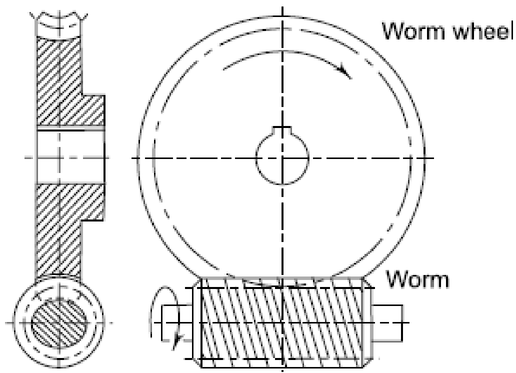

Fig.1.5. Worm Gears

The worm gears, as shown in Fig. 1.5, consist of a worm and a worm wheel. The worm is in the form of a threaded screw, which meshes with the matching wheel. The threads on the worm can be single or multi-start and usually have a small lead. Worm gear drives are used for shafts, the axes of which do not intersect and are perpendicular to each other. The worm imposes high thrust load, while the worm wheel imposes high radial load on the shafts. Worm gear drives are characterized by high speed reduction ratio.

1.2.2. LAW OF GEARING:

The fundamental law of gearing states 'The common normal to the tooth profile at the point of contact should always pass through a fixed point, called the pitch point, in order to obtain a constant velocity ratio'. It has been found that only involute and cycloidal curves satisfy the fundamental of law of gearing

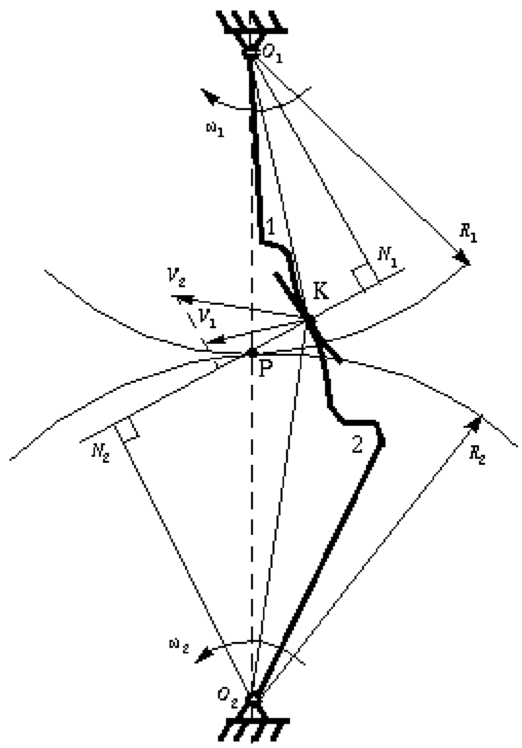

Fig.1.6 Law of Gearing

The meaning of these curves is as follows:

(i) An involute is a curve traced by a point on a line as the line rolls without slipping on a circle.
(ii) A cycloid is a curve traced by a point on the circumference of a generating circle as it rolls without slipping along the inside and outside of another circle. The cycloid profile consists of two curves, namely, epicycloids and hypocycloid. An epicycloid is a curve traced by a point on the circumference of a generating circle as it rolls without slipping on the outside of the pitch circle. A hypocycloid is a curve traced by a point on the circumference of a generating circle as it rolls without slipping on the inside of the pitch circle.

Cycloidal tooth offers the following advantages compared with involute tooth:
(i) In case of cycloidal gears, a convex flank on one tooth comes in contact with the concave flank of the mating tooth. This increases the contact area and also the wear strength. In involute gears, the contact is between two convex surfaces on mating TOOTH, resulting in smaller contact area and lower wear strength.
(ii) The phenomenon of interference does not occur at all in cycloidal gears.

However, cycloidal TOOTH are rarely used in practice due to the following disadvantage:
Cycloidal tooth is made of two curves- hypocycloid curve below the pitch circle and epicycloid curve above the pitch circle. It is very difficult to manufacture an accurate profile consisting of two curves. The profile of an involute tooth is made of a single curve and only one cutter is necessary to manufacture one complete set of pinion and gear. This results in reduction in manufacturing cost.

1.2.3.TERMINOLOGY OF SPUR GEARS:

The terminology of gears includes a number of terms peculiar to gears and it forms the basis of gear language. The terminology applied to spur gears is illustrated below
(i) Pinion: A pinion is the smaller of the two mating gears.
(ii) Gear: A gear is the larger of the two mating gears.

(iii) Velocity Ratio (i): Velocity ratio is the ratio of angular velocity of the driving gear to the angular velocity of the driven gear. It is also called the speed ratio.

(iv) Transmission Ratio (i'): The transmission ratio (i') is the ratio of the angular speed of the first driving gear to the angular speed of the last driven gear in a gear train.

(v) Pitch Surface: The pitch surfaces of the gears are imaginary planes, cylinders or cones that roll together without slipping.

(vi) Pitch Circle: The pitch circle is the curve of intersection of the pitch surface of revolution and the plane of rotation. It is an imaginary circle that rolls without slipping with the pitch circle of a mating gear. The pitch circles of a pair of mating gears are tangent to each other.

(vii) Pitch Circle Diameter: The pitch circle diameter is the diameter of the pitch circle. The sizeof the gear is usually specified by the pitch circle diameter. It is also called pitch diameter. The pitch circle diameter is denoted by d $\not\subset$.

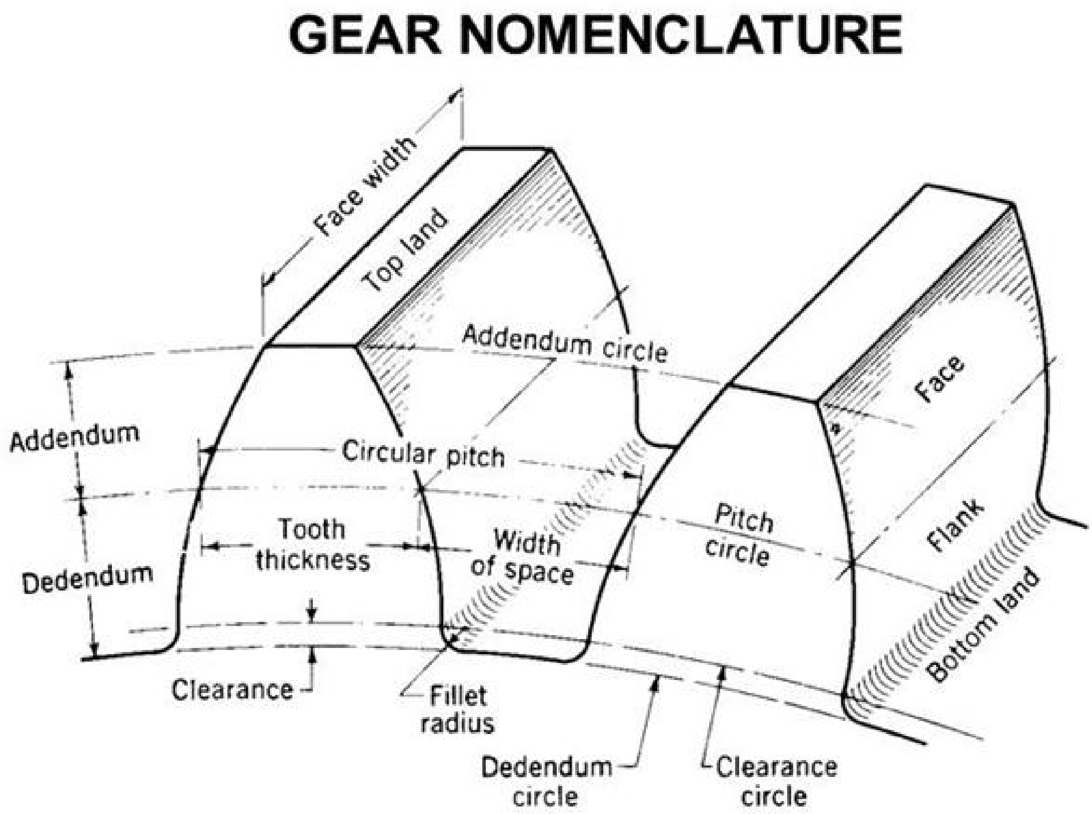

Fig.1.7 Gear nomenclature

(viii) Pitch Point: The pitch point is a point on the line of centers of two gears at which two pitch circles of mating gears are tangent to each other.

(ix) Top Land: The top land is the surface of the top of the gear tooth.

(x) Bottom Land: The bottom land is the surface of the gear between the flanks of adjacent TOOTH.

(xi) Involute: An involute is a curve traced by a point on a line as the line rolls without slipping on a circle.

(xii) Base Circle: The base circle is an imaginary circle from which the involute curve of the tooth profile is generated. The base circles of two mating gears are tangent to the pressure line.

(xiii) Addendum Circle: The addendum circle is an imaginary circle that borders the tops of gear TOOTH in the cross section.

(xiv) Addendum (ha): The addendum (ha) is the radial distance between the pitch and the addendum circles. Addendum indicates the height of the tooth above the pitch circle.

(xv) DedendumCircle: Thededendum circle is an imaginary circle that borders the bottom of spaces between TOOTH in the cross section. It is also called root circle.

(xvi) Dedendum (hf): Thededendum (hf) is the radial distance between pitch and the dedendum circles. The dedendum indicates the depth of the tooth below the pitch circle.

(xvii) Clearance (c): The clearance is the amount by which the dedendum of a given gear exceeds the addendum of its mating tooth.

(xviii) Face of Tooth: The surface of the gear tooth between the pitch cylinder and the addendum cylinder is called the face of tooth.

(xix) Flank of Tooth: The surface of the gear tooth between the pitch cylinder and the root cylinder is called flank of the tooth.

(xx) Face Width (b): Face width is the width of the tooth measured parallel to the axis.

(xxi) Fillet Radius: The radius that connects the root circle to the profile of the tooth is called fillet radius.

(xxii) Circular Tooth Thickness: The length of the arc on the pitch circle subtending a single gear tooth is called circular tooth thickness. Theoretically, circular tooth thickness is half of the circular pitch.

(xxiii) Tooth Space: The width of the space between two adjacent TOOTH measured along the pitch circle is called the tooth space. Theoretically, tooth space is equal to circular tooth thickness or half the circular pitch.

(xxiv) Working Depth (hk): The working depth is the depth of engagement of two gear TOOTH, that is, the sum of their addendums.

(xxv) Whole Depth (h): The whole depth is the total depth of the tooth space, that is, the sum of the addendum and dedendum. Whole depth is also equal to working depth plus clearance.

(xxvi) Centre Distance: Thecentre distance is the distance between centres of pitch circles of mating gears. It is also the distance between centres of base circles of mating gears.

(xxvii) Pressure Angle: The pressure angle is the angle which the line of action makes with the common tangent to the pitch circles. The pressure angle is also called the angle of obliquity. It is denoted by a.

(xxviii) Line of Action: The line of action is the common tangent to the base circles of mating gears. The contact between the involute surfaces of mating TOOTH must be on this line to give a smooth operation. The force is transmitted from the driving gear to the driven gear on this line.

(xxix) Arc of Contact: The arc of contact is the arc of the pitch circle through which a tooth moves from the beginning to the end of contact with mating tooth.

(xxx) Arc of Approach: The arc of approach is the arc of the pitch circle through which a tooth moves.

(xxxi) Arc of Recess: The arc of recess is the arc of the pitch circle through which a tooth moves from the contact at the pitch point until the contact ends.

(xxxii) Contact Ratio (mp): the number of pairs of tooth that are simultaneously engaged is called contact ratio. If there are two pairs of tooth in contact all the time, the contact ratio is 2. As the two gears rotate, smooth and continuous transfer of power from one pair of meshing tooth to the following pair is achieved when the contact of the first pair continues until the following pair has established contact. Some overlapping is essential for this purpose. Therefore, the contact ratio is always more than 1. Other things being, the greater the contact ratio, the smoother the action of gears. The contact ratio for smooth transfer of motion is usually taken as 1.2. In industrial gearboxes for power transmission, the contact ratio is usually more than 1.4 (1.6 to 1.7).

(xxxiii) circular pitch(p): the circular pitch (p) is the distance measured along the pitch circle between two similar points on adjacent tooth.

(xxxiv) diametralpitch(p): the diametral pitch (p) is the ratio of the number of tooth to the pitch circle diameter.

(xxxv) module(m): the module (m) is defined as the inverse of the diametral pitch.

(xxxvi) root circle: the circle that passes through the bottom of the tooth spaces.

(xxxvii) root diameter: the diameter of the root circle.

1.2.4. STANDARD SYSTEMS OF GEAR TOOTH :

All standard systems describe the involute profile for gear tooth. The results are as follows:

- The involute profile satisfies the fundamental law of gearing at any central distance.
- All involute gears of a given module and pressure angle can be machined from one single.
- The basic rack of an involute profile has straight sides. It is comparatively easy to machine straight sides. Further, straight sides can be more accurately machined compared with a curved surface.
- A slight change in centre distance, which may be caused by incorrect mounting, has no effect upon the shape of involute. The velocity ratio also remains constant.

There are three standard systems for the shape of gear TOOTH. They are as follows:

- 14.5^o full depth involute system.
- 20^o full depth involute system.
- 20^o sub involute system.

The 20^o pressure angle system with full depth involute TOOTH is widely used in practice. It is also recommended by the Bureau of Indian standards.

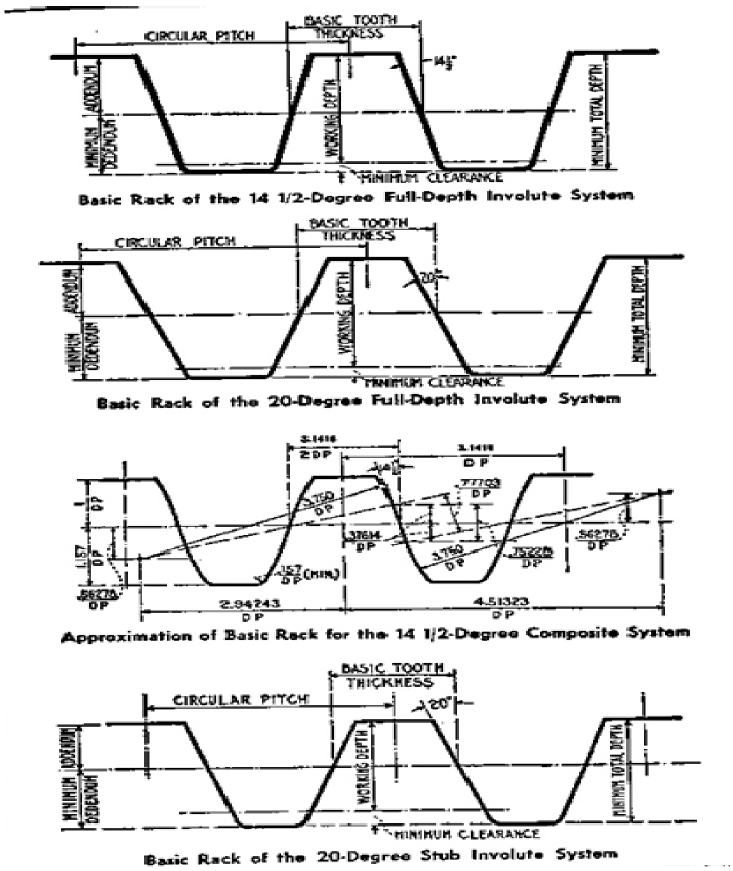

FIG. 1.8. Basic racks for Standard Gear Systems

1.2.5. BEAM STRENGTH OF GEAR TOOTH:

The analysis of bending stresses in gear tooth was done by Wilfred Lewis in his paper. 'The investigation of the strength of gear tooth' in 1892. Even today, the Lewis equation is considered as the basic equation in design of gears. In the Lewis analysis, the gear tooth is treated as a cantilever beam as shown in Fig.1.9. The tangential component (P_t) causes the bending moment about the base of the tooth. The Lewis equation is based on the following assumptions:

- The effect of radial component (P_r), which induces compressive stresses, is neglected.
- It is assumed that the tangential component (P_t) is uniformly distributed over the face width of the gear. It is possible when the gears are rigid and are accurately machined.

- The effect of stress concentration is neglected.
- It is assumed that at any time, only one pair of TOOTH is in contact and takes the total load.

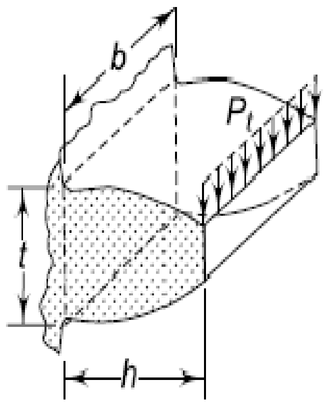

Fig. 1.9 Gear tooth as cantilever beam

It is observed that the cross-section of the tooth varies from free end to fixed end. Therefore, a parabola is constructed within the tooth profile and shown by a dotted line in fig.1.10. The advantage of parabolic outline is that, it is a beam of uniform strength. For this beam the stress at any cross-section is outline is uniform or same. The weakest section of the gear tooth is at the section XX, where the parabola is tangent to the tooth profile.

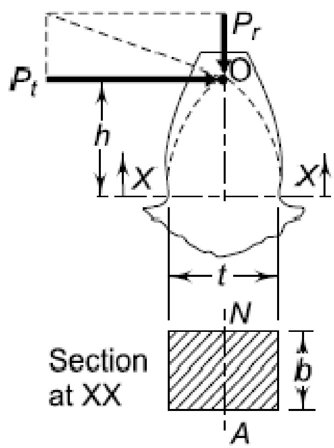

Fig. 1.10 Gear tooth as parabolic beam

At the section XX, $M_b = P_t * h$

$$I = \left(\frac{1}{12}\right) bt^3$$

$$y = \frac{t}{2}$$

The bending stresses are given by,

$$\sigma_b = \frac{M_b y}{I} = \frac{(P_t \times h)\left(\frac{t}{2}\right)}{\left[\left(\frac{1}{12}\right) b t^3\right]}$$

Rearranging the terms,

$$P_t = b\sigma_b \left(\frac{t^2}{6h}\right)$$

Multiplying the numerator and denominator of the right-hand side by m,

$$P_t = mb\sigma_b \left(\frac{t^2}{6hm}\right)$$

Defining a factor Y,

$$Y = \left(\frac{t^2}{6hm}\right)$$

the equation is rewritten as,

$P_t = mb\sigma_b Y$ ……………..…………………..(1.1)

In the above equation, Y is called the Lewis form factor. Equation (1.1) gives the relationship betweenthe tangential force (P_t) and the correspondingstresssb. When the tangential force is increased,the stress also increases. When the stress reaches the permissible magnitude of bending stresses, thecorresponding force (P_t) is called the beam strength. Therefore, the beam strength (S_b) is the maximum value of the tangential force that the tooth can transmit without bending failure. Replacing (Pt) by(S_b), Eq. (1.1) is modified in the following way:

$S_b = mb\sigma_b Y$……………………………..(1.2)

where,

S_b= beam strength of gear tooth (N)

σ_b= permissible bending stress (N/mm2)

The values of the Lewis form factor Y for 20° full-depth involute system, are given in Table.

Table 1.1Values of the Lewis form factor Y for 20° full-depth involute system

z	Y	Z	Y	z	Y
15	0.289	27	0.348	55	0.415
16	0.295	28	0.352	60	0.421
17	0.302	29	0.355	65	0.425
18	0.308	30	0.358	70	0.429
19	0.314	32	0.364	75	0.433
20	0.320	33	0.367	80	0.436
21	0.326	35	0.373	90	0.442
22	0.330	37	0.380	100	0.446
23	0.333	39	0.386	150	0.458
24	0.337	40	0.389	200	0.463
25	0.340	45	0.399	300	0.471
26	0.344	50	0.408	Rack	0.484

In order to avoid the breakage of gear tooth due to bending, the beam strength should be more than the effective force between the meshing TOOTH.

Therefore,

$$\mathbf{Sb \geq P_{eff}}$$

In the design of gears, it is required to decide the weaker between the pinion and gear.

Rewriting the Lewis equation,

$$S_b = mbsbY$$

It is observed that m and b are same for pinion as well as for gear. When different materials are used, the product (σ_bXY) decides the weaker between pinion and gear. The Lewis form factor Y is always less for a pinion compared with gear. When the same material is used for the pinion and gear, the pinion is always weaker than the gear.

1.2.6. PERMISSIBLE BENDING STRESS

The tooth of the gear is subjected to fluctuatingbending stress as it comes in contact with themeshing tooth.

Theendurance limit stress of the gear tooth dependsupon the following factors:

i. Surface finish of the gear tooth.
ii. Size of the gear tooth.
iii. Reliability used in design.
iv. Stress concentration in the gear tooth.
v. Gears rotating in one direction or bothdirections.
vi. Gears tooth subjected to stress in onedirection or both directions.

In practice, it is difficult to get the abovementioneddata for each and every case of gear design.

Earle Buckingham has suggested that the endurancelimit stress of gear tooth is approximately one-thirdof the ultimate tensile strength of the material.

$$\sigma_b = S_e = \left(\frac{1}{3}\right) S_{ut}$$

In case of bronze gears, the endurance limit stress istaken as 40% of the ultimate tensile strength.

