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# Study of Stress Relieving Features in Spur Gear

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**Abstract:** *Gear drive is used to transmit power from one shaft to another when distance between two shafts is very small. It maintains the constant velocity ratio without slip. Gears may be classified as spur gear, helical gear, bevel gear, rack and pinion type, worm and worm wheel etc. Spur gears i.e. gears with their teeth parallel to axis of the gear are widely used, as they can be manufactured easily and are capable of withstanding normal loads and are good for low speeds. Gearing is one of the most critical components in a mechanical power transmission system, and in most industrial rotating machinery. It is possible that gears will predominate as the most effective means of transmitting power in future machines due to their high degree of reliability and compactness. In addition, the rapid shift in the industry from heavy industries such as shipbuilding to industries such as automobile manufacture and office automation tools will necessitate a refined application of gear technology.Gears analyses in the past were performed using analytical methods, which required a number of assumptions and simplifications. In general, gear analyses are multidisciplinary, including calculations related to the tooth stresses and to tribological failures such as like wear or scoring. In this thesis, static contact and bending stress analyses were performed, while trying to design spur gears to resist bending failure and pitting of the teeth, as both affect transmission error. As computers have become more and more powerful, people have tended to use numerical approaches to develop theoretical models to predict the effect of whatever is studied. This has improved gear analyses and computer simulations. Numerical methods can potentially provide more accurate solutions since they normally require much less restrictive assumptions. The model and the solution methods, however, must be chosen carefully to ensure that the results are accurate and that the computational time is reasonable.*

**Keywords:** *Stress, Relieving, Features, Spur Gear.*

## 1. Introduction

Gearing is one of the most critical components in a mechanical power transmission system, and in most industrial rotating machinery. It is possible that gears will predominate as the most effective means of transmitting power in future machines due to their high degree of reliability and compactness. In addition, the rapid shift in the industry from heavy industries such as shipbuilding to industries such as automobile manufacture and office automation tools will necessitate a refined application of gear technology.

### 1.1 Failure Modes of Gear Teeth

- *Tooth Breakage – Bending Fatigue*

Bending fatigue failure occurs over a long period of time. The initiation of crack takes place at the weakest point, normally at the root of the tooth or at the fillet where high stress concentration exists together with highest tensile stress from bending or from the surface defects as shown in Fig. 2.4. The crack slowly propagates over 80 to 90% of the life.

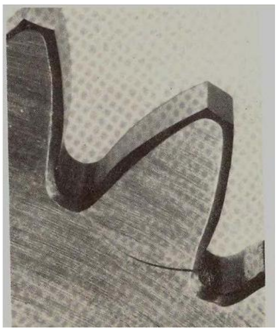

**Figure 2.4** *Root crack*

Then crack propagates fast and suddenly results in fracture of the tooth as shown in Fig. 2.5 The fractured surface will exhibit beach marks in the slow crack propagation region and brittle fracture behaviour in sudden fracture region. Since time taken for the failure is very long, it is known as high cycle fatigue.

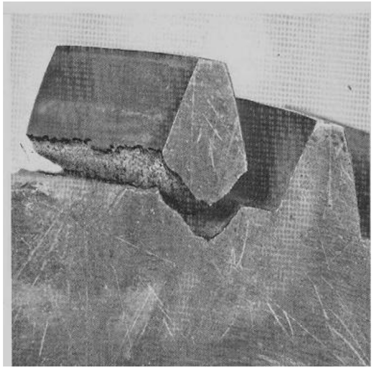

**Figure 2.5** *Tooth breakage*

- *Tooth Breakage – High Cycle Fatigue*

The tooth breakage in case of high cycle fatigue is shown in Fig. 2.6

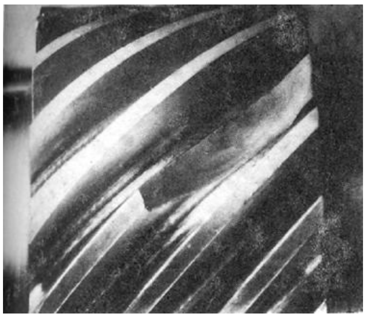

**Figure 2.6** *High cycle fatigue*

- *Tooth Breakage – Low Cycle Fatigue (Over Load)*

Overload breakage or short (low) cycle fatigue causes stringy fibrous appearance in broken ductile material. In harder materials this break has a more silky or crystalline appearance as shown in Fig. 2.7.