1.3. APPLICATIONS OF SPUR GEAR:

Spur gears have a wide range of applications. They are used in:

1. Metal cutting machines
2. Power plants
3. Marine engines
4. Mechanical clocks and watches
5. Fuel pumps
6. Washing Machines
7. Gear motors and gear pumps
8. Rack and pinion mechanisms
9. Material handling equipments
10. Automobile gear boxes
11. Steel mills
12. Rolling mills

CHAPTER - 2

2.LITERATURE REVIEW

Andrews J.D, [1]investigated the finite element analysis of bending stresses in involute gears. This paper describes the use of the finite element method for predicting the fillet stress distribution experienced by loaded spur gears. The location of the finite element model boundary and the element mesh density are investigated. Fillet stresses predicted by the finite element model are compared with the results of photo elastic experiments. Both external and internal spur gear tooth forms are considered.

Costopoulos Th. [2] studied the reduction of gear fillet stresses by using one-sided involute tooth. For increasing the load carrying capacity of geared power transmissions several tooth designs alternative to the standard involute have been proposed. The use of non-involute tooth has a number of disadvantages and for this reason asymmetric involute-type tooth have been studied as a promising alternative. In this paper the idea of one-sided involute asymmetric spur gear tooth is introduced to increase load carrying capacity and combine the good meshing properties of the driving involute and the increased strength of non involute curves. These novel tooth are intended for constant direction of rotation although they can be used in a limited way for reverse rotation. Both flanks are fully generated by a hob, the design of which is also investigated. The increase in load carrying capacity can reach up to 28% compared to the standard 200 involute tooth.

Hebbal M.S, [3] explains the reduction in root fillet stress in spur gear using internal stress relieving features. Gear TOOTH failure due to fatigue is a common phenomenon observed. Even a slight reduction in the root tensile stress results in great increase in the fatigue life of a gear. If gear fails in tensile fatigue, the results are catastrophic and occur with little or no warnings. Therefore for all the reasons mentioned above, this work is of more practical importance. For many years, gear design has been improved by using improved material, hardening surfaces with heat treatment and carburization, and shot penning to improve surface finish etc. Few more efforts have been made to improve the durability and strength by altering the pressure angle, using the asymmetric TOOTH, altering the geometry of root fillet curve and so on. Most of the above methods do not guarantee the interchangeability of the existing gear systems. This work presents the possibilities of using the stress redistribution techniques by introducing the stress relieving features in the stressed zone to the advantage of reduction of root fillet stress in spur gear. It also ensures interchangeability of the existing gear systems. In this work, combination of circular and elliptical stress relieving features are used and better results are obtained than using circular stress relieving features alone which were used by earlier researchers. A finite element model with a segment of three TOOTH is considered for analysis and stress relieving features of various sizes are introduced on gear TOOTH at various locations. Analysis revealed that, combination of elliptical and circular stress relieving features at specific, locations are beneficial than single circular, single elliptical, two circular or two elliptical stress reliving features.

Senthil Kumar V. [4] focused the optimization of asymmetric spur gear drives to improve the bending load capacity. In a given size of symmetric involute gear designed through conventional approach, as the load carrying capacity is restricted at the higher pressure angle due to tipping formation, the use of the asymmetric toothed gear to improve the fillet capacity in bending is examined in this study. Non-standard asymmetric rack cutters with required pressure angles and module are developed to generate the required pinion and gear of a drive with asymmetric involute surfaces and trochoidal fillet profiles. The respective profiles thus generated are optimized for balanced fillet stresses that are equal and possibly the lowest also. For this study of optimization, several non-standard asymmetric rack cutters are designed to accommodate different combinations in the values of pressure angle, top land thickness ratio, profile shift, speed ratio and the asymmetric factors. However for any drive with a given center distance and a speed ratio, only two non-standard asymmetric rack cutters, one for the pinion and other for the gear are used to generate a required numbers of pinion and gear with different cutter shift values for the purpose of optimization. The influence of these parameters on the maximum fillet stress has been analyzed to suggest the optimum values of these parameters that improve the fillet capacity in bending. The optimization of the asymmetric spur gear drive is carried out using an iterative procedure on the calculated maximum fillet stresses through FEM for different rack cutter shifts and finally the optimum values of rack cutter shifts are suggested for the given center distance and the speed ratio of an asymmetric gear drive. Comparisons have also been made successfully with the results of the AGMA and the ISO codes for symmetric gears to justify the results of the finite element method pertaining to this study.

Ulrich T.W.[5] explains the auxiliary holes for stress reduction. Stress concentration reduction in a plate is accomplished by introducing optimum size holes and regularly placed. This paper presents a method based on boundary elements and mathematical programming to determine these auxiliary holes. The mathematical programming method consists of a modified Newton's method and subsequent parallel tangents method (PARTAN). A solution is presented for an elliptical hole in a tension strip.

Mrs. Shinde S.P., Mr. Nikama.A., Mr. Mulla T.S. [6] In this paper bending stress analysis will be performed, While trying to design spur gears to resist bending failure of the TOOTH, as it affects transmission error. First the finite element models and solution methods needed for the accurate calculation using ANSYS, were compared to the results obtained from existing methods.

N. Lenin Rakesh, V. Palanisamy and Sidhant Das [7] A spur gear is taken into consideration. The need for this project is that in earlier times design of any machine element were carried out manually. This was tedious and time consuming. In this emerging world of technology, new software's for modeling and analyzing are also available. That makes design, modeling and analysis easier

CHAPTER– 3

3.REDUCTION OF STRESS CONCENTRATION

Although it is not possible to completely eliminate the effect of stress concentration, there are methods to reduce stress concentrations. This is achieved by providing a specific geometric shape to the component. In order to know what happens at the abrupt change of cross-section or at the discontinuity and reduce the stress concentration, understanding of flow analogy is useful. There is a similarity between velocity distribution in fluid flow in a channel and the stress distribution in an axially loaded plate shown in Fig. The equations of flow potential in fluid mechanics and stress potential in solid mechanics are same.

In practice, reduction of stress concentration is achieved by the following methods:

3.1. ADDITIONAL NOTCHES AND HOLES IN TENSION

Member A flat plate with a V-notch subjected to tensile force is shown in Fig.. It is observed that a single notch results in a high degree of stress concentration. The severity of stress concentration is reduced by three methods: (a) use of multiple notches; (b) drilling additional holes; and (c) removal of undesired material. These methods are illustrated in Fig. (c) and (d) respectively. The method of removing undesired material is called the principle of minimization of the material. In these three methods, the sharp bending of a force flow line is reduced and it follows a smooth curve.

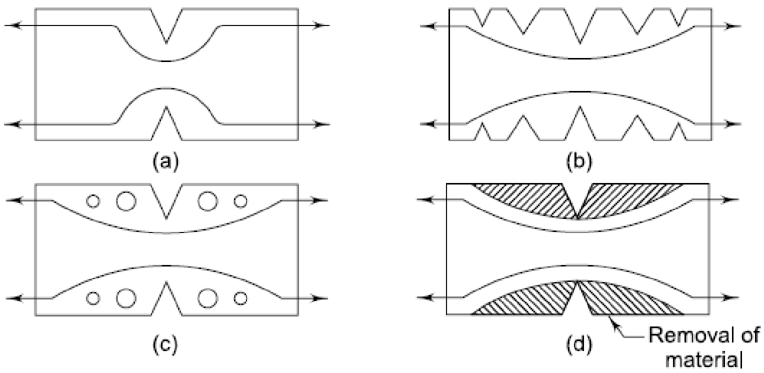

Fig.3.1: Reduction of Stress Concentration due to V-notch: (a) Original Notch (b) Multiple Notches(c) Drilled Holes (d) Removal of Undesirable Material

3.2. FILLET RADIUS, UNDERCUTTING AND NOTCH FORMEMBER IN BENDING

A bar of circular cross-section with a shoulder and subjected to bending moment is shown in Fig.. Ball bearings, gears or pulleys are seated against this shoulder. The shoulder creates change in cross-section of the shaft, which results in stress concentration. There are three methods to reduce stress concentration at the base of this shoulder. Figure shows the shoulder with a fillet radius r. This results in gradual transition from small diameter to a large diameter. The fillet radius should be as large as possible in order to reduce
stress concentration. In practice, the fillet radius is limited by the design of mating components. The fillet radius can be increased by undercutting the shoulder. A notch results in stress concentration. Surprisingly, cutting an additional notch is an effective way to
Reduce stress concentration.

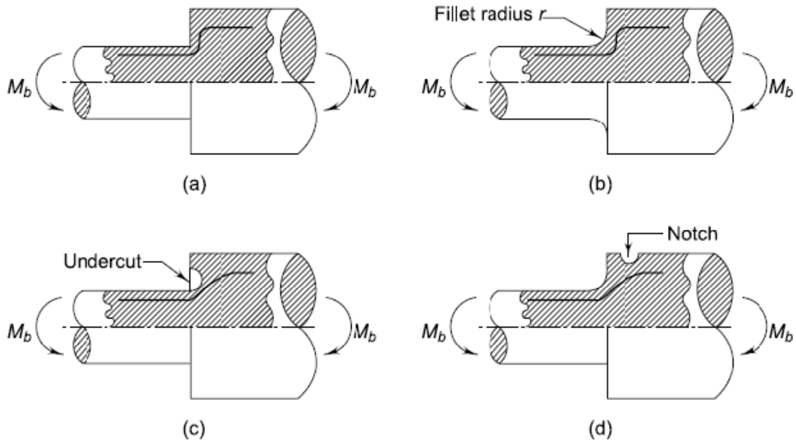

Fig.3.2: Reduction of Stress Concentration due to Abrupt Change in Cross-section: (a) Original Component(b) Fillet Radius (c) Undercutting (d) Addition of Notch

CHAPTER-4

4.MODELLING

Design models are required to evaluate, manipulate and refine the design. The engineering designer has to model the function of a design, its structure, and the form of component parts, materials, surface conditions and the dimensions. Geometric modelling is classified into 3 types. They are:

1. Wire frame modelling.
2. Surface modelling.
3. Solid modelling.

4.1. WIRE FRAME MODELLING:

Wireframe modelling is one of the methods used in geometric modelling systems.
Wireframe modelling is the process of visual presentation of a three-dimensional or physical object used in 3-D computer graphics. It is an abstract edge or skeletal representation of a real-world 3-D object using lines and curves. Because each object that makes up a wireframe model must be independently drawn and positioned, this type of modelling can be extremely time-consuming.

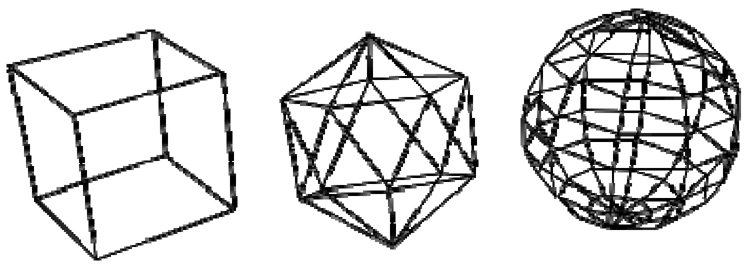

Fig. 4.1. WireFrame Modelling

4.2. SURFACE MODELLING:

Surface modelling is a mathematical technique for representing solid-appearing objects. Surface modelling is a more complex method for representing objects than wireframe modelling, but not as sophisticated as solid modelling. Surface modelling is widely used in CAD (computer-aided design), solid works, for illustrations and architectural renderings. It is also used in 3D animation for games and other presentations.

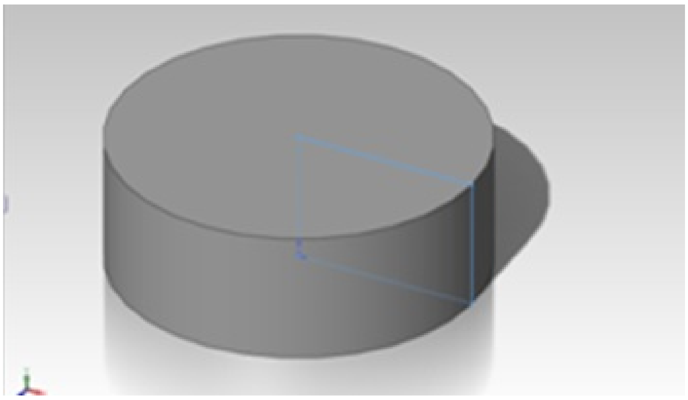

Fig. 4.2. Surface Modelling

4.3. SOLID MODELLING:

Solid modelling is the most advanced method of geometric modelling in three dimensions. Solid modelling is the representation of the solid parts of the object on your computer. The typical geometric model is made up of wire frames that show the object in the form of wires. This wire frame structure can be two dimensional, two and half dimensional or three dimensional. Providing surface representation to the wire three dimensional views of geometric models makes the object appear solid on the computer screen and this is what is called as solid modelling.

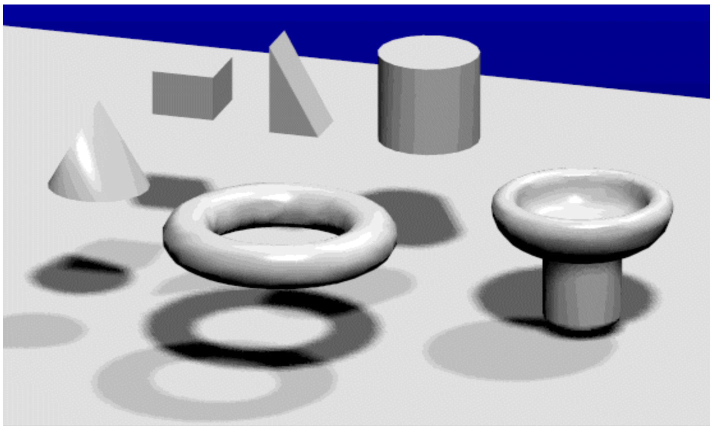

Fig. 4.3. Solid Modelling

CHAPTER-5

5.SOLID WORKS

5.1. INTRODUCTION TO SOLIDWORKS:

The Solid works CAD software is a mechanical design automation application that lets designers quickly sketch out ideas, experiment with features and dimensions, and produce models and detailed drawings.

SolidWorkis design automation software. In SolidWorks, you sketch ideas and experiment with different designs to create 3D models. SolidWorks is used bystudents, designers, engineers, and other professionals to produces complexparts, assemblies, anddrawings.

5.2. SOLIDWORKS FUNDAMENTALS:

Parts are the basic building blocks in the solid works software. Assemblies contain parts or other assemblies, called sub-assemblies. A solid works model consists of 3D geometry that defines its edges, faces, and surfaces .the solid work software lets you design models quickly and precisely. Solid work models are

- Defined by 3D design
- Based on components
- 3D design

Solid works uses a 3D design approach as you design a part from the initial sketch to the final result, you create a 3D model .From this model, you can create a 2D drawings or mate components consists of parts and subassemblies to create a 3D assemblies. You can also create 2D drawings of 3D assemblies. When designing a model using solid works, you can visualize it in three dimensions, the way the model exists once it is manufactured.

- Component Based : One of the most powerful features in the solidworks application is that any change you make to a part is reflected in all associated drawings or assemblies.
- Origin : Appears as two blue arrows and represent the (0,0,0) coordinate of the model. When a sketch is active, a sketch origin origin appears in red and represents the (0,0,0) coordinate of the sketch.
- Plane : You can use planes for adding 2D sketch, section view of a model, or a neutral plane in a draft feature.
- Axis : Straight line used to create model geometry, features, or patterns. You can create an axis in different ways, including intersecting two planes. The solidworks application creates temporary axes implicitly for every conical or cylindrical face in a model.
- Face : Boundaries that help to define the shape of a model or a surface. A face is a selectable area (planar or non-planar) of a model or surface.
- Edge : Locate where two or more faces intersect and are joined together.
- Vertex : Point at which two or more lines or edges intersect. You can select vertices for sketching and dimensioning.
- User Interface : The Solidworks application includes user interface tools and capabilities to help you create and edit models efficiently.
- Windows Function : The Solidworks applications includes familiar windows functions, such as dragging and resizing windows. Many of the same icons, such as print, open, save, cut, and paste are also part of the solidworks application.
- SolidWorks Document Window: Solidworks document windows have two panels. The left panel, or Manager Panel, contains
- Feature Manager Design Tree: Displays the structure of the part, assembly, or drawing.
- Property Manager: Provides settings for many functions such as sketches, fillet features, and assembly mates.
- Configuration Manager: Create, select, view multiple configurations of parts and assemblies in a document.

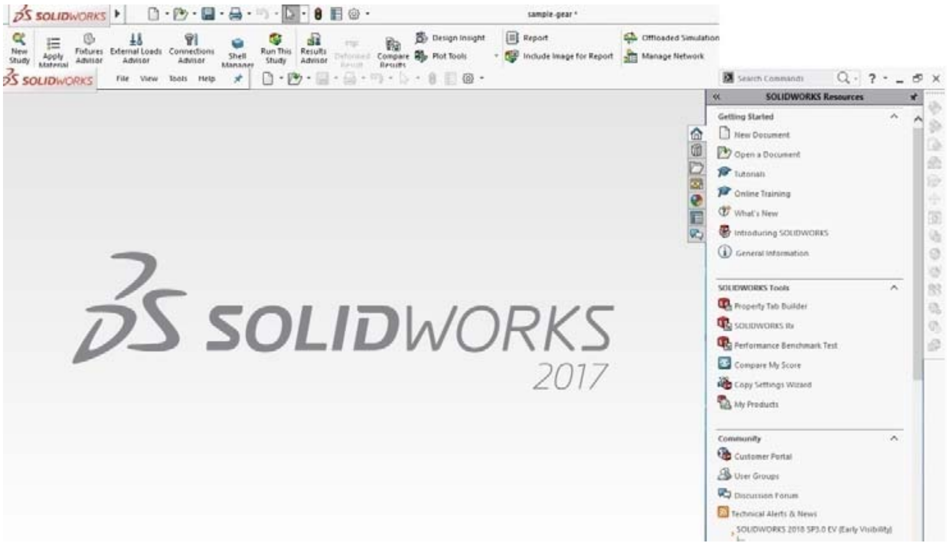

Fig. 5.1. SolidWorks

5.3. DESIGN OF GEAR TOOTH USING SOLIDWORKS

5.3.1. SPUR GEAR SPECIFICATIONS

Table.2: Parameters of Spur Gear

PARAMETER	SYMBOL	VALUE
No. of TOOTH	Z	40
Module	M	10mm
Pressure Angle	A	20^{o}
Circular Pitch	P	31.41mm
Pitch Circle Diameter	D	400mm
Face Width	B	50mm

Open the SOLIDWORKS interface and select sketch and select proper plane

- Then select circle from the sketch tools and draw two concentric circles and give their dimensions using smart dimensions.

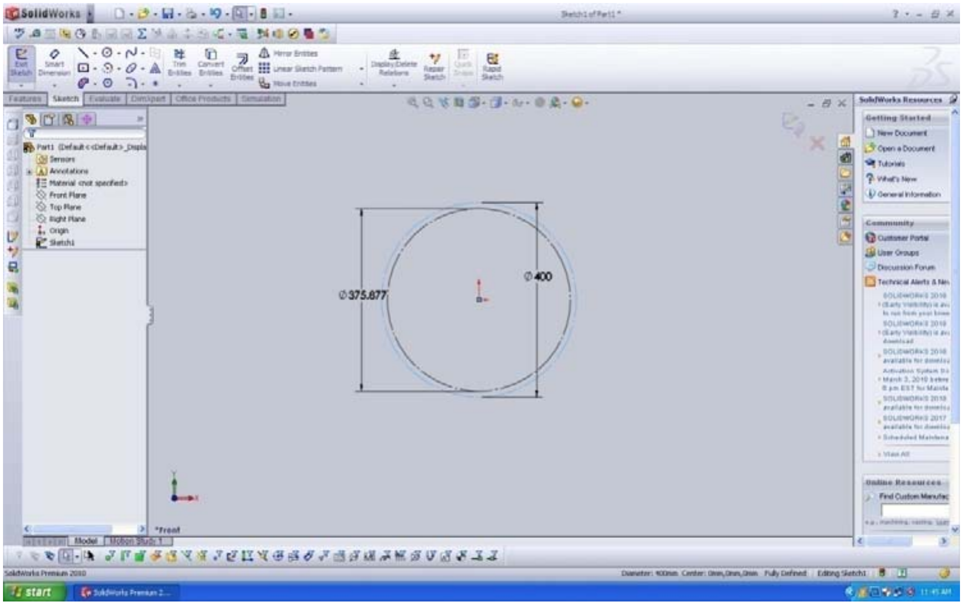

Fig. 5.2. Design of Spur Gear tooth

- Then draw a line at 40 degrees to the vertical and create the similar lines along the arc length by using " circular sketch pattern".
- Provide equal spacing between the lines and the number of parts is selected as 8.
- Then draw tangents at the intersection points of lines and base circle .
- The length of tangents will be in such a way that it is equal to the arc length between corresponding line and vertical line
- The arc lengths can be found by using the "smart dimension feature''.
- Now the dimensions of the lines are adjusted to that of arc length again by using "smart dimension" .

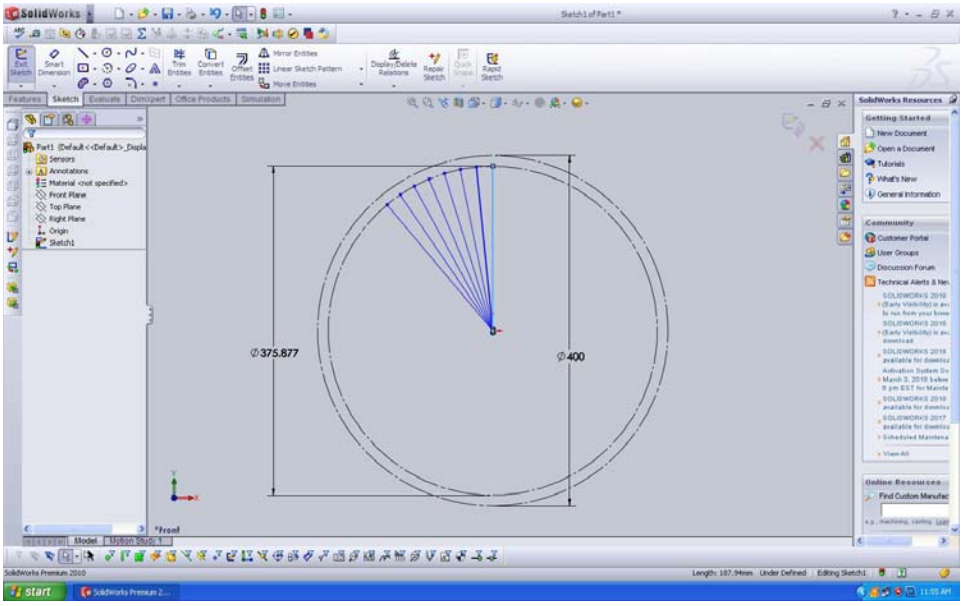

Fig. 5.3. Design of Spur Gear tooth

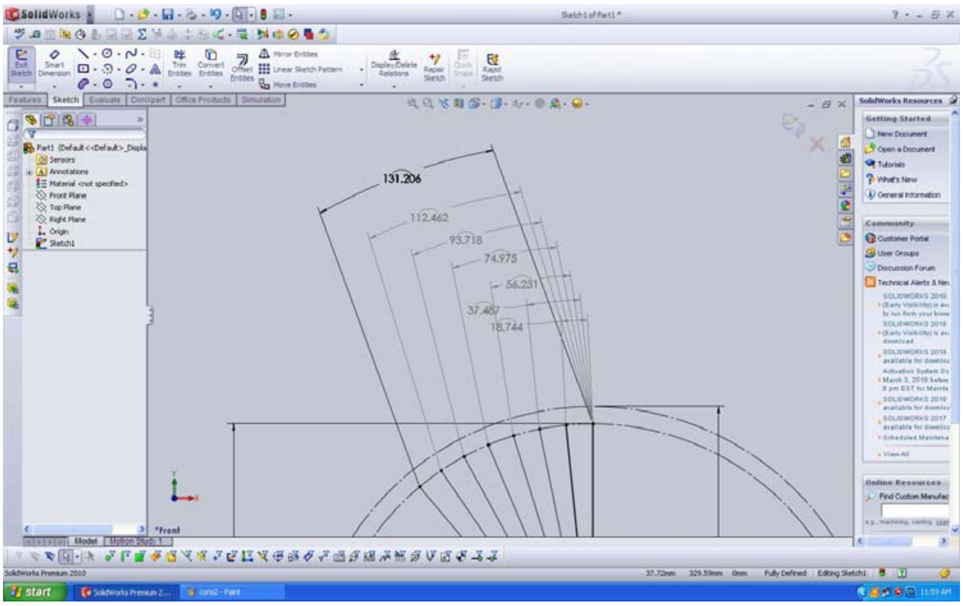

Fig. 5.4. Design of Spur Gear tooth

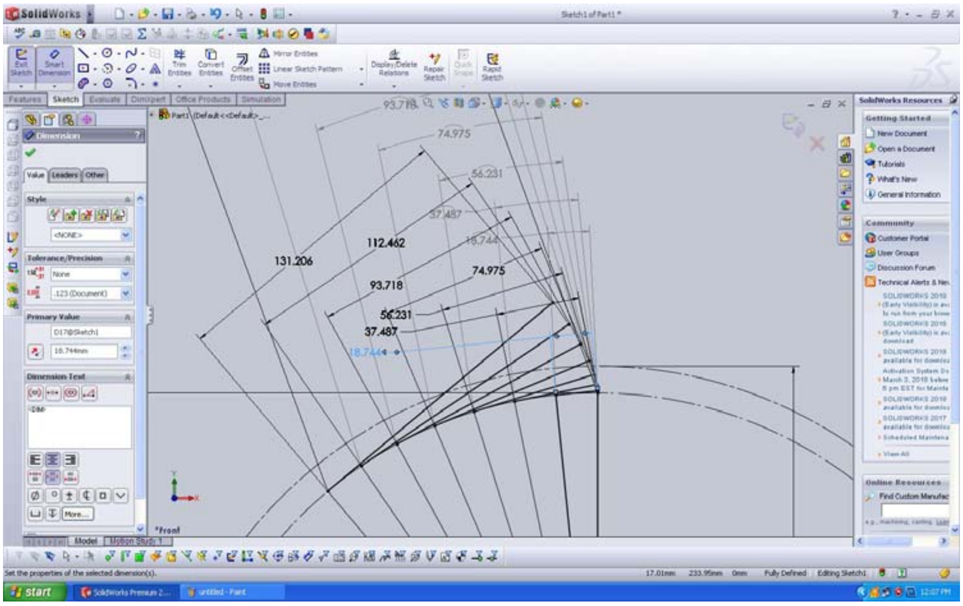

Fig. 5.5. Design of Spur Gear tooth

- Now join the end points of the tangents by using spline curve.

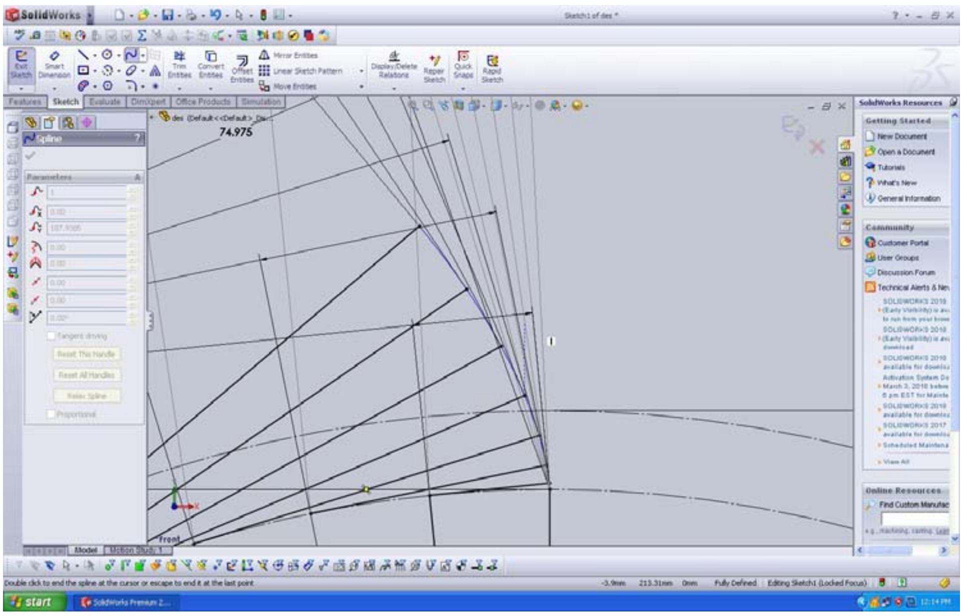

Fig. 5.6. Design of Spur Gear tooth

- After joining the end points of tangents, now draw a construction line with an angle of “half TOOTH angle” of required gear.
- Then mirror the spline portion about the construction line to get the involute portion on the other side using “mirror entities”.

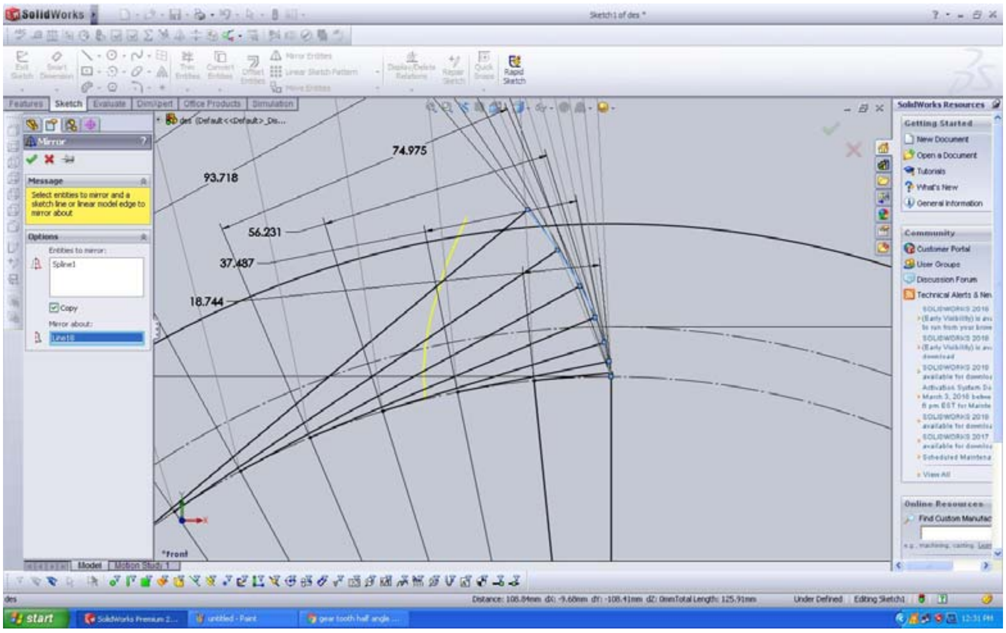

Fig. 5.7. Design of Spur Gear tooth

- Also draw the addendum circle with an offset equal to module by using “offset entities”.

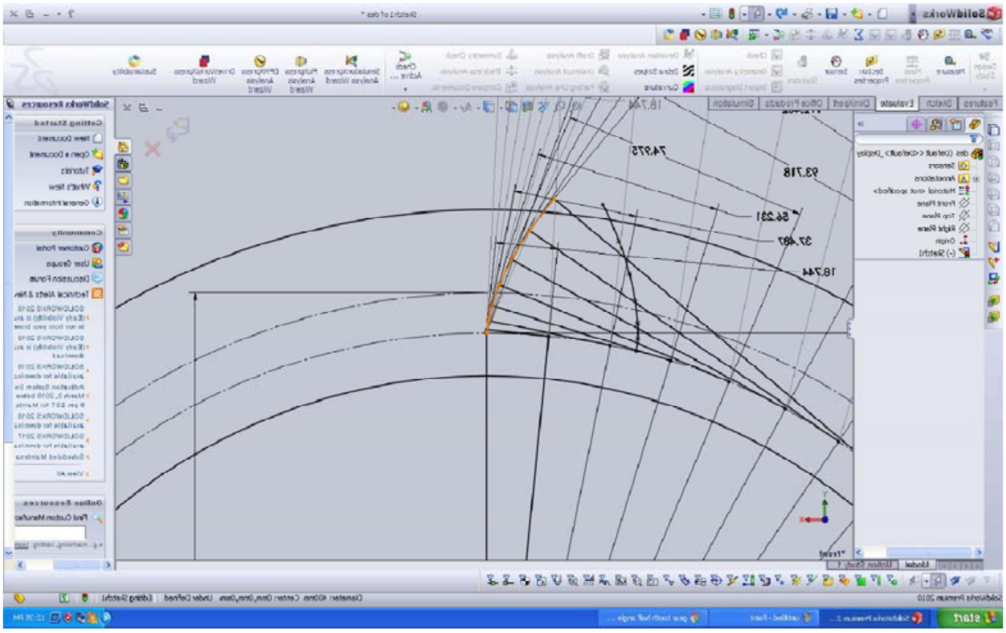

Fig. 5.8. Design of Spur Gear tooth

- Follow the same procedure to draw deddendum circle.
- Now trim all the unwanted portions on the sketch using “trim”.

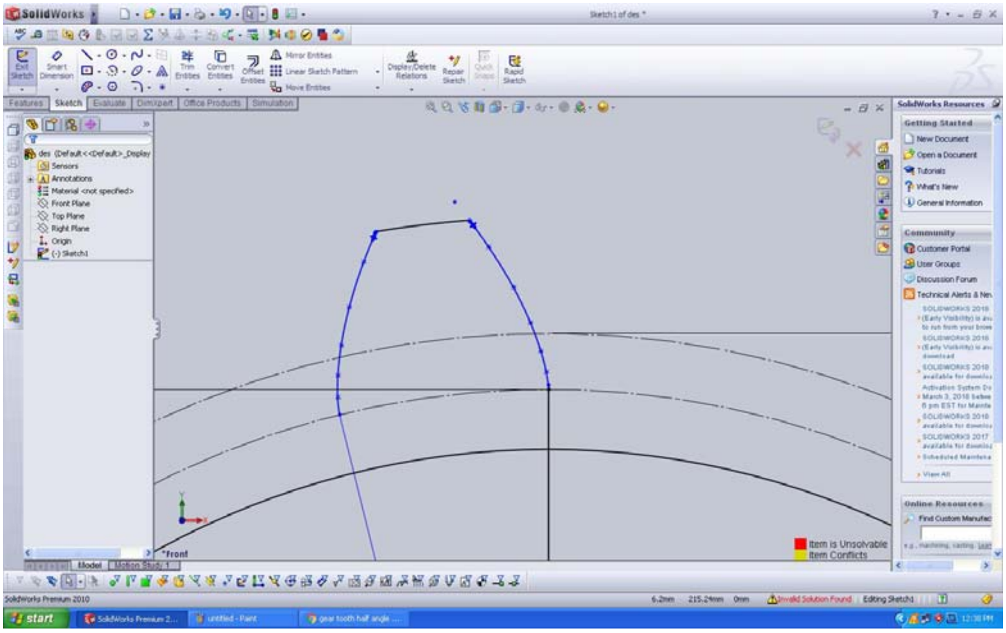

Fig. 5.9. Design of Spur Gear tooth

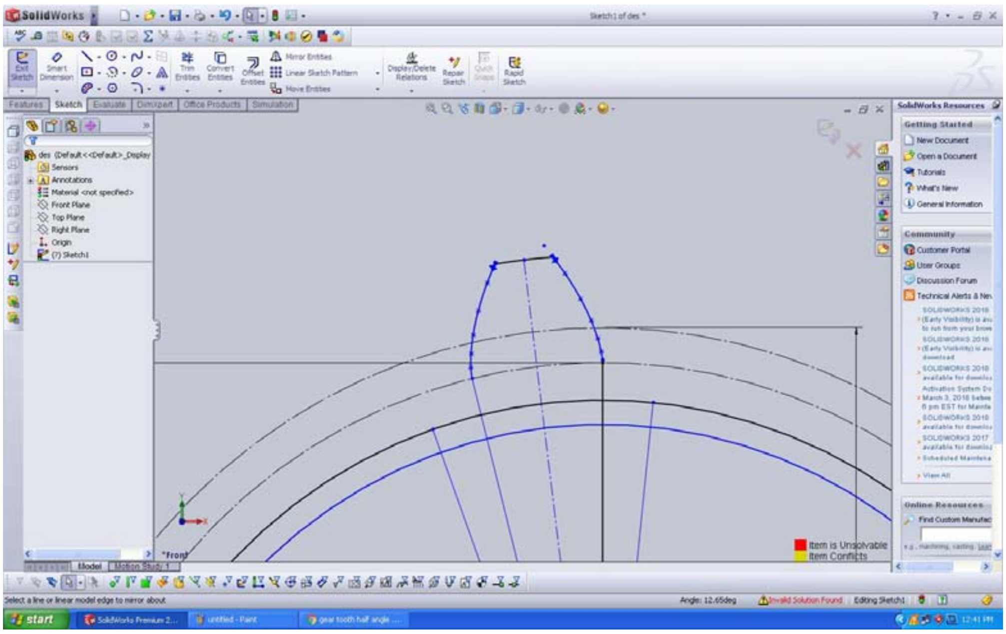

Fig. 5.10. Design of Spur Gear tooth

- Now close the contour of the required gear TOOTH portion and again trim all the unwanted curves and lines.

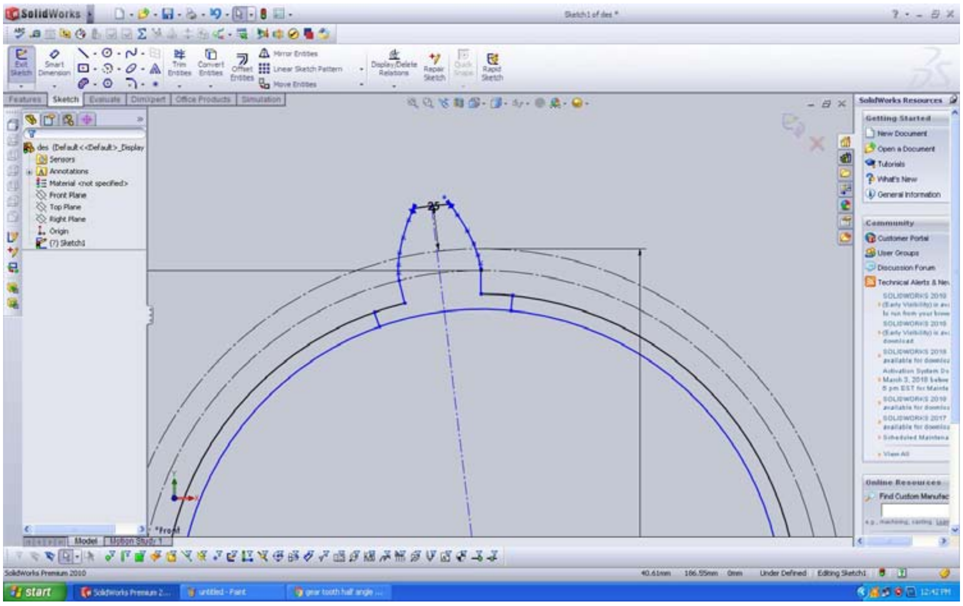

Fig. 5.11. Design of Spur Gear tooth

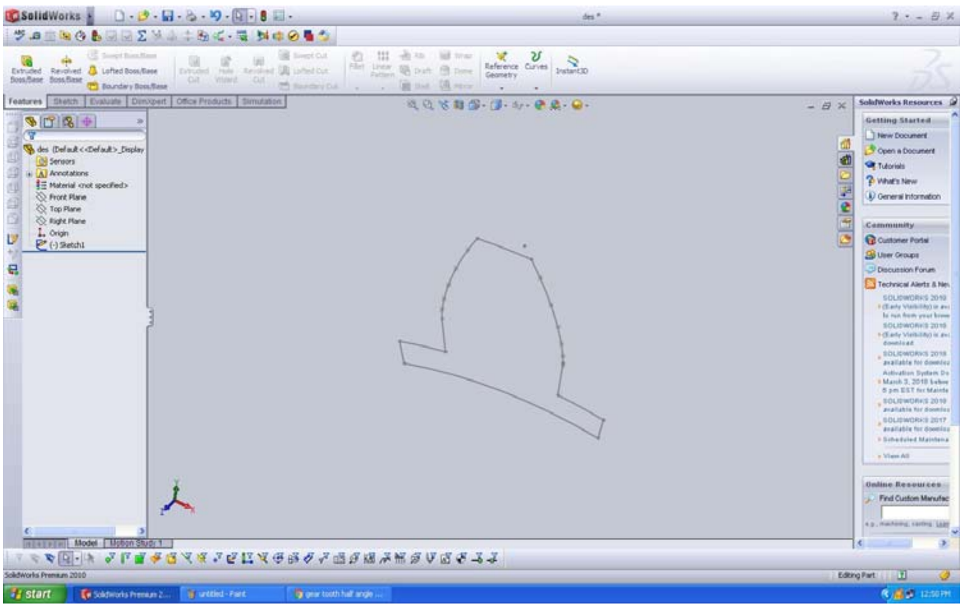

Fig. 5.12. Design of Spur Gear tooth

- Now after getting the required portion of the gear TOOTH, extrude the portion to a length equal to face width of the gear tooth.