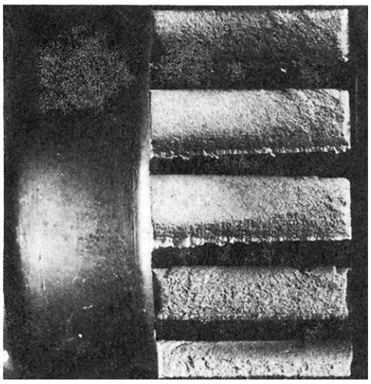

**Figure 2.7** *Low cycle fatigue (over load)*

- *Tooth Breakage – Bending Fatigue*

The Fig. 2.8 shows tooth fatigue by bending fatigue.

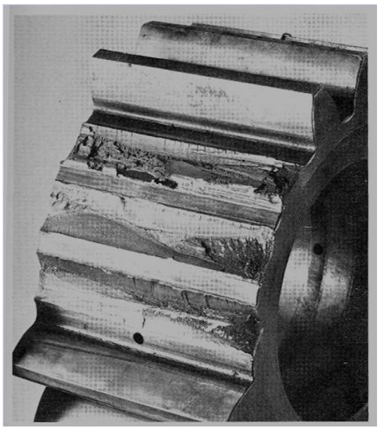

**Figure 2.8** *Bending fatigue*

## 2. Problem Definition

Gearing is one of the most critical components in a mechanical power transmission system, and in most industrial rotating machinery. Due to fatigue failure of gears various stresses generated in gears like bending, compressive, tensile, etc. Aim of this work is to study stress relieving features in spur gear by studying stress relieving technique using FEA.

## 3. Scope of Work

### 3.1 Bending Stresses in Gears

There are several failure mechanisms for spur gears. Bending failure of the teeth is one of the main failure modes. The bending stresses in a spur gear are another interesting problem. When loads are too large, bending failure will occur. Bending failure in gears is predicted by comparing the calculated bending stress to experimentally-determined allowable fatigue values for the given material. This bending stress equation was derived from the Lewis formula. Wilfred Lewis (1892) was the first person to give the formula for bending stress in gear teeth using the bending of a cantilevered beam to simulate stresses acting on a gear tooth. The Lewis equation considers only static loading and does not take the dynamics of meshing teeth into account.

Different factors required for the calculation, can be obtained from the books on machine design. This analysis considers only the component of the tangential force acting on the tooth, and does not consider effects of the radial force, which will cause a compressive stress over the cross section on the root of the tooth. Suppose that the greatest stress occurs when the force is exerted at top of tooth, which is the worst case. When the load is at top of the tooth, usually there are at least two tooth pairs in contact. In fact, the maximum stress at the root of tooth occurs when the contact point moves near the pitch circle because there is only one tooth pair in contact and this teeth pairs carries the entire torque. When the load is moving at the top of the tooth, two teeth pairs share the whole load if the ratio is larger than one and less than two. If one tooth pair was considered to carry the whole load and it acts on the top of the tooth this is adequate for gear bending stress fatigue.

Fatigue or yielding of a gear tooth due to excessive bending stresses is one important gear design considerations. In order to predict fatigue and yielding, the maximum stresses on the tensile and compressive sides of the tooth, respectively, are required. In the past, the bending stress sensitivity of a gear tooth has been calculated using photo elasticity or relatively coarse FEM meshes. However, with present computer developments we can make significant improvements for more accurate FEM simulations.

### 3.2 Stress Relieving Features

The stress features like circular holes are being used for many years to reduce the stresses in the components. These stress relieving features can also be used for reduction of bending stresses in gears. Also now a day, various other features like combination of different types or sizes of holes are being studied. The reduction in the bending stresses in the components will lead to increased factor of safety. Thus study of various types of stress relieving features for the components like gears is important.

## 4. OBJECTIVES OF PROPOSED WORK

The objectives the works are:

- To analyze the bending stress in spur gears using Stress Relieving Technique
- To study the effect of circular hole as a stress relieving feature for gear

## 5. METHODOLOGY FOR WORK

Work will be carried out in the following steps.

- Analytical Approach.
- Modeling of Gears
- Finite Element Analysis

### 5.1 Calculation of Tangential Force

As the tangential force is responsible for torque & hence bending stress, the same is calculated as below.