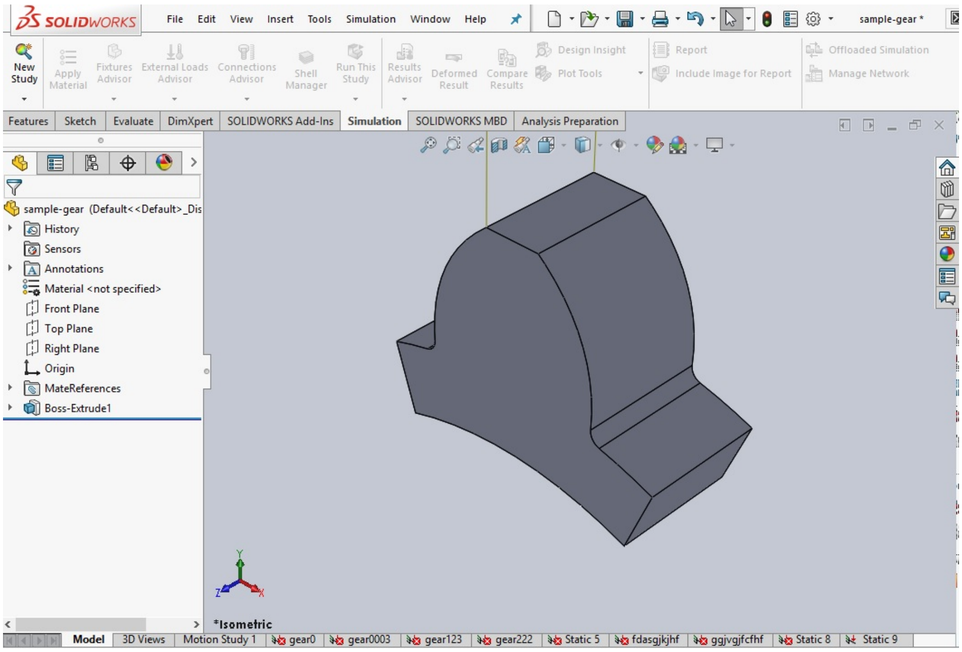

Fig. 5.13. Extruding the Gear tooth to Face Width

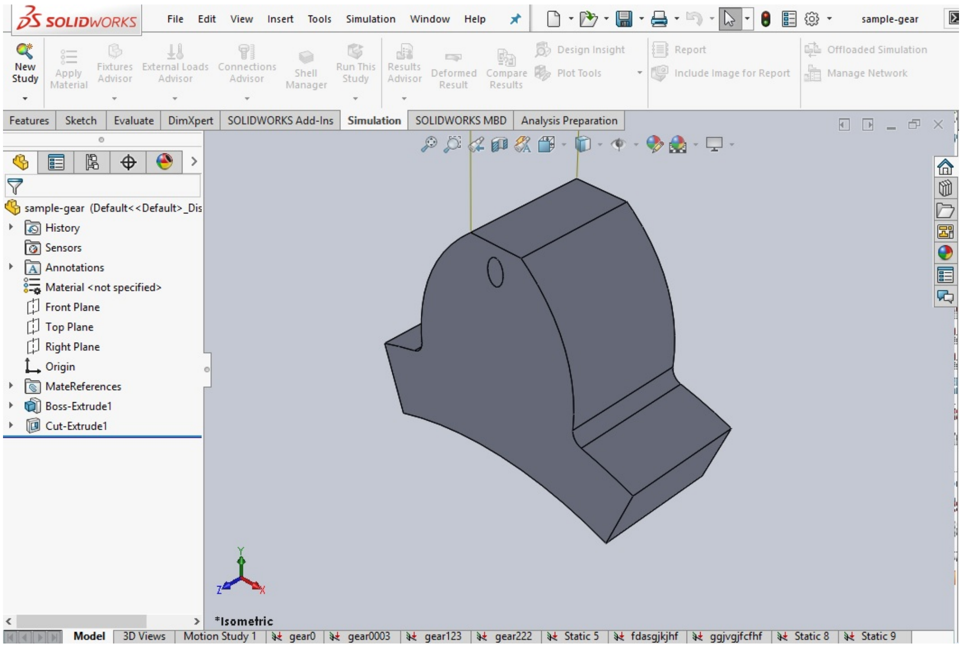

Fig. 5.14. Extruding Hole in Gear tooth

CHAPTER-6

6.SIMULATION USING SOLIDWORKS

6.1. INTRODUCTION TO FEM:

The finite element method (FEM) has now become a very important tool of engineering analysis. Its versatility is reflected in its popularity among engineers and designers belong to nearly all the engineering disciplines. Whether a civil engineer designing bridges, dams or mechanical engineers designing auto engines, rolling mills, machine tools or an aerospace engineer interested in the analysis of dynamics of an aero plane or temperature rise in the heat shield of a space shuttle or a metallurgist concerned about the influence of a rolling operation on the microstructure of a rolled product or an electrical engineer interested in analysis of the electromagnetic field in an electric machinery all find the finite element method handy and useful. it is not that these problems remained un prove before the finite element method came into vogue; rather this method has become popular due to its relative simplicity of approach and accuracy of results. Traditional methods of engineering analysis, while attempting to solve an engineering problem mathematically, always try for simplified formulation in order to overcome the various complexities involved in exact mathematical formulation. In the modern technological environments the conventional methodology of design cannot complete with the modern trends of Computer Aided Engineering (CAE) TECHNIQES. To build highly optimized product, this is the basic requirement of survival in the global market today. All round efforts were put forward in this direction. Software professional and technologists have developed various design packages.

6.2. NEED AND OBJECTIVE:

The stress analysis in the fields of civil, mechanical and aerospace engineering, nuclear engineering is invariably complex and for many of the problems it is extremely difficult and tedious to obtain analytical solutions. In these situations engineers usually resort to numerical methods to solve the problems. With the advent of computers, one of the most powerful techniques that have been developed in the engineering analysis is the finite element method and the method being used for the analysis of structures/solids of complex shapes and complicated boundary conditions. Due to development of computers and subsequent development of numerical methods, it is now possible to model to components, simulate the conditions and perform testing on computer without actual model making, some of the most popular numerical methods used in the Finite Element (FEM) offered by existing CAD/CAM/CAE, SOLID WORKS.

6.3. USE SOLIDWORKS SIMULATION TO PERFORM STATIC ANALYSIS ON THEDESIGNED GEAR TOOTH

- Save the designed gear in .SLDPRT format.
- Now name the study in the desired dialog box.

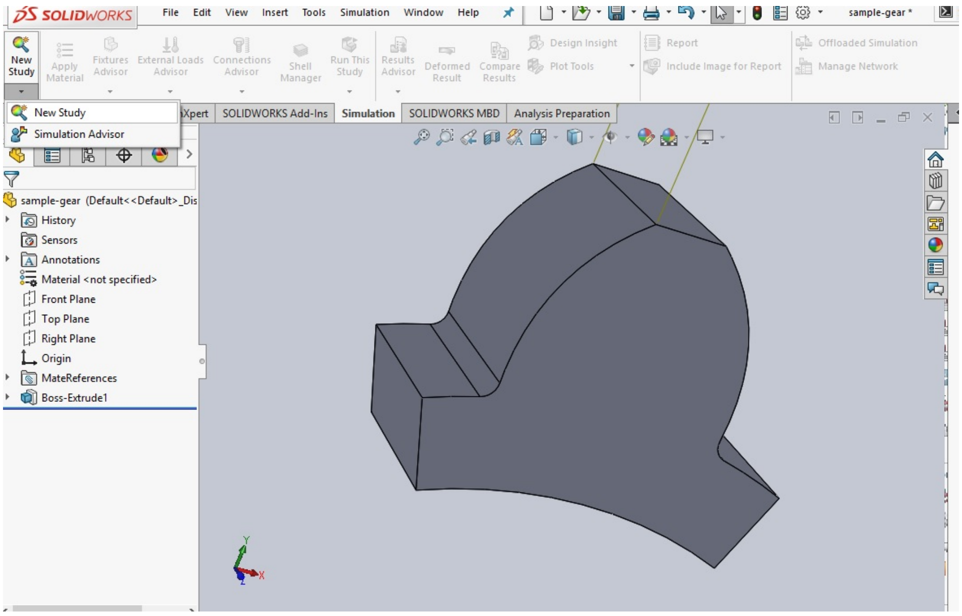

Fig. 6.1. Creating New Study

- Now apply material, in this analysis we have used 3 materials and they are Cast Carbon Steel(CCS), Grey Cast Iron, and Brass.

- Go to apply material to complete the procedure.

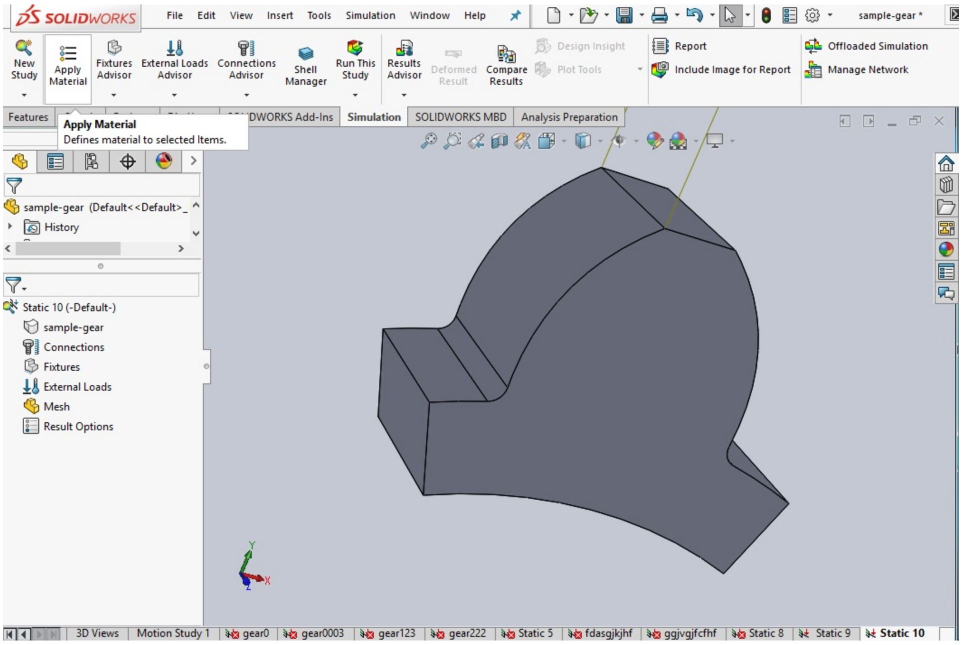

Fig. 6.2. Applying Material to the Gear Tooth

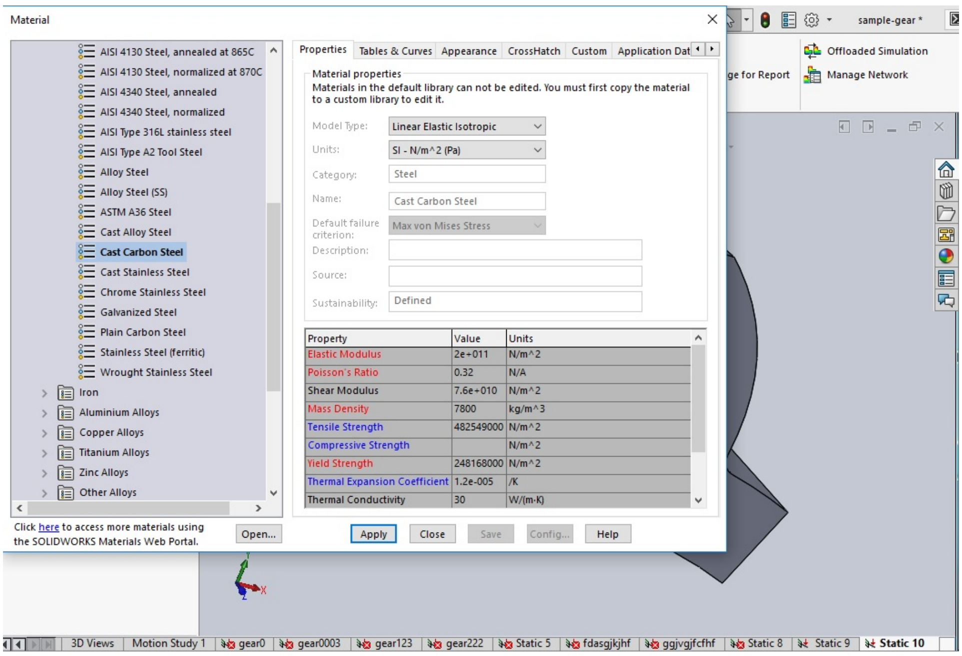

Fig. 6.3. Applying Material to the Gear tooth

- Now in order to fix the bottom curved face, Go to "Fixture Advisor" and click on the required face and then click 'OK'.

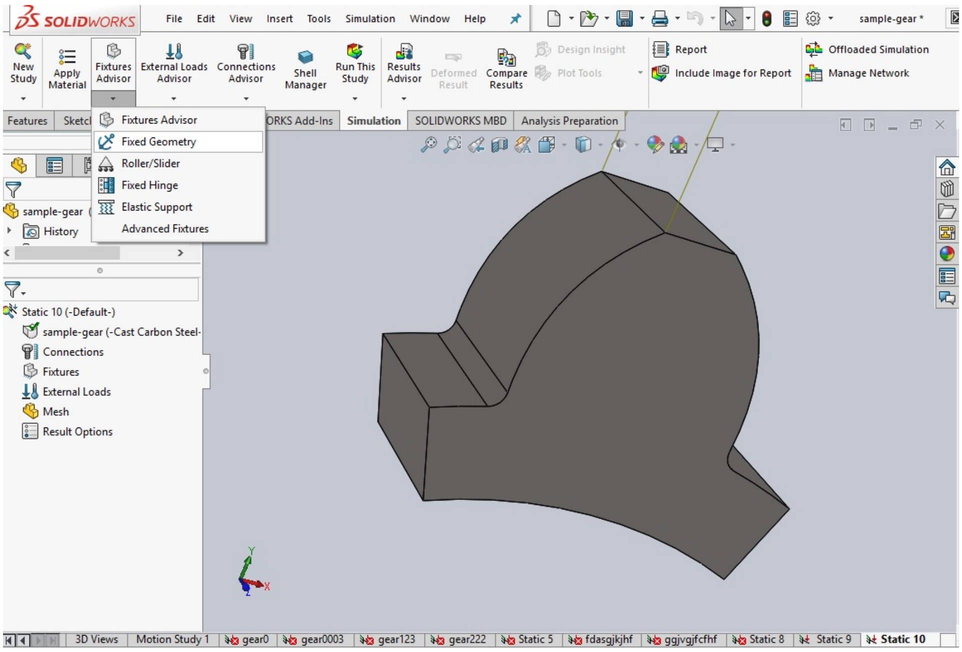

Fig. 6.4. Applying Fixtures

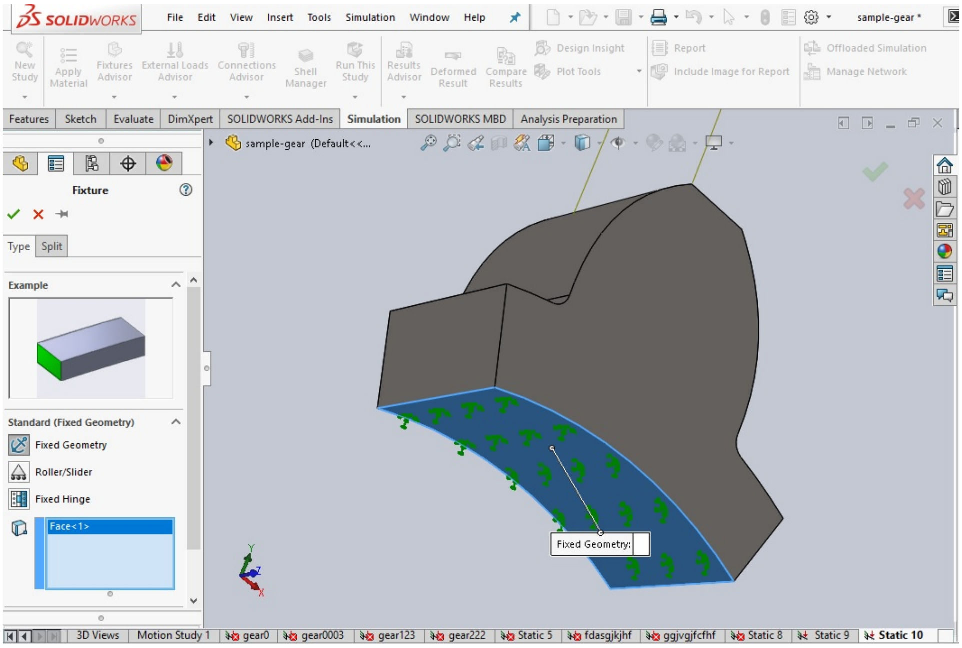

Fig. 6.5. Fixing the Geometry Face

- It is required to apply force for that Go to “External Load Advisor” and then select “force”.

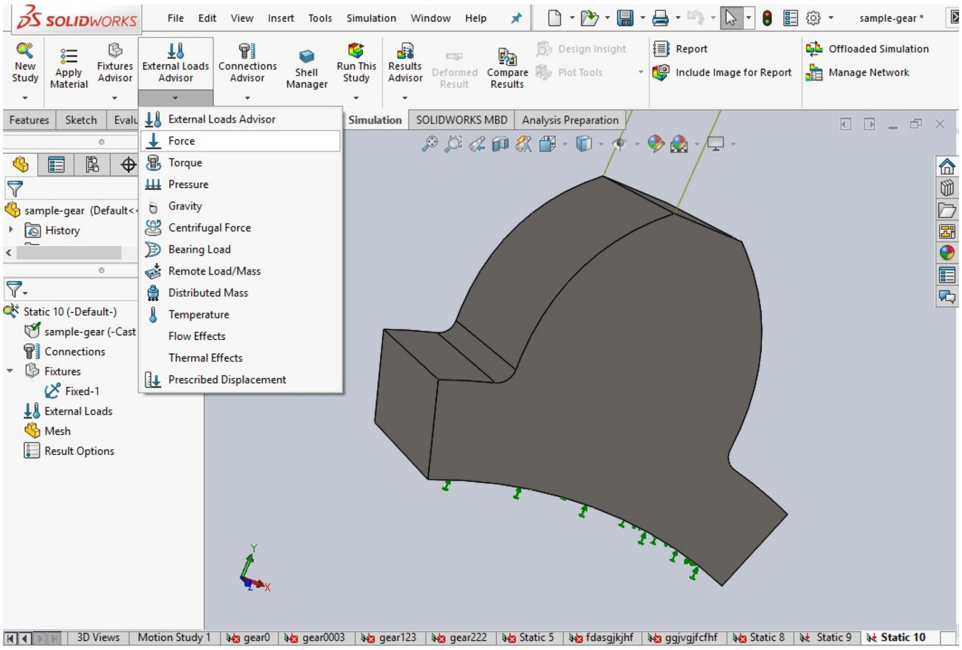

Fig. 6.6. Applying External loads

* Then select the required face in which the force is to be applied and enter the required magnitude i.e., the value of effective load.

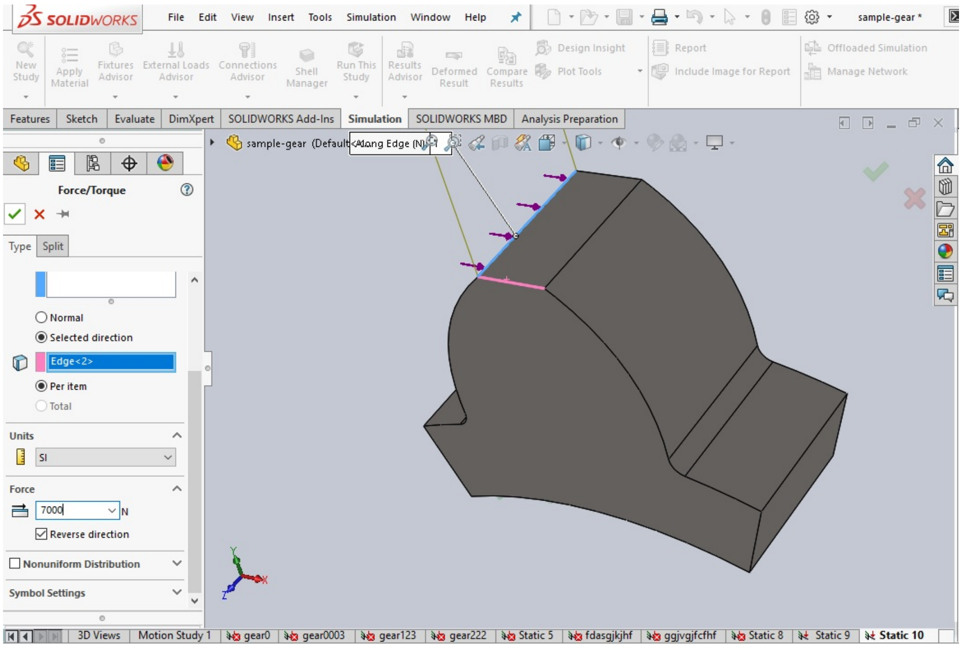

Fig. 6.7. Applying force

- Now select mesh and adjusting mesh size and click on “RUN”.

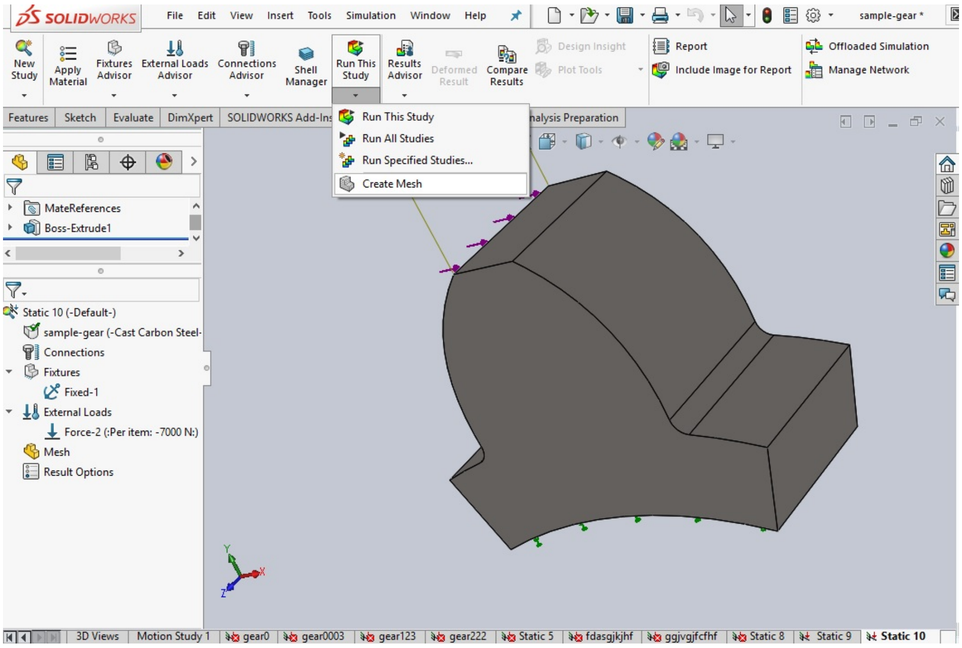

Fig. 6.8. Creating the Mesh

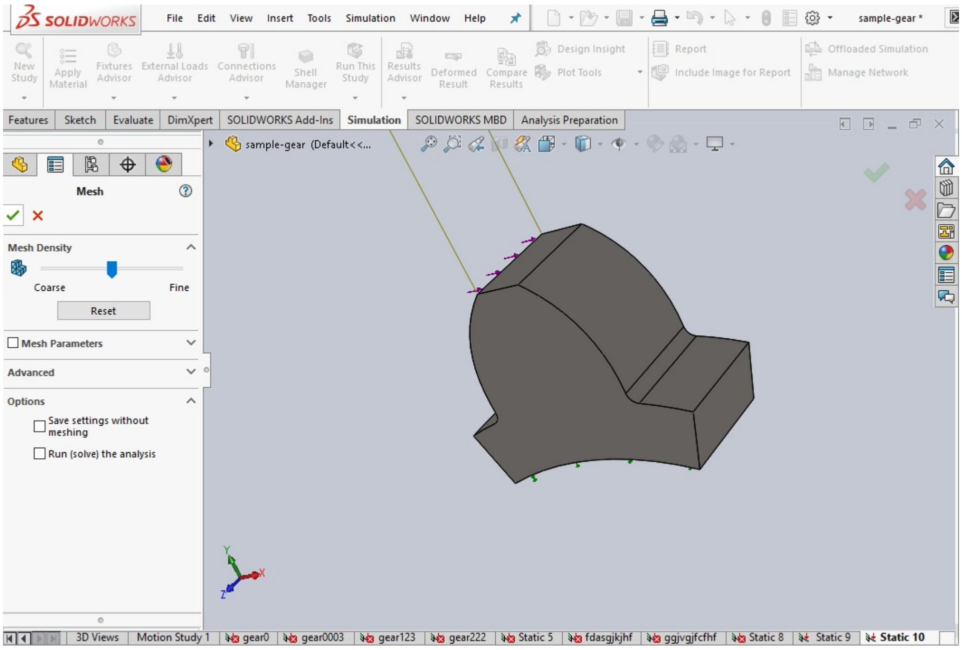

Fig.6.9. Adjusting the Mesh Size

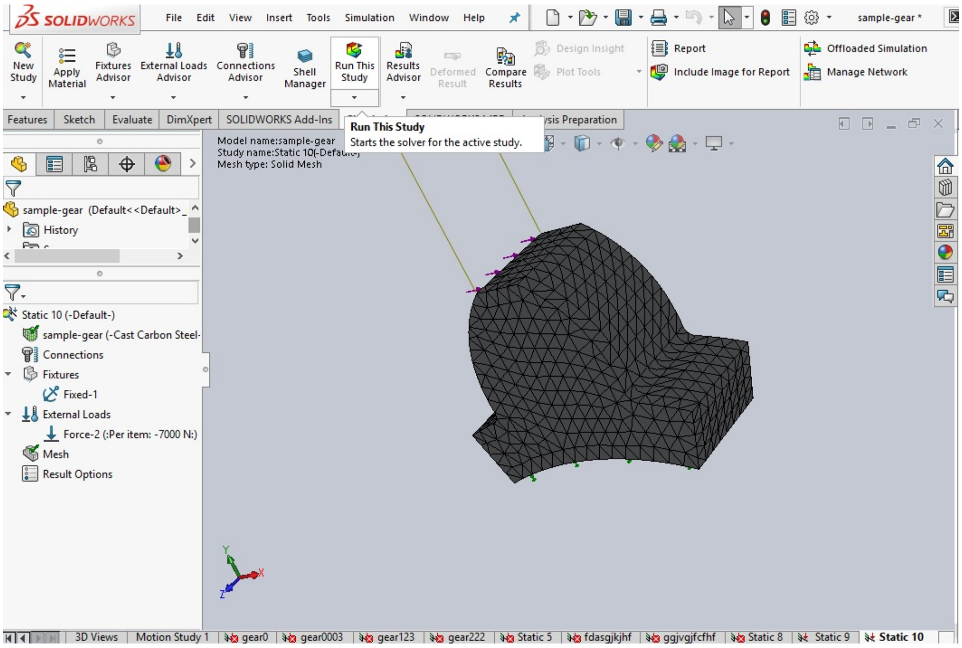

Fig.6.10. For Running the Study

- Go to “result advisor” then “new plot” and then click on “factor of safety”.

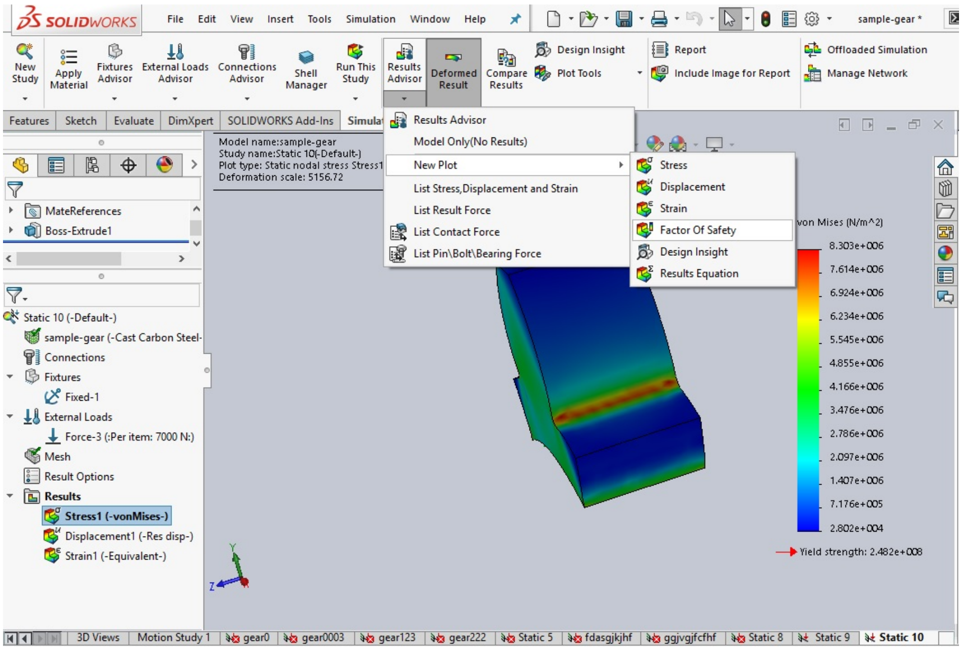

Fig. 6.11. For Factor of Safety Values

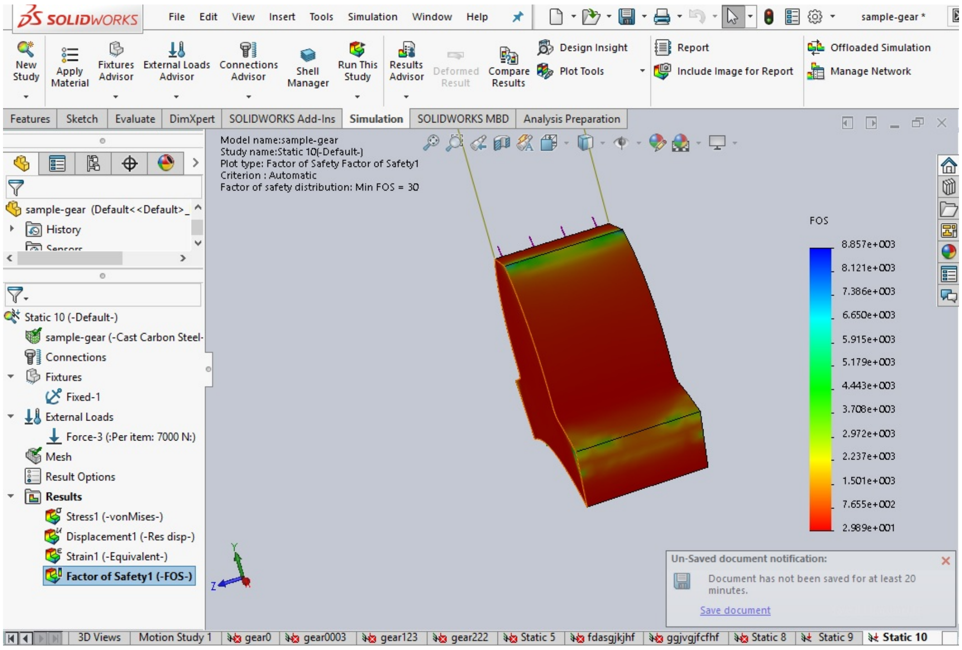

Fig. 6.12. Factor of Safety analysis

- Now repeat the same procedure by increasing the mesh density until the value of stress also converge to constant value.

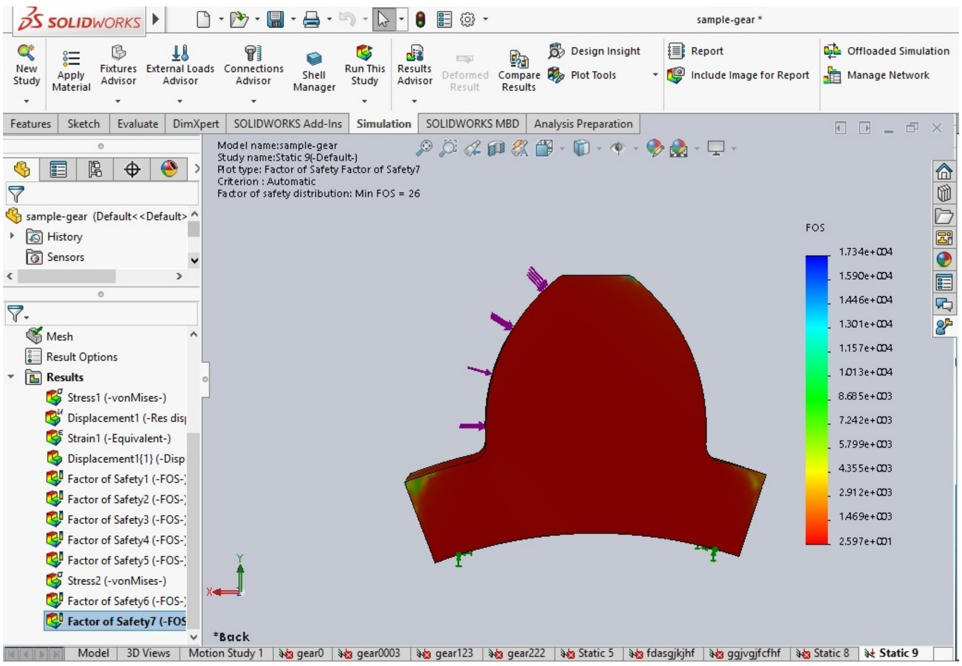

Fig. 6.13. Converging the Stress values

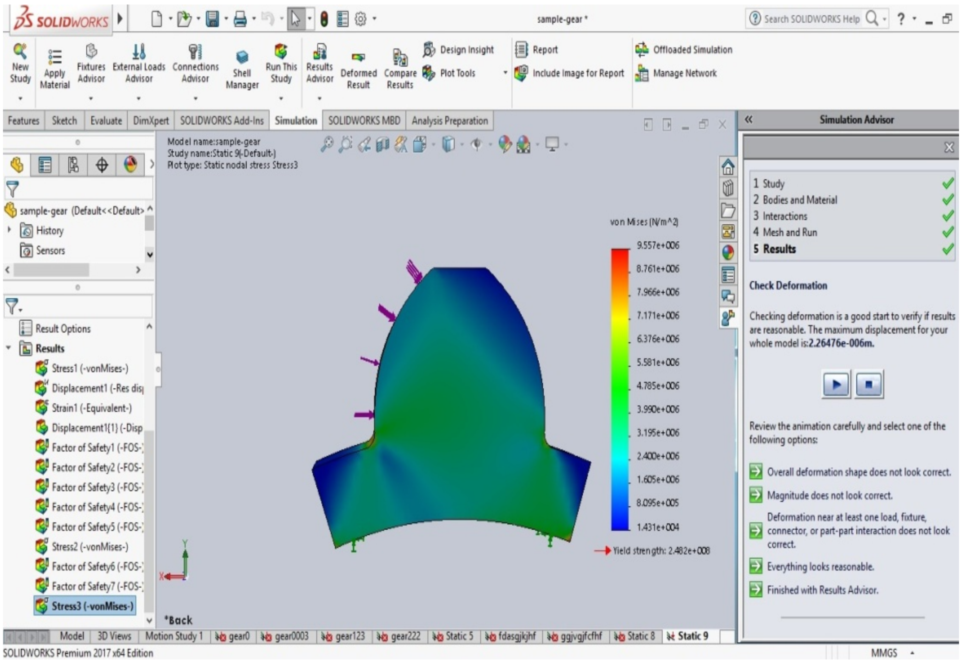

Fig. 6.14. Converging of Stress Values

6.4. "VON-MISES STRESSES" INDUCED IN GEAR TOOTH FOR VARIOUS MATERIALS:

The Von-Mises stresses induced in the gear tooth for some common materials, used in the manufacturing of gears, are shown in figures below.

6.4.1. GEAR TOOTH WITHOUT HOLE:

Gray Cast Iron:

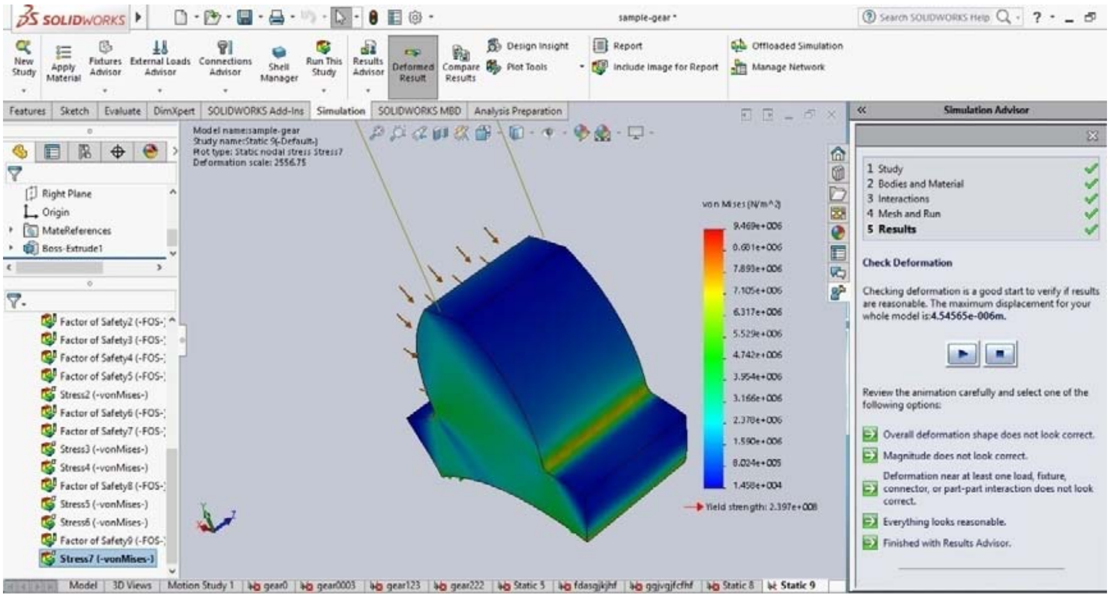

Fig. 6.15. Von-Mises Stresses in Gear Tooth for Gray Cast Iron(without hole)

Cast Carbon Steel:

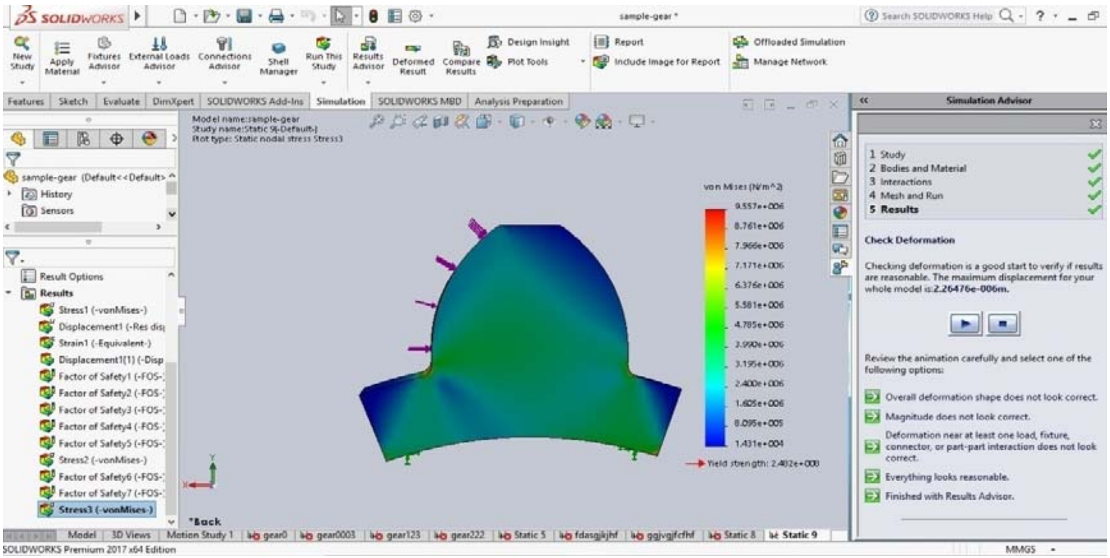

Fig. 6.16. Von-Mises Stresses in Gear Tooth for Cast Carbon Steel(without hole)

Brass:

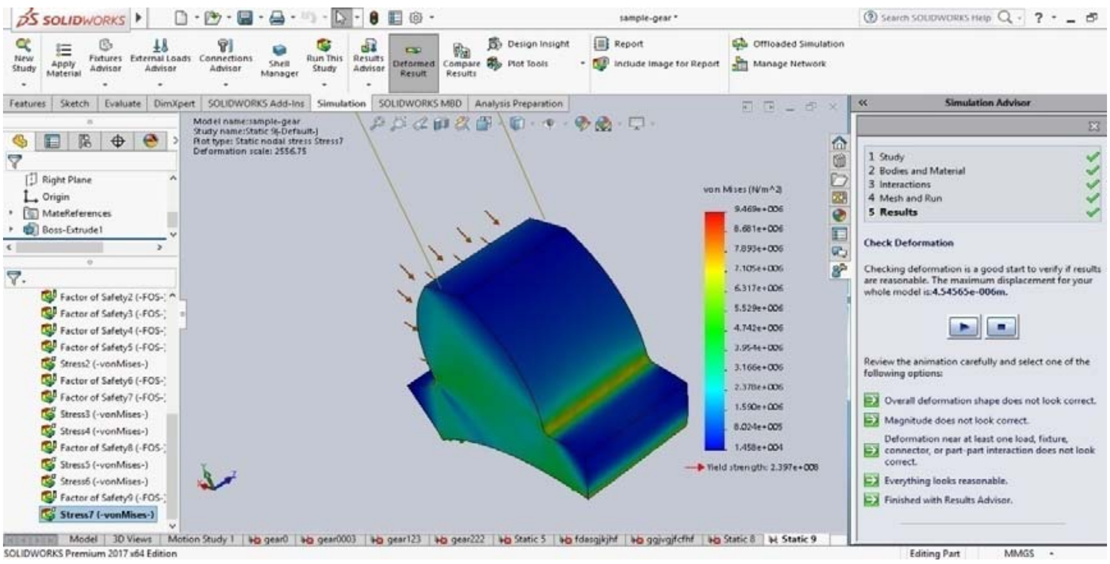

Fig. 6.17. Von-Mises Stresses in Gear Tooth for Brass(without hole)

6.4.2. GEAR TOOTH WITH A CIRCULAR HOLE OF DIA. 5MM AT A DISTANCE OF 5MM FROM TOP LAND

Gray Cast Iron:

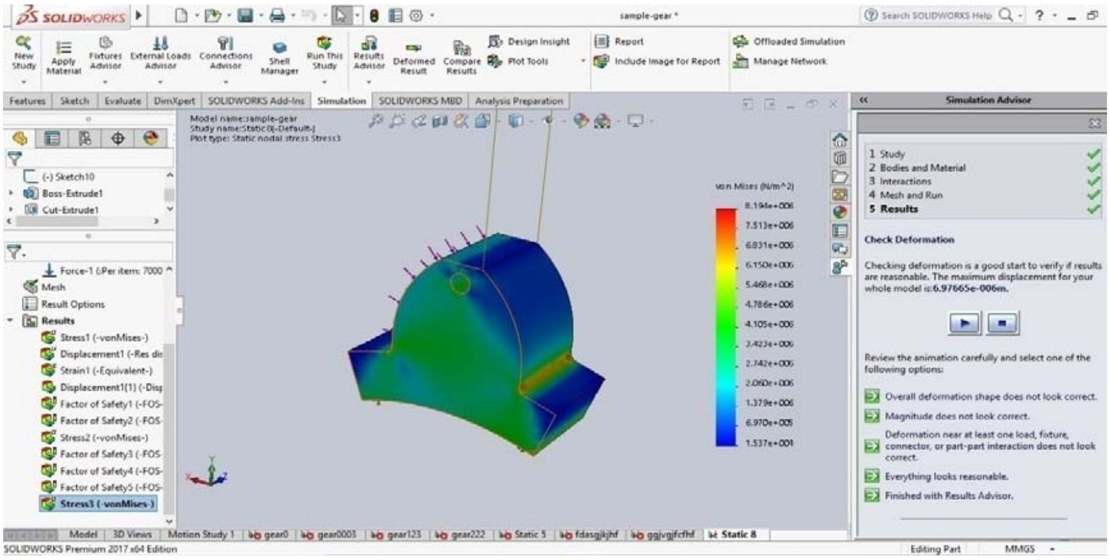

Fig. 6.18. Von-Mises Stresses in Gear Tooth for Gray Cast Iron(hole at 5mm dist.)