The forces between Gear No. 1 & 2 are calculated in the following way.

$F_t = \frac{P}{V}$ where,

$F_t$ is tangential force in Newton,

P is power in Watts, &

V is the pitch line velocity in m/s.

P = 5 H.P x 746 = 3730 Watts.

$$V = \frac{\pi . d_1 n_1}{60}$$

$d_1$ = 62.374 mm

$$V = \frac{\pi \text{ x } 0.062374 \times 750}{60}$$

V = 2.449 m/s

$$F_t = \frac{3730}{2.449}$$

$F_t$ = 1523 N

## 6. FEA PART MODELLING & MESHING

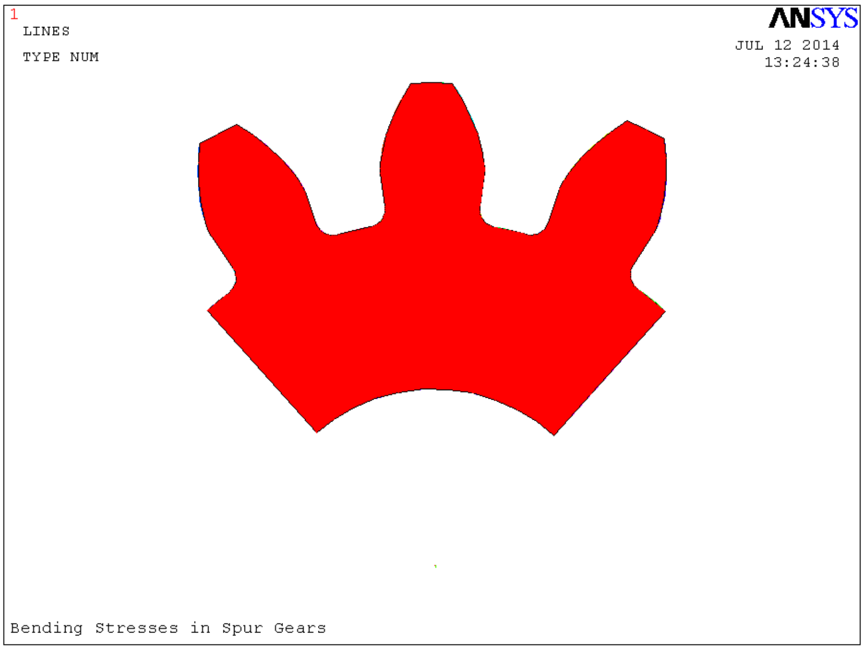


**Fig. 6** *FEA PART MODELLING*

### 6.1 Meshing of Model

For meshing of the model, quadrilateral element with mid-side nodes (PLANE82) is chosen. The geometry of this element is as shown in the figure 4.3 & meshed model is shown in figure.

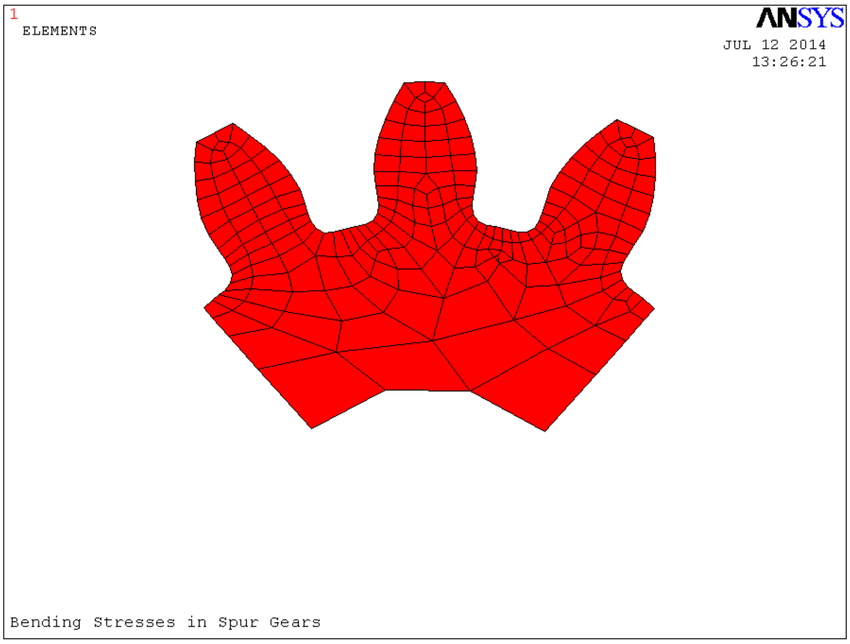


*Figure 6.1 Meshed Model of Pinion*

### 6.2 Boundary Condition

Rotary & linear movements of all points lying on the radial lines & inner circumferential line are restricted. i.e. All degrees of freedom have been locked for these lines. Figure 4.5 shows model with the boundary conditions applied. The small triangles on the surfaces show that the displacement of these surfaces is restricted.