Cast Carbon Steel:

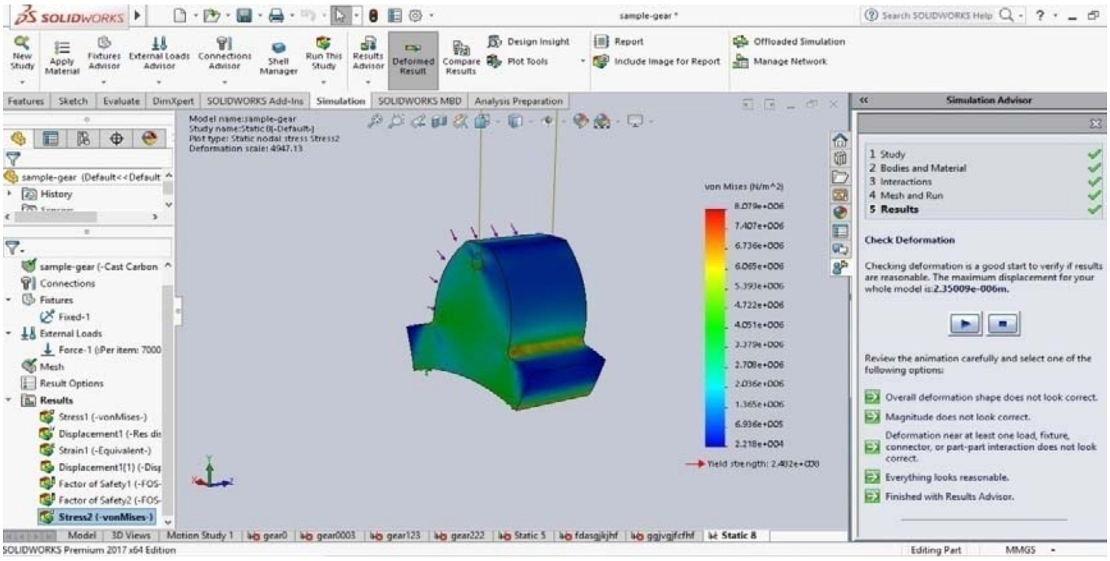

Fig. 6.19. Von-Mises Stresses in Gear Tooth for Cast Carbon Steel(hole at 5mm dist.)

Brass:

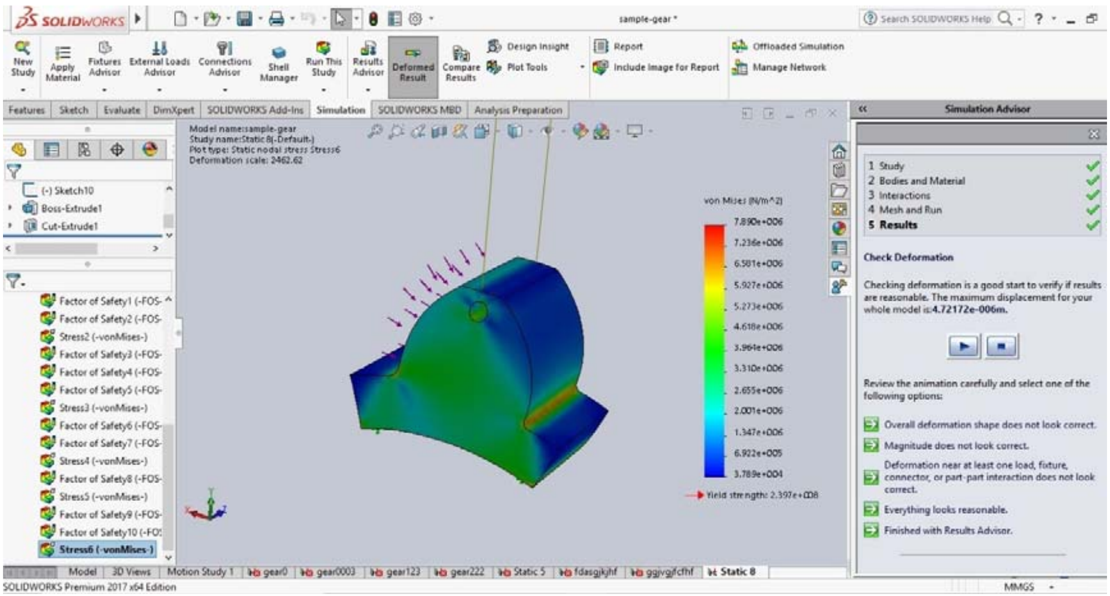

Fig. 6.20. Von-Mises Stresses in Gear Tooth for Brass(hole at 5mm dist.)

6.4.3. GEAR TOOTH WITH A CIRCULAR HOLE OF DIA. 5MM AT A DISTANCE OF 10MM FROM TOP LAND

Gray Cast Iron:

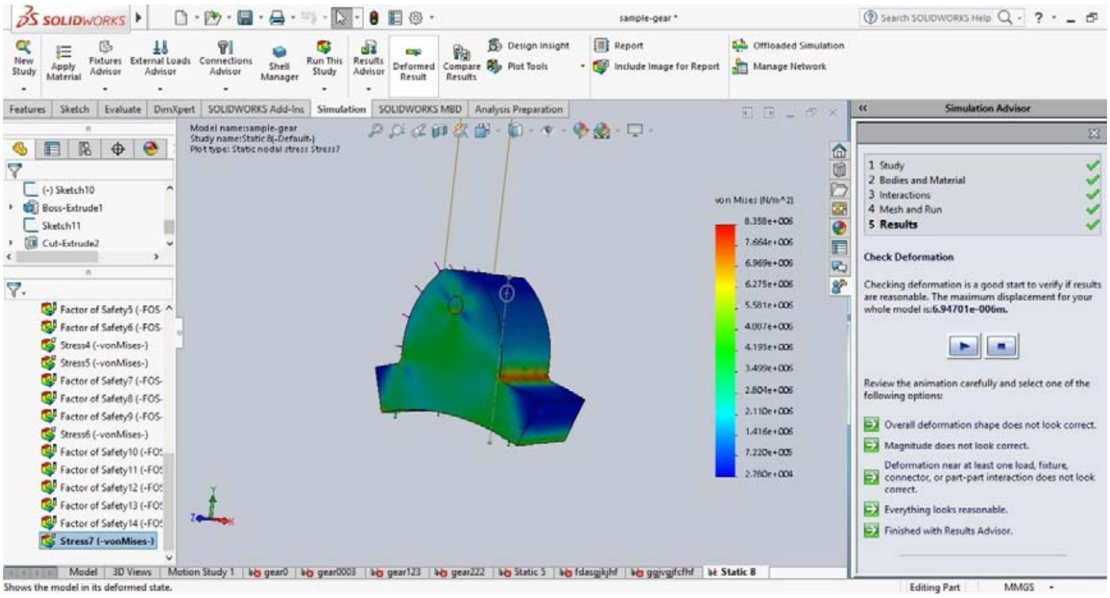

Fig. 6.21. Von-Mises Stresses in Gear Tooth for Gray Cast Iron(hole at 10mm dist.)

Cast Carbon Steel:

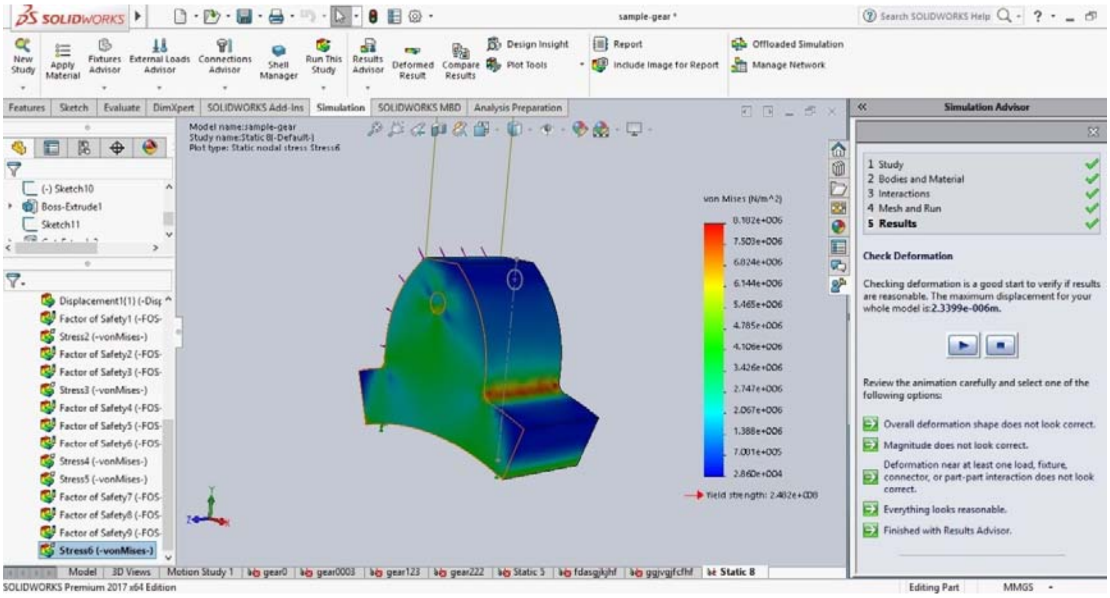

Fig. 6.22. Von-Mises Stresses in Gear Tooth for Cast Carbon Steel(hole at 10mm dist.)

Brass:

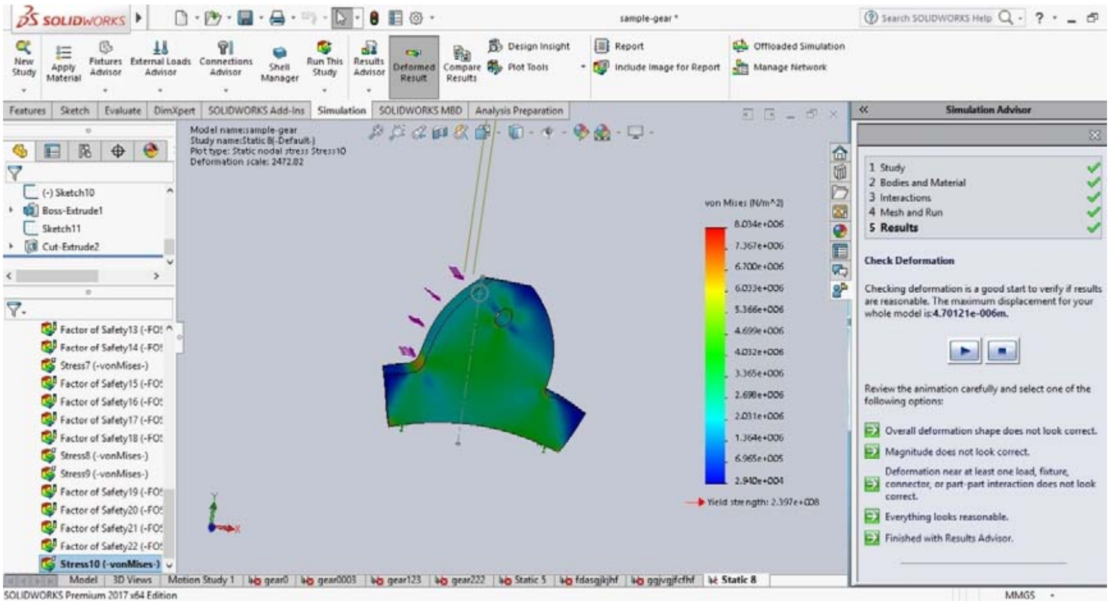

Fig. 6.23. Von-Mises Stresses in Gear Tooth for Brass(hole at 10mm dist.)

6.4.4. GEAR TOOTH WITH A CIRCULAR HOLE OF DIA. 5MM AT A DISTANCE OF 15MM FROM TOP LAND

Gray Cast Iron:

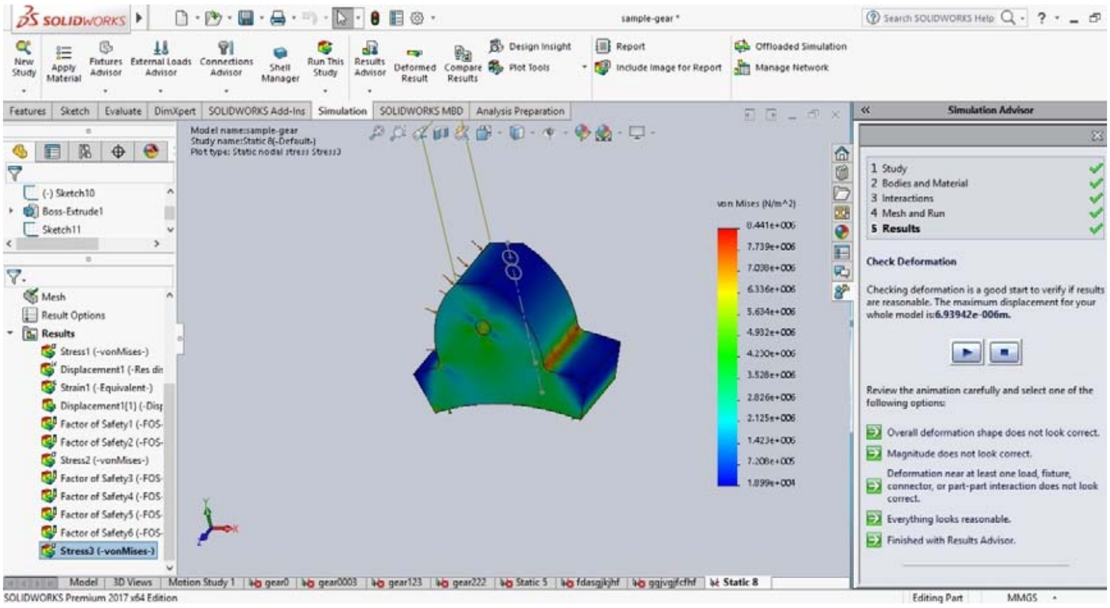

Fig. 6.24. Von-Mises Stresses in Gear Tooth for Gray Cast Iron(hole at 15mm dist.)

Cast Carbon Steel:

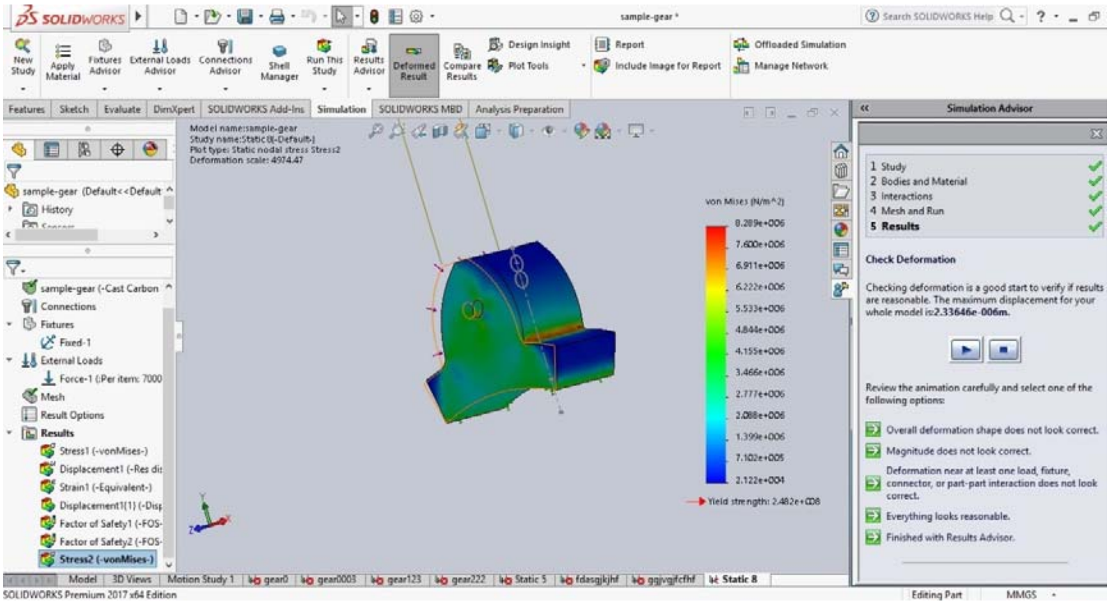

Fig. 6.25. Von-Mises Stresses in Gear Tooth for Cast Carbon Steel(hole at 15mm dist.)

Brass:

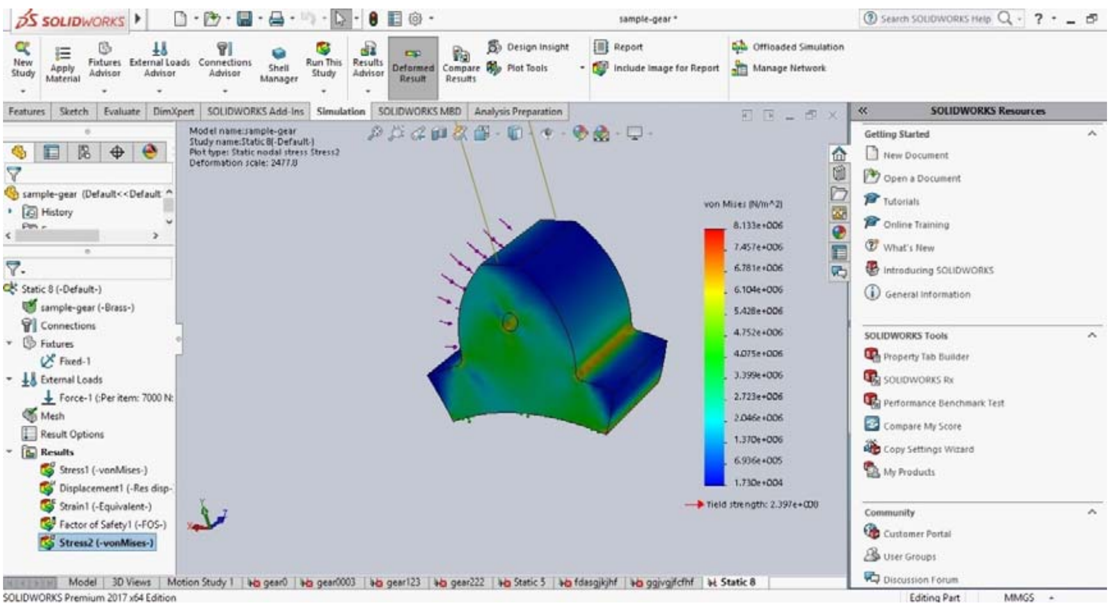

Fig. 6.26. Von-Mises Stresses in Gear Tooth for Brass(hole at 15mm dist.)

6.4.5. GEAR TOOTH WITH A CIRCULAR HOLE OF DIA. 5MM AT A DISTANCE OF 20MM FROM TOP LAND

Gray Cast Iron:

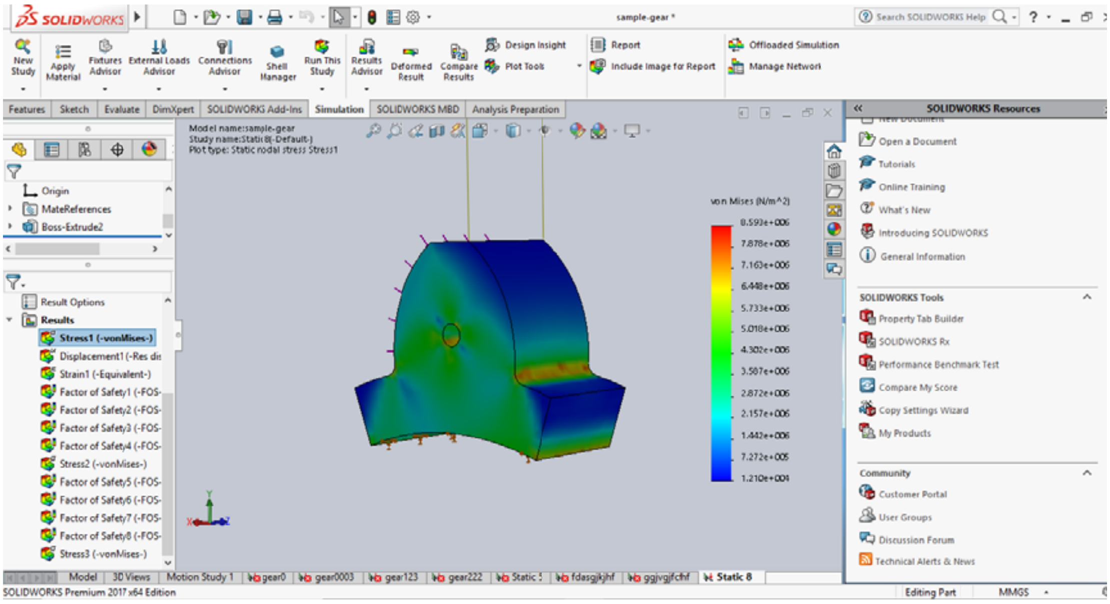

Fig. 6.27. Von-Mises Stresses in Gear Tooth for Gray Cast Iron(hole at 20mm dist.)

Cast Carbon Steel:

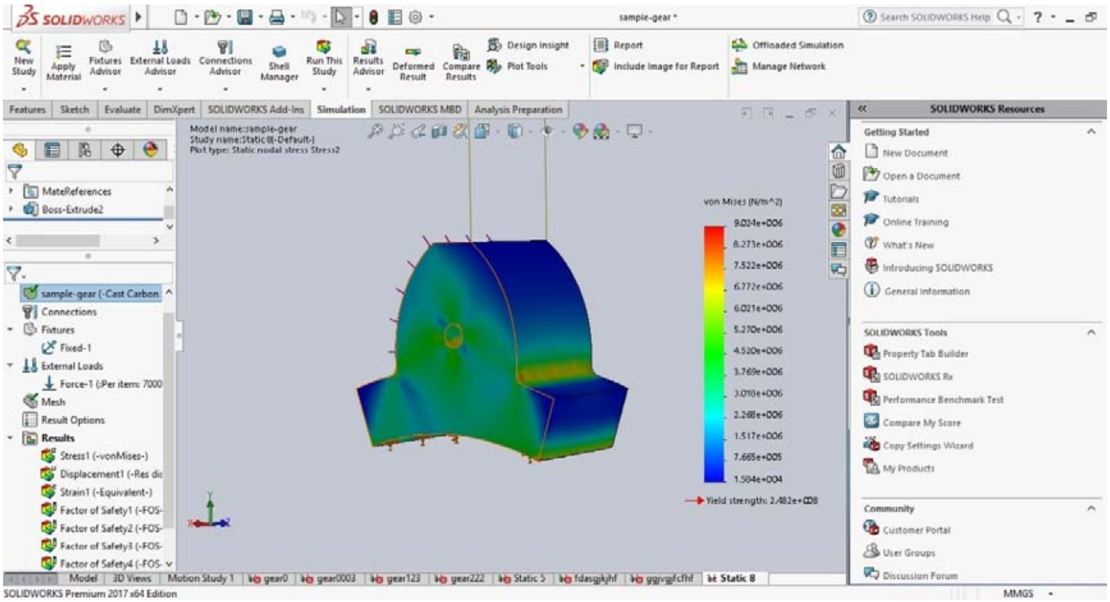

Fig. 6.28. Von-Mises Stresses in Gear Tooth for Cast Carbon Steel(hole at 20mm dist.)

Brass:

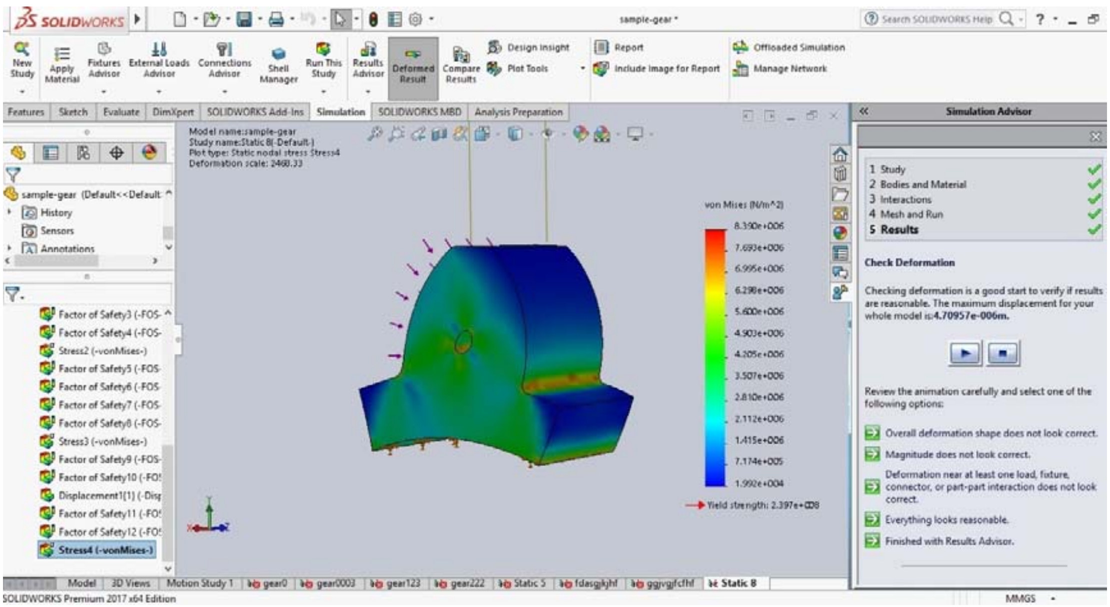

Fig. 6.29. Von-Mises Stresses in Gear Tooth for Brass(hole at 20mm dist.)

6.5. FACTOR OF SAFETY OF GEAR TOOTH FOR VARIOUS MATERIALS:

6.5.1. GEAR TOOTH WITHOUT HOLE:

Gray Cast Iron:

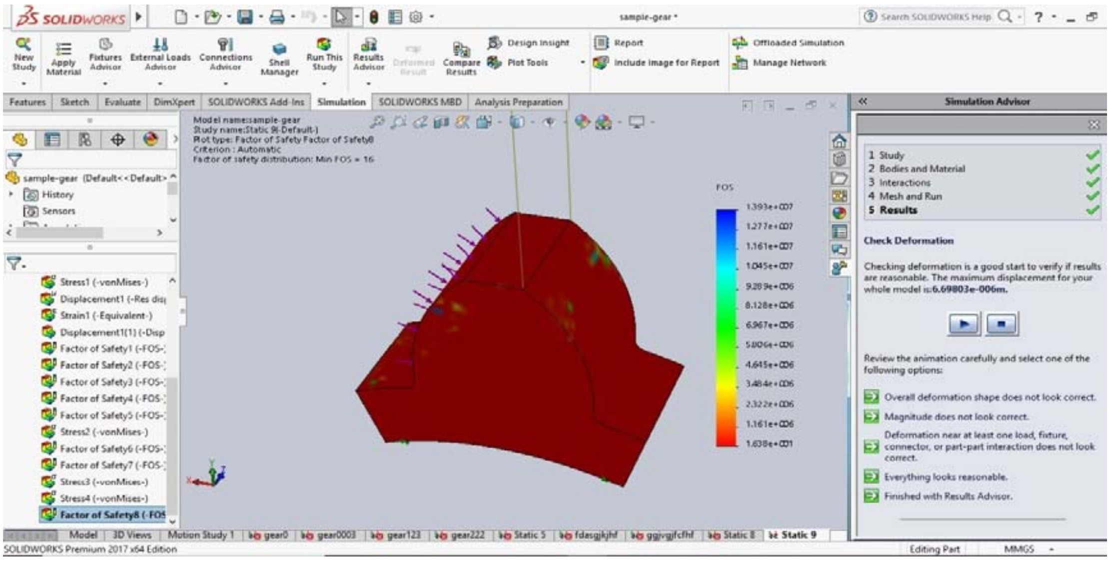

Fig.6.30. Factor of Safety of Gear Tooth for Gray Cast Iron (without hole)

Cast Carbon Steel:

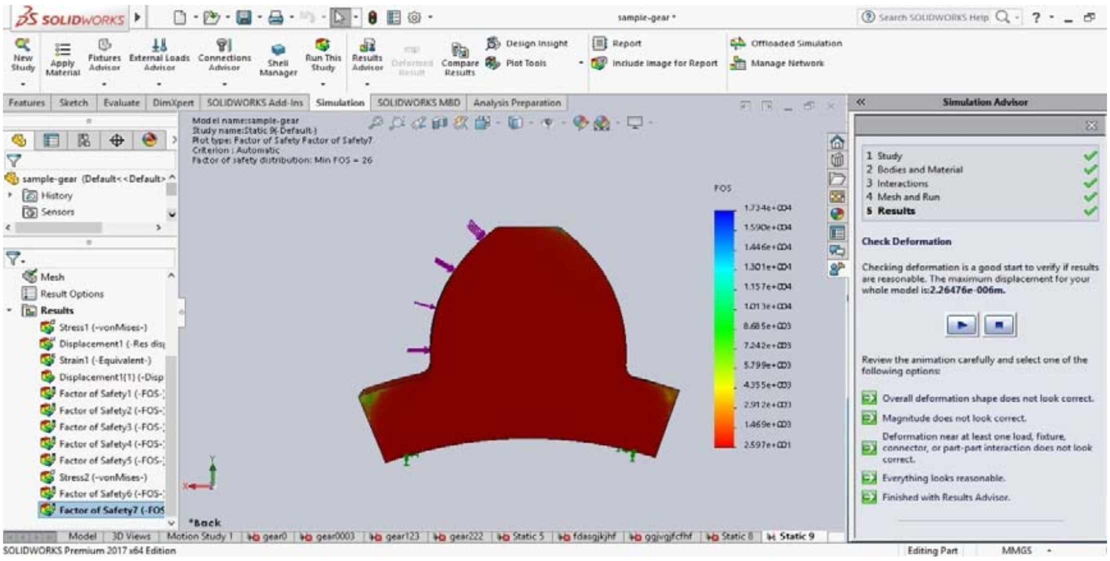

Fig.6.31. Factor of Safety of Gear Tooth for Cast Carbon Steel (without hole)

Brass:

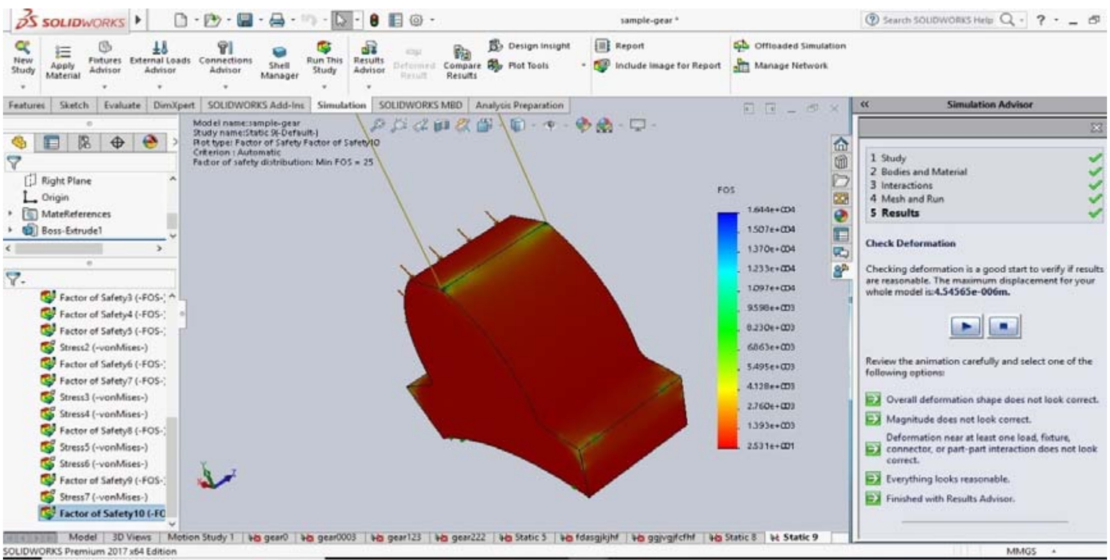

Fig.6.32. Factor of Safety of Gear Tooth for Brass (without hole)

6.5.2. GEAR TOOTH WITH A CIRCULAR HOLE OF DIA. 5MM AT A DISTANCE OF 5MM FROM TOP LAND

Gray Cast Iron:

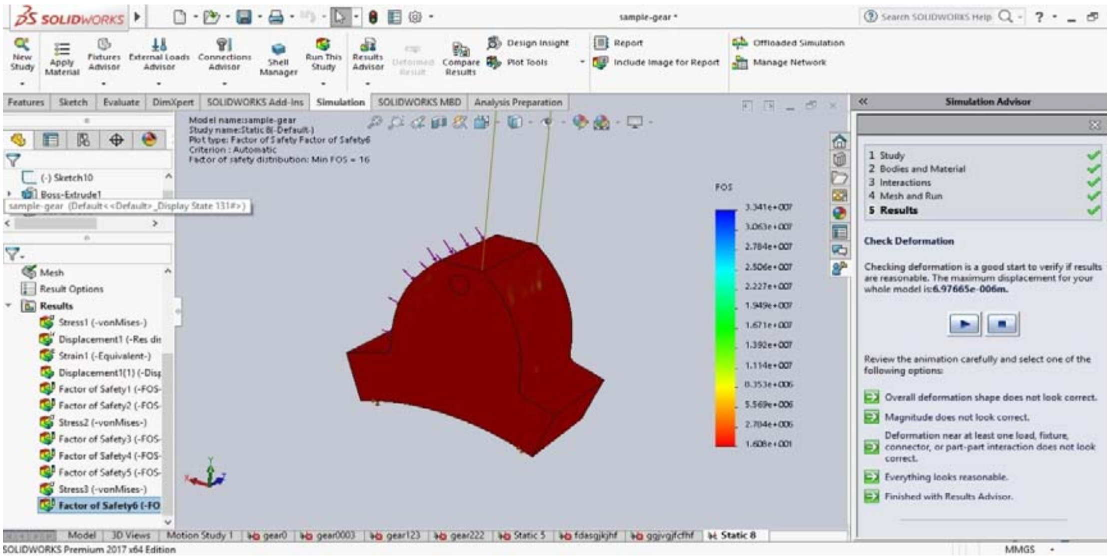

Fig.6.33. Factor of Safety of Gear Tooth for Gray Cast Iron

(hole at 5mm dist.)

Cast Carbon Steel:

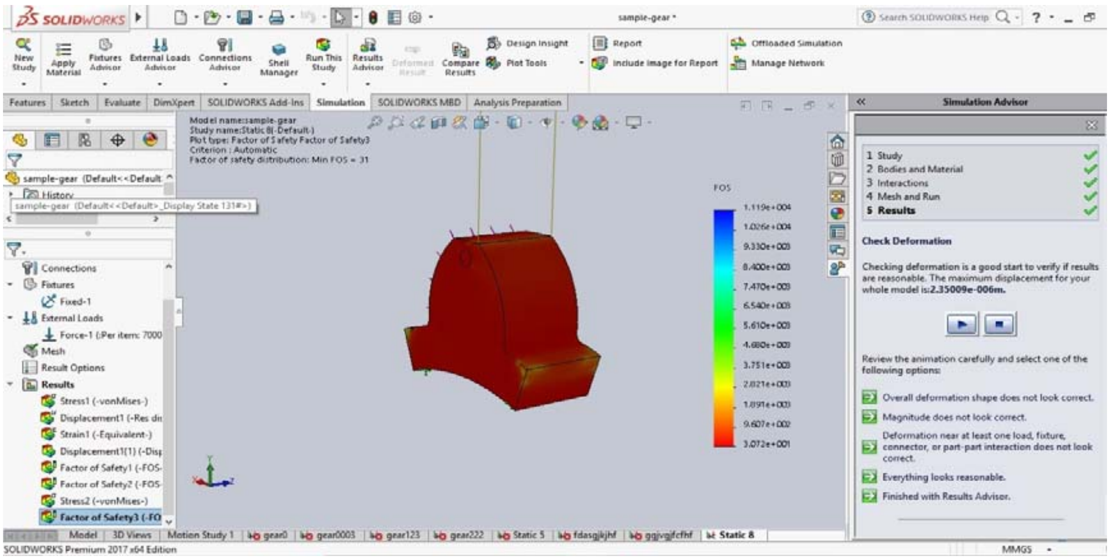

Fig.6.34. Factor of Safety of Gear Tooth for Cast Carbon Steel (hole at 5mm dist.)

Brass:

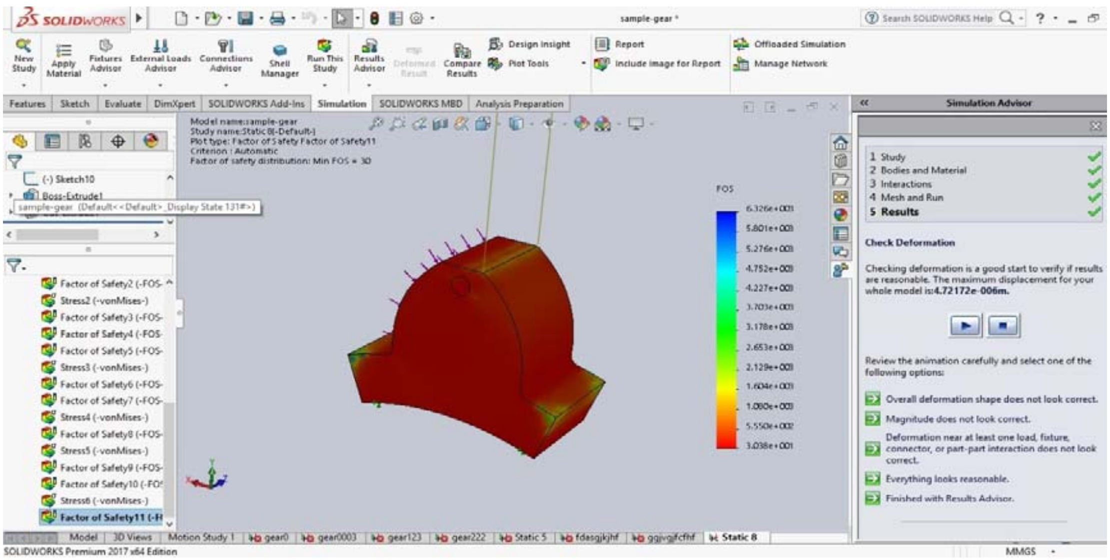

Fig.6.35. Factor of Safety of Gear Tooth for Brass (hole at 5mm dist.)

6.5.3. GEAR TOOTH WITH A CIRCULAR HOLE OF DIA. 5MM AT A DISTANCE OF 10MM FROM TOP LAND

Gray Cast Iron:

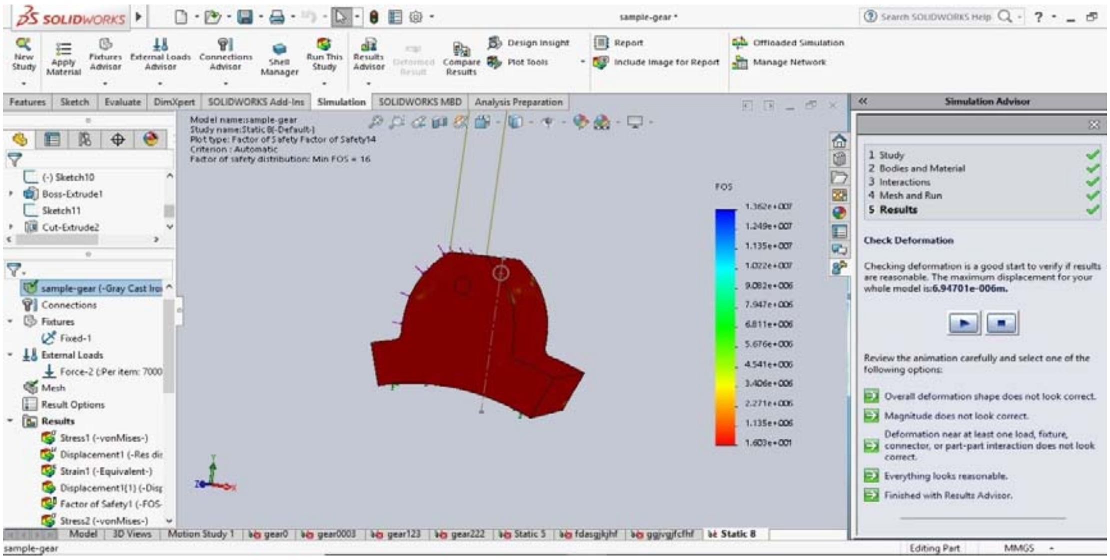

Fig.6.36. Factor of Safety of Gear Tooth for Gray Cast Iron (hole at 10mm dist.)