### 6.3 Load Condition

The tangential load acting is equal to 1523 N for 20 mm thick gear. i.e. A load of 1523/20 = 76.15 N/mm is applied to the model at the tip of the tooth. Figure 4.6 shows the model with both boundary conditions & load conditions.

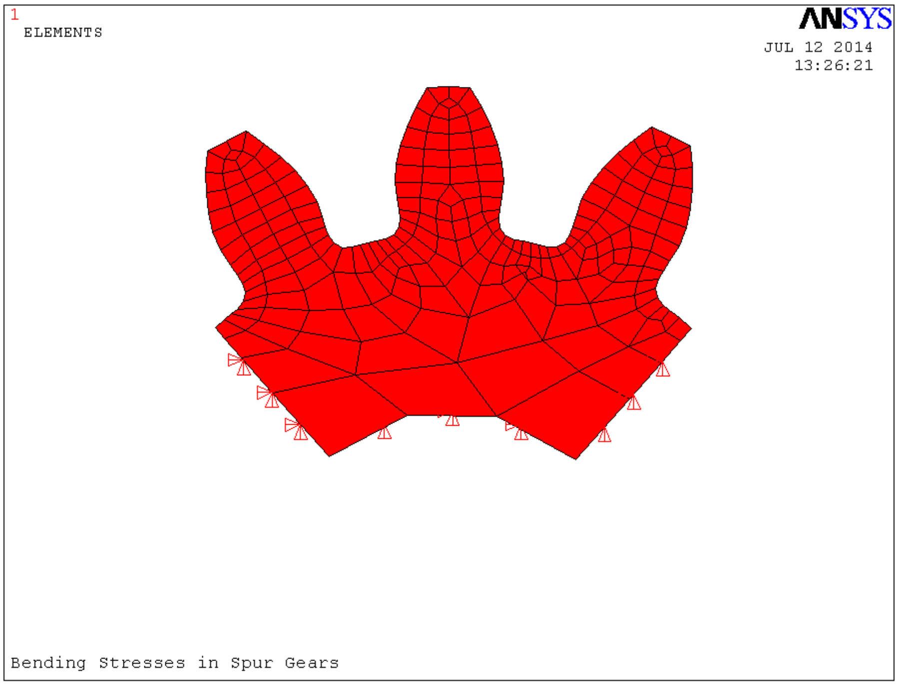


**Fig. 6.3.1** *Boundary Conditions applied to the model*

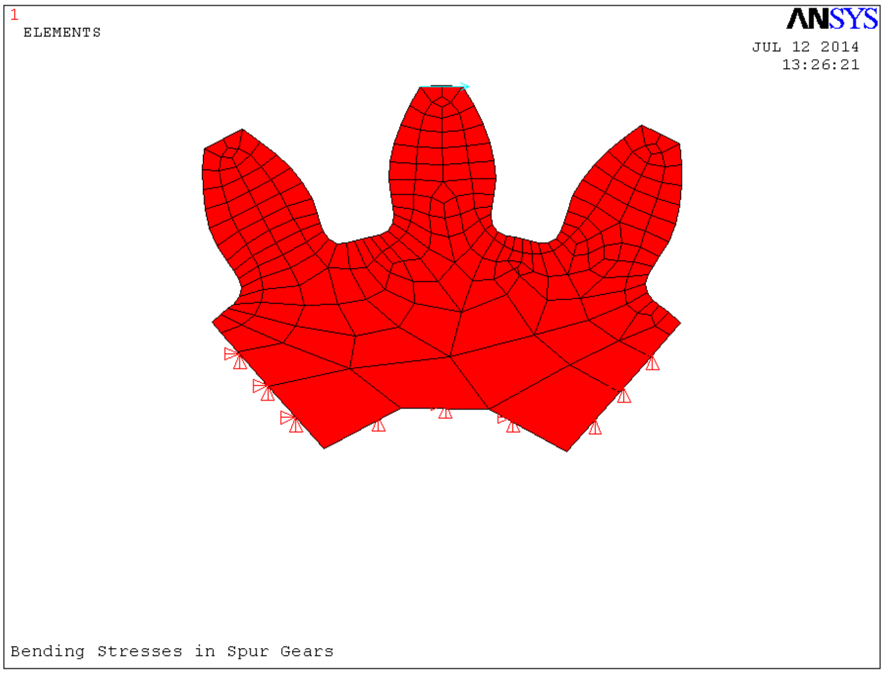


**Fig 6.3.2** *Meshed Model of Pinion along with load condition & boundary condition*

## 7. Results & Discussion

This work deals with ‘Determination of Bending Stress in spur gear & reduction of these stresses using few stress relieving features’. The deformation is also noted during the analysis.