Cast Carbon Steel:

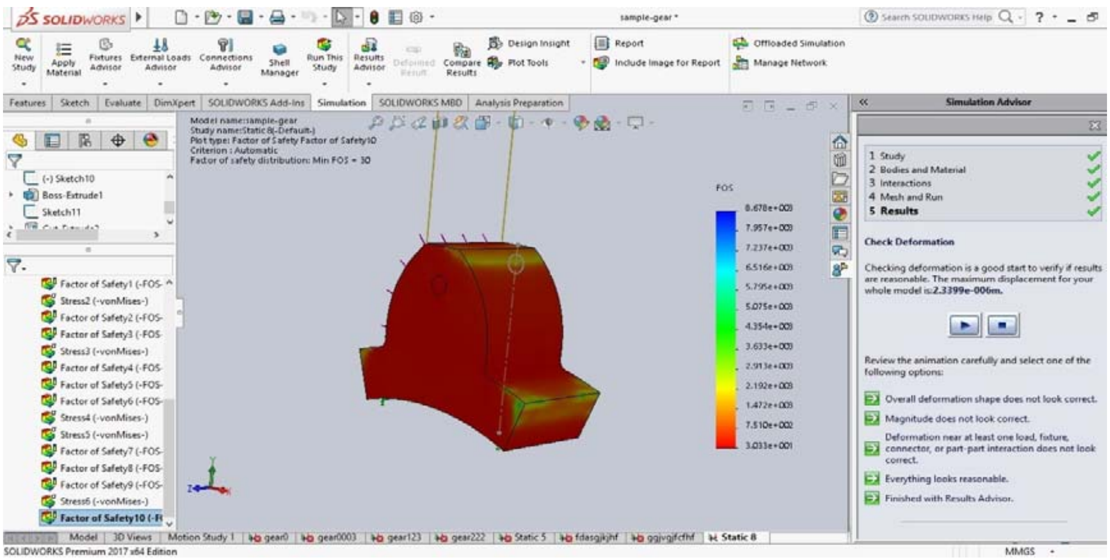

Fig.6.37. Factor of Safety of Gear Tooth for Cast Carbon Steel (hole at 10mm dist.)

Brass:

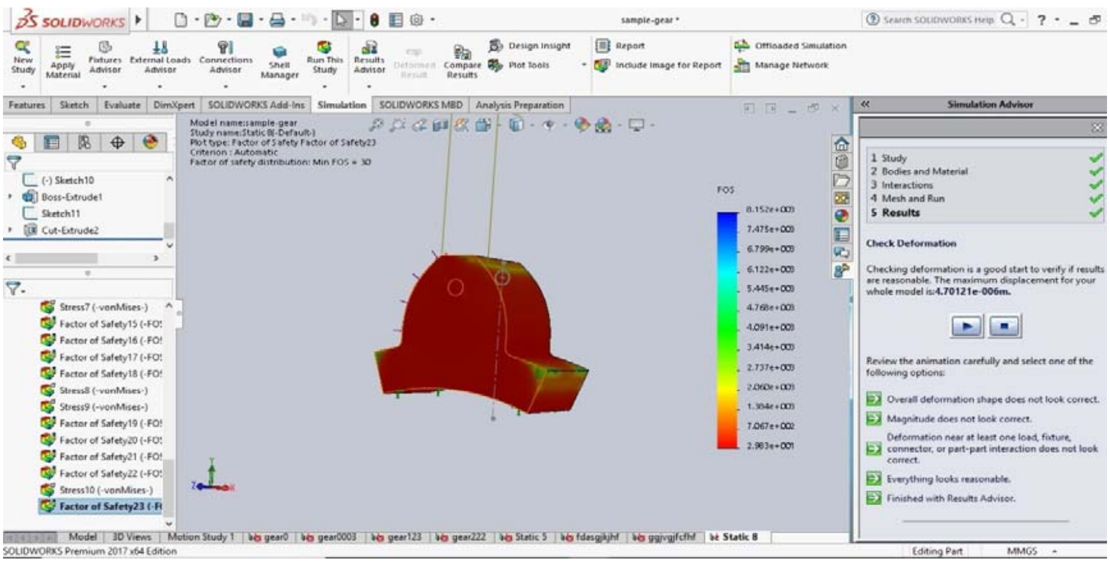

Fig.6.38. Factor of Safety of Gear Tooth for Brass (hole at 10mm dist.)

6.5.4. GEAR TOOTH WITH A CIRCULAR HOLE OF DIA. 5MM AT A DISTANCE OF 15MM FROM TOP LAND

Gray Cast Iron:

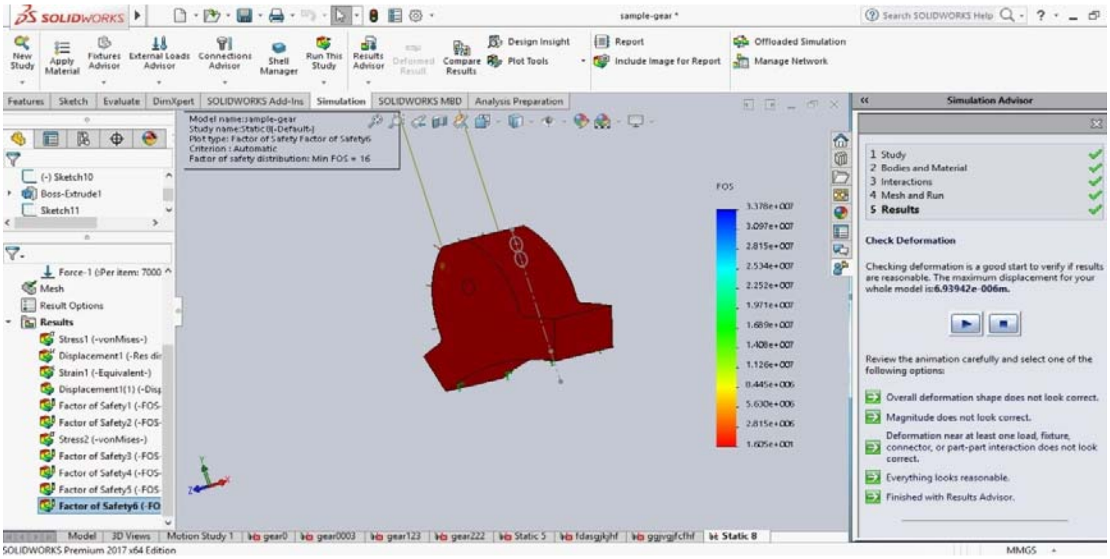

Fig.6.39. Factor of Safety of Gear Tooth for Gray Cast Iron (hole at 15mm dist.)

Cast Carbon Steel:

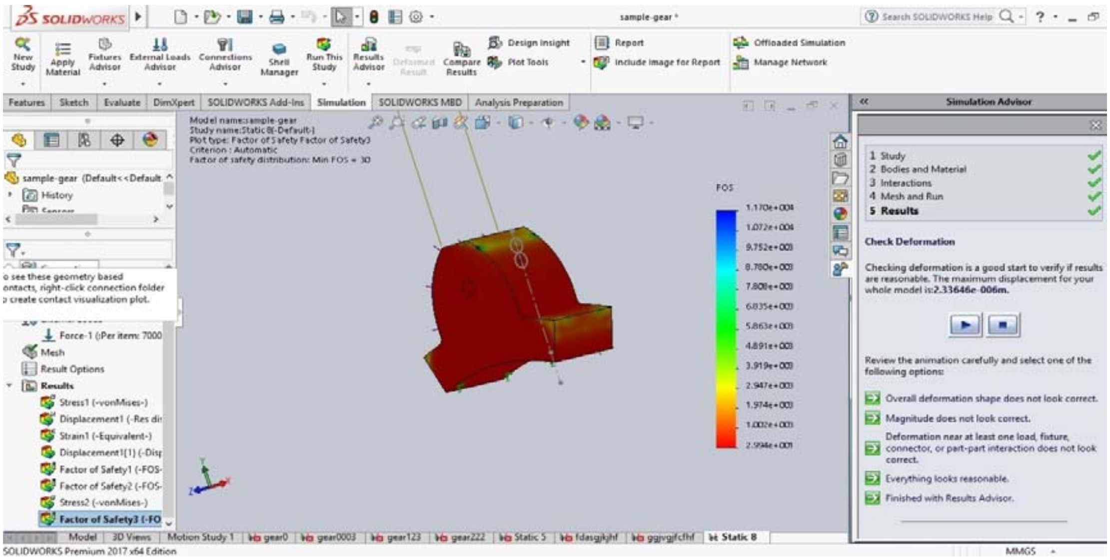

Fig.6.40. Factor of Safety of Gear Tooth for Cast Carbon Steel (hole at 15mm dist.)

Brass:

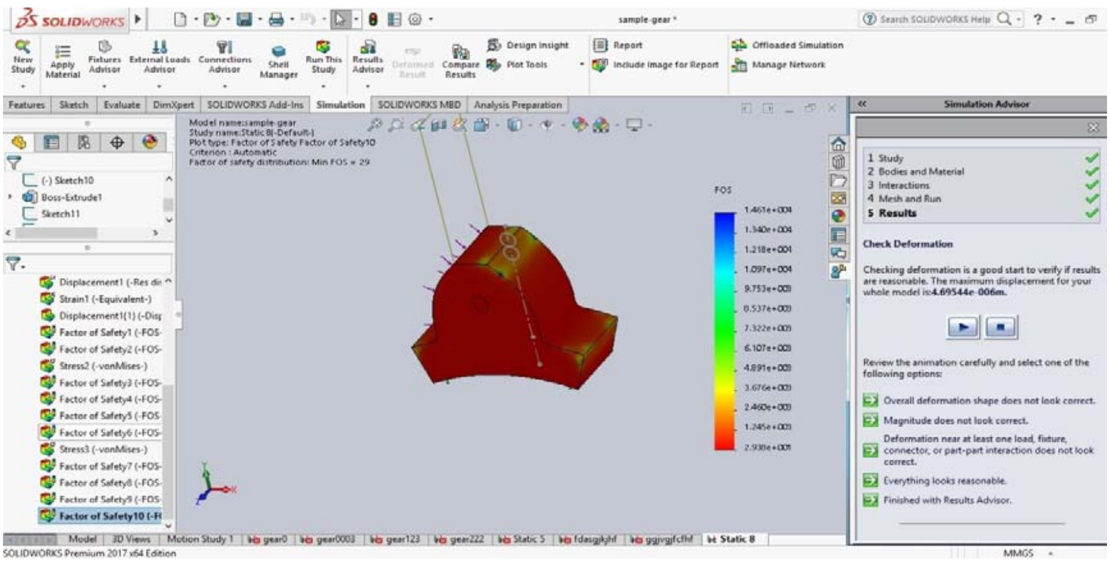

Fig.6.41. Factor of Safety of Gear Tooth Brass (hole at 15mm dist.)

6.5.5. GEAR TOOTH WITH A CIRCULAR HOLE OF DIA. 5MM AT A DISTANCE OF 20MM FROM TOP LAND

Gray Cast Iron:

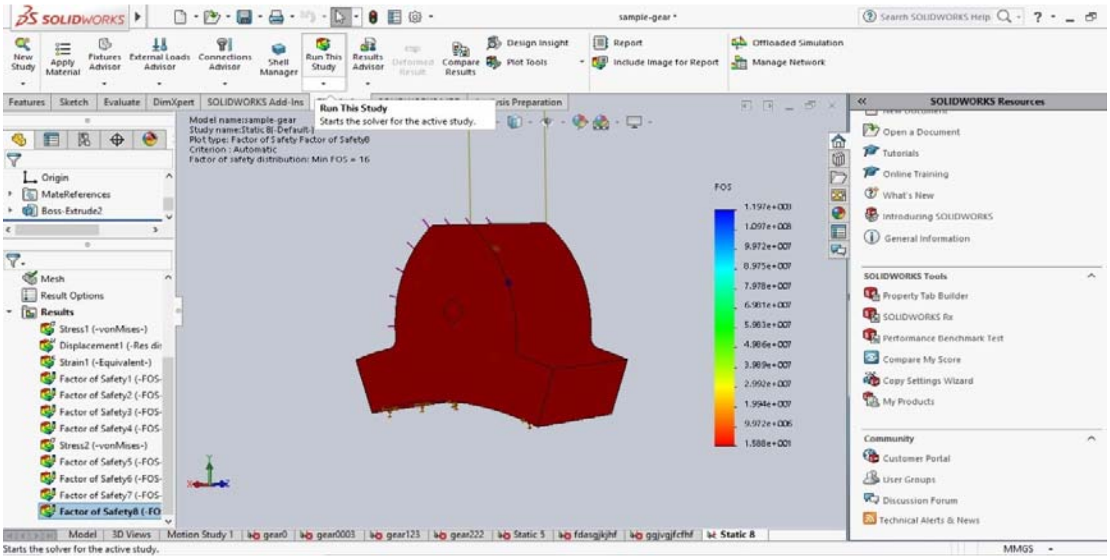

Fig.6.42. Factor of Safety of Gear Tooth for Gray Cast Iron (hole at 20mm dist.)

Cast Carbon Steel:

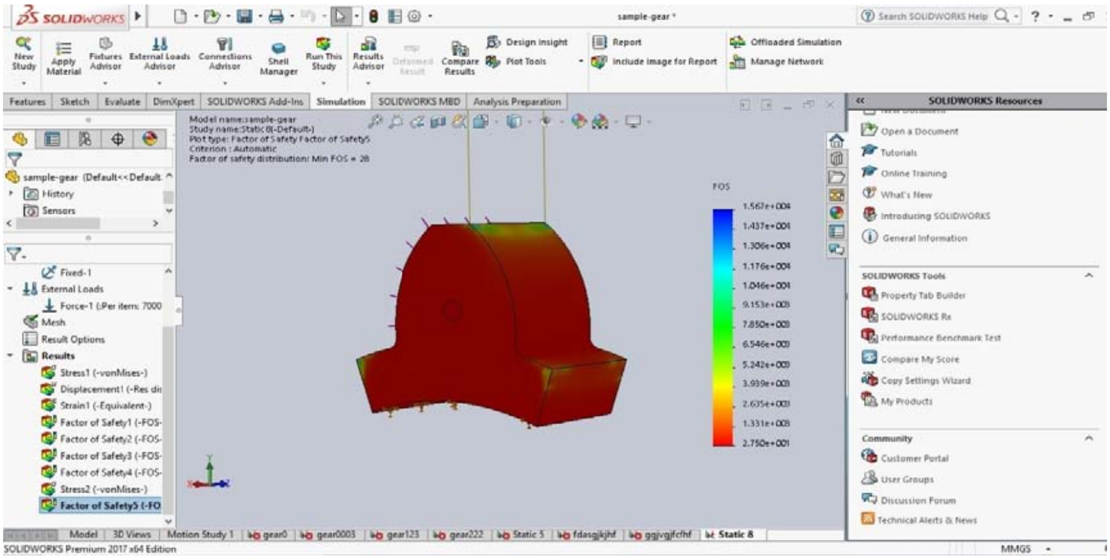

Fig.6.43. Factor of Safety of Gear Tooth for Cast Carbon Steel (hole at 20mm dist.)

Brass:

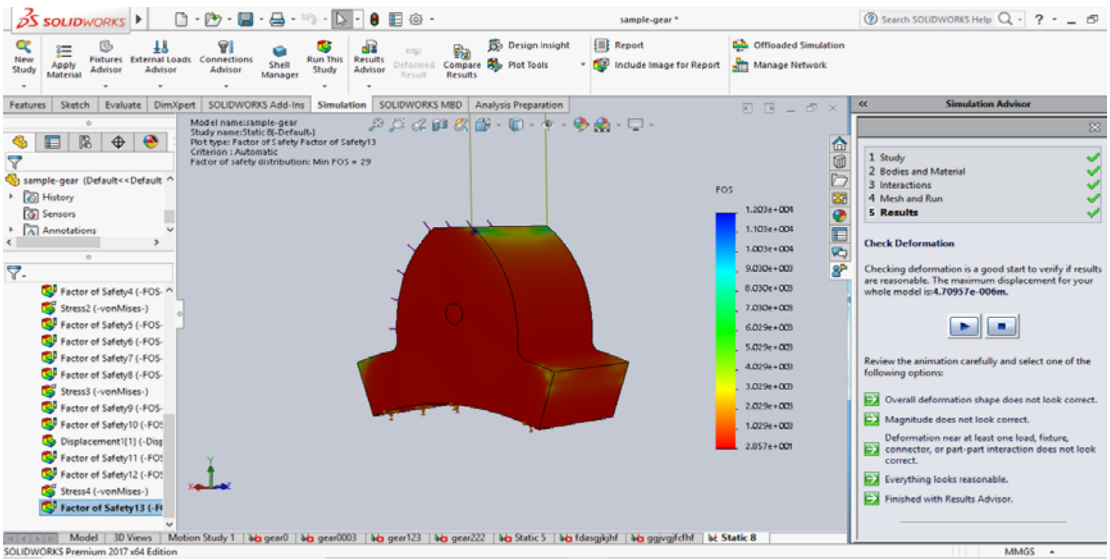

Fig.6.44. Factor of Safety of Gear Tooth for Brass (hole at 20mm dist.)

CHAPTER-7

7.RESULTS & CONCLUSIONS

7.1. VON-MISES STRESS AND FACTOR OF SAFETY VALUES OF GEAR TOOTH WITHOUT HOLE

Table. 3. Experimental Values Of Gear Tooth Without A Hole:

MATERIAL	MAX. VON-MISES STRESS 10^6 N/m^2 (or) MPa	FACTOR OF SAFETY
Gray Cast Iron (GCI)	8.834	16
Cast Carbon Steel (CCS)	9.557	26
Brass	9.469	25

7.2. VON-MISES STRESS AND FACTOR OF SAFETY VALUES OF GEAR TOOTH WITH HOLE AT VARIOUS LOCATIONS

After Introducing Hole in Gear Tooth, We calculate the Von-Mises stress values and Factor of Safety of the Gear Tooth for Different Materials Experimentally.

Hence those values are tabulated below ,

Table. 4. Experimental Values Of Gear Tooth With Hole At A Dist. Of 5mm From Top Land:

MATERIAL	MAX. VON-MISES STRESS 10^6 N/m^2 (or) MPa	FACTOR OF SAFETY
Gray Cast Iron (GCI)	8.194	16
Cast Carbon Steel (CCS)	8.079	31
Brass	7.89	30

Table. 5. Experimental Values Of Gear Tooth With Hole At A Dist. Of 10mm From Top Land:

MATERIAL	MAX. VON-MISES STRESS 10^6 N/m^2 (or) MPa	FACTOR OF SAFETY
Gray Cast Iron (GCI)	8.358	16
Cast Carbon Steel (CCS)	8.182	30
Brass	8.034	30

Table. 6. Experimental Values Of Gear Tooth With Hole At A Dist. Of 15mm From Top Land:

MATERIAL	MAX. VON-MISES STRESS 10^6 N/m^2 (or) MPa	FACTOR OF SAFETY
Gray Cast Iron (GCI)	8.441	16
Cast Carbon Steel (CCS)	8.289	30
Brass	8.133	29

Table. 7. Experimental Values Of Gear Tooth With Hole At A Dist. Of 20mm From Top Land:

MATERIAL	MAX. VON-MISES STRESS 10^6 N/m^2 (or) MPa	FACTOR OF SAFETY
Gray Cast Iron (GCI)	8.593	16
Cast Carbon Steel (CCS)	9.024	28
Brass	8.390	29

7.3. BASED ON EXPERIMENTAL RESULTS:

Table. 8. Comparision of Von-Mises Stress Values of Gear Tooth without hole to With Hole:

Von-Mises Stress Values 10^6 N/m^2 (or) MPa					
MATERIAL	**Gear Tooth Without hole**	**Gear tooth With hole at 5mm from Top land**	**Gear Tooth with hole at 10mm from Top land**	**Gear Tooth with hole at 15mm from Top land**	**Gear Tooth with hole at 20mm from Top land**
Gray Cast Iron	8.834	8.194	8.358	8.441	8.593
Cast Carbon Steel	9.557	8.079	8.182	8.289	9.024
Brass	9.469	7.89	8.034	8.133	8.390

Table. 9. Comparision of Factor of Safety Values of Gear Tooth without hole to With Hole:

Factor of Safety Values					
MATERIAL	**Gear Tooth Without hole**	**Gear tooth With hole at 5mm from Top land**	**Gear Tooth with hole at 10mm from Top land**	**Gear Tooth with hole at 15mm from Top land**	**Gear Tooth with hole at 20mm from Top land**
Gray Cast Iron	16	16	16	16	16
Cast Carbon Steel	26	31	30	30	28
Brass	25	30	30	29	29

7.4. FUTURE SCOPE:

- With the help of above results we can say by introducing holes on the gear tooth the stress is reduced at root fillet than the gear tooth without hole.
- We can also analyse that by introducing holes we can save lot of material in large gears (i.e., Marine Gears) that can be used for other purposes (casting of components).
- We can also analyse that the maximum stress at the fillet is reduced, When the hole is introduced above pitch circle.