### 7.1 Deformed Shape

The deformed shape is as shown below.

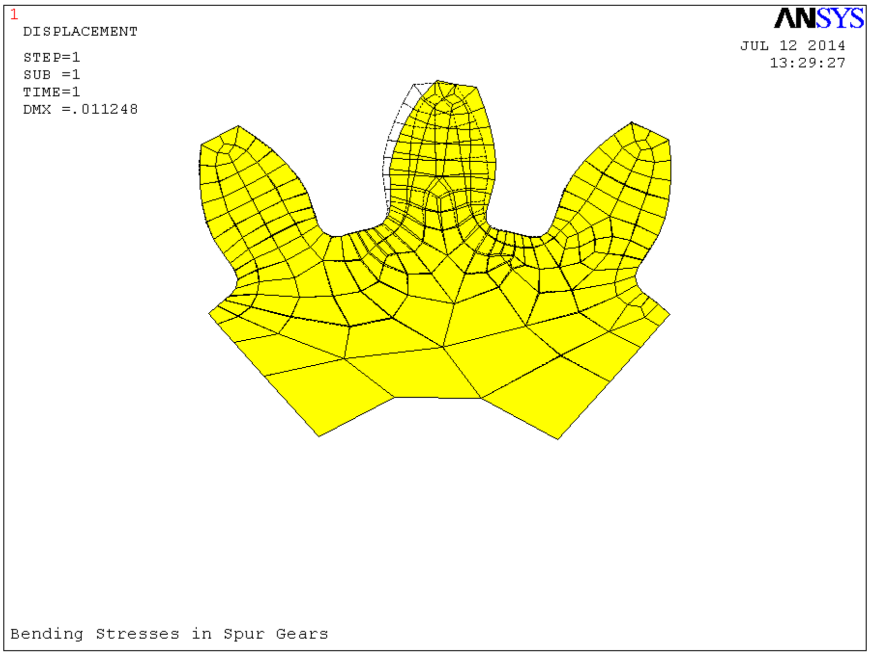


**Figure 7.1** *Deformed Shape & Original Shape*

From the above figure, we can say that, maximum deformation is at the top of the teeth which is far away from the base of the teeth. The value of maximum deformation is 0.011248 mm.

### 7.2 Bending Stresses

Following figure shows the bending stresses. Here as the load is applied at the tip of the tooth, the maximum stress will be at the tip as this load is point load. So neglecting the stresses at the point of application of load, stresses at the root of the teeth are observed.

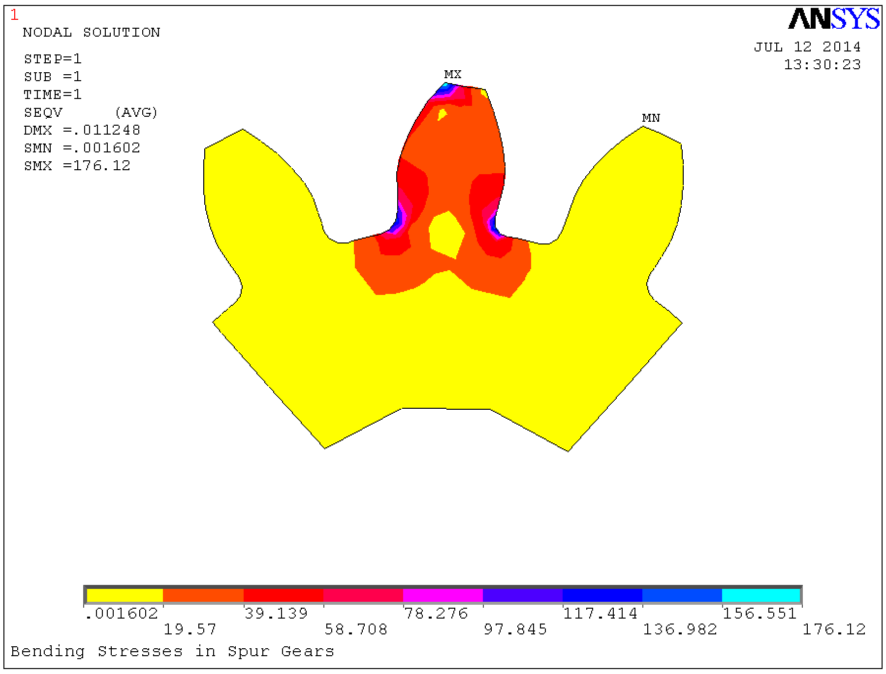


**Figure 7.2.1** *Bending Stresses (Von Mises)*

The following figure shows the exaggerated view of the root of the teeth.

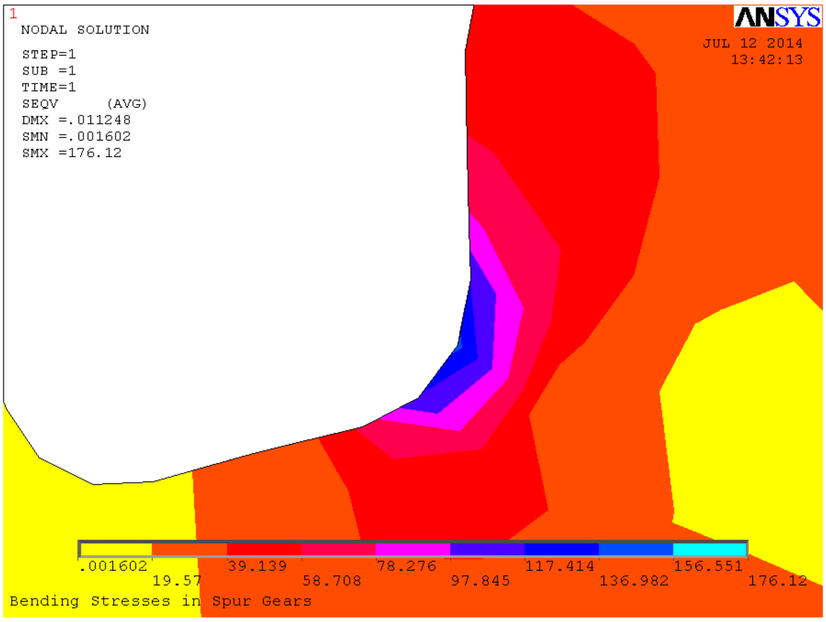


**Figure 7.2.2** *Exaggerated view of Bending Stresses (Von Mises)*

## 7.3 A Single Circular Hole as a Stress Relieving Feature

A single circular hole is used for reducing the stresses. The details are as below.

For this, number of holes are taken & effort is made to reduce the stresses. Also hole diameter is changed during this study & its effect on the stresses is observed. Maximum Von Mises stresses & Maximum deformation is observed throughout this study.

A number of holes are used for stress reduction. The holes lie on two lines which are offset to the teeth profile. The offset distances are 0.75 mm & 1.25 mm. Following figures shows holes made in the gear.

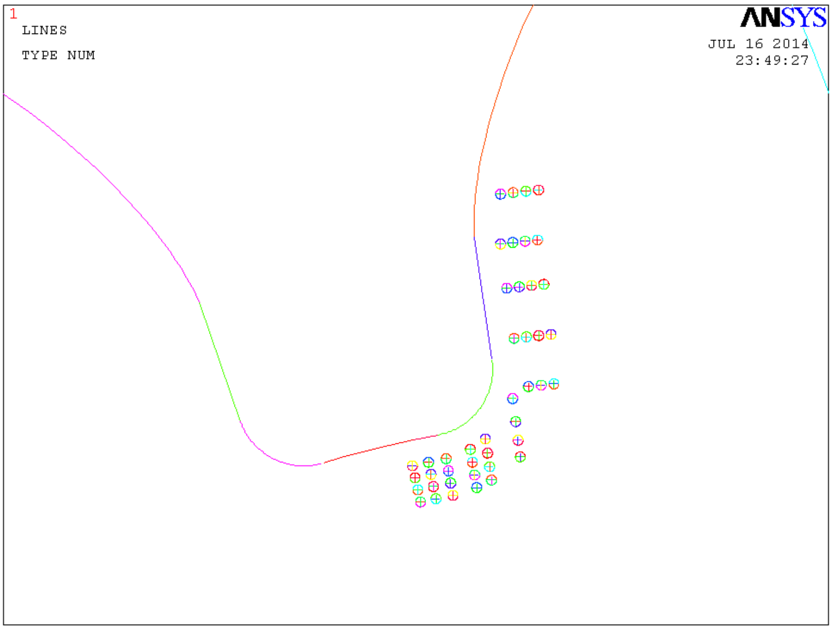


**Figure 7.3** *Holes as stress reducing feature*

## 7.4 Effect of holes of Radius = 0.5 mm & Offset distance = 0.75 mm

Following are the results obtained by using a single circular hole of 0.5 mm radius & the centers of holes lying at a distance of 0.75 mm from the teeth profile.

**Table 7.4.1** *Effect of holes of Radius = 0.5 mm & Offset distance = 0.75 mm*

| Hole Number | Von Mises Stress (Max.) | Deflection (Max.) |
|---|---|---|
| 2 | 176.667 | 0.011518 |
| 6 | 180.531 | 0.011644 |
| 10 | 222.89 | 0.011801 |
| 14 | 319.73 | 0.012119 |
| 17 | 281.932 | 0.012086 |
| 21 | 319.269 | 0.012023 |
| 25 | 209.701 | 0.011877 |
| 29 | 227.553 | 0.011792 |
| 33 | 180.195 | 0.011637 |
| 37 | 183.299 | 0.011538 |
| 41 | 183.783 | 0.011458 |

<u>Effect of radial distance of holes on stresses for offset distance = 0.75 mm & radius of hole = 0.5 mm</u>

Now the values corresponding to the holes lying at same offset distance from the profile but at different radial distances are considered to observe the effect of the radial distance on the values of the stresses.

**Table 7.4.2** *Effect of radial distance of holes on stresses for offset distance = 0.75 mm*

| Radial distance of holes (mm) | Hole Number | Max. Von Mises Stress (MPa) |
|---|---|---|
| 26.187 | 17 | 281.932 |
| 27.187 | 14 | 319.73 |
| 28.187 | 10 | 222.89 |
| 29.187 | 6 | 180.531 |
| 30.187 | 2 | 176.667 |

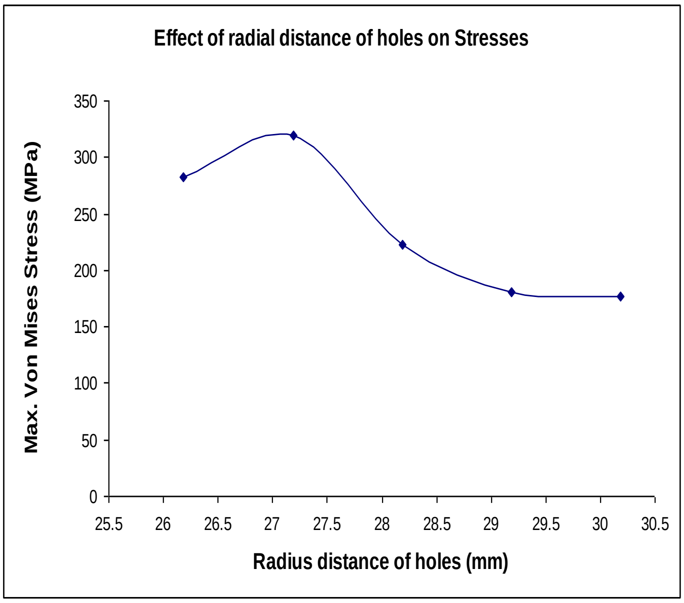


**Figure 7.4.1** *Effect of radial distance of holes on stresses*
(for hole radius = 0.5 mm & offset distance = 0.75 mm)

From the above graph it is seen that initially as the radial distance of hole increases there is increase in the Max. Von Mises stress. But afterwards as the radial distance increases, the stresses decrease & remain almost constant.

<u>Effect of angular position of holes for offset = 0.75 mm & radius of hole = 0.5 mm</u>

Here the data concerned with the holes lying at different angular locations but on same offset line is considered.

**Table 7.4.2** *Effect of angular location of holes on stresses for offset = 0.75 mm*

| Hole Number | Angle of radial centerline of hole with centerline of teeth ($^0$) | Max. Von Mises Stress (MPa) |
|---|---|---|
| 21 | 7.5 | 319.269 |
| 25 | 8.5 | 209.701 |
| 29 | 9.5 | 227.553 |
| 33 | 10.5 | 180.195 |
| 37 | 11.5 | 183.299 |
| 41 | 12.5 | 183.783 |

A graph is plotted with the above data.

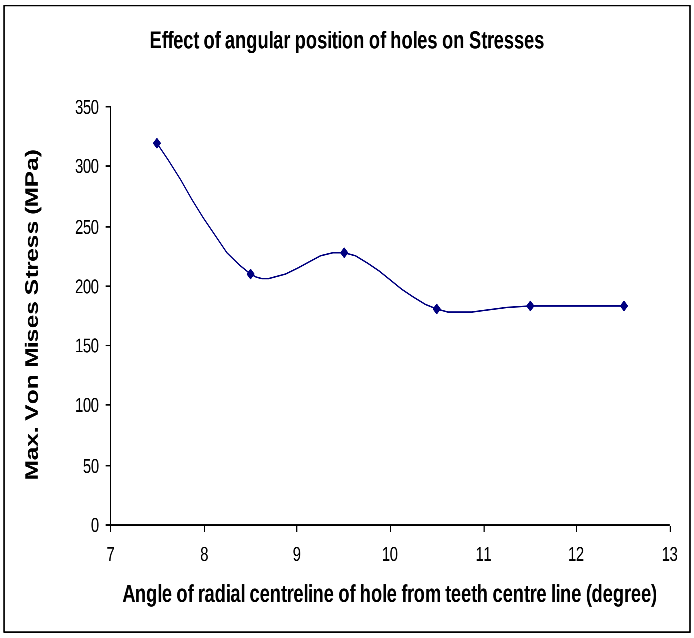


**Figure 7.4.2** *Effect of angular position of holes on stresses for hole radius = 0.5 mm & Offset distance = 0.75 mm*

From the above graph, it is seen that as the angle of radial centerline increases initially the maximum Von Mises stresses decrease initially. Afterwards there is increase in stresses. Again there is decrease in stresses & they almost remain constant.

## 8. Effect of Holes of Radius = 0.5 mm & Offset Distance = 1.25 mm

Following are the results obtained by using a single circular hole of 0.5 mm radius & the centers of holes lying at 1.25 mm from the teeth profile.

**Table 8.1** *Effect of holes of Radius = 0.5 mm & Offset distance = 1.25 mm*

| Hole Number | Von Mises Stress (Max.) | Deflection (Max.) |
|---|---|---|
| 4 | 183.889 | 0.011506 |
| 8 | 186.766 | 0.01159 |
| 12 | 182.633 | 0.011648 |
| 16 | 181.27 | 0.011694 |
| 19 | 183.881 | 0.011621 |
| 23 | 184.558 | 0.011617 |
| 27 | 183.433 | 0.011586 |
| 31 | 177.4 | 0.011437 |
| 35 | 183.968 | 0.011516 |
| 39 | 183.146 | 0.011465 |
| 43 | 183.073 | 0.011425 |

Effect of radial distance of holes on stresses for offset distance = 1.25 mm & radius of hole = 0.5 mm

Here also the holes lying at same offset but different radial distance are considered as above.

**Table 8.2** *Effect of radial distance of holes on stresses for offset distance = 1.25 mm*

| Radial distance of holes (mm) | Hole Number | Max. Von Mises Stress (MPa) |
|---|---|---|
| 26.187 | 4 | 183.889 |
| 27.187 | 8 | 186.766 |
| 28.187 | 12 | 182.633 |
| 29.187 | 16 | 181.27 |
| 30.187 | 19 | 183.881 |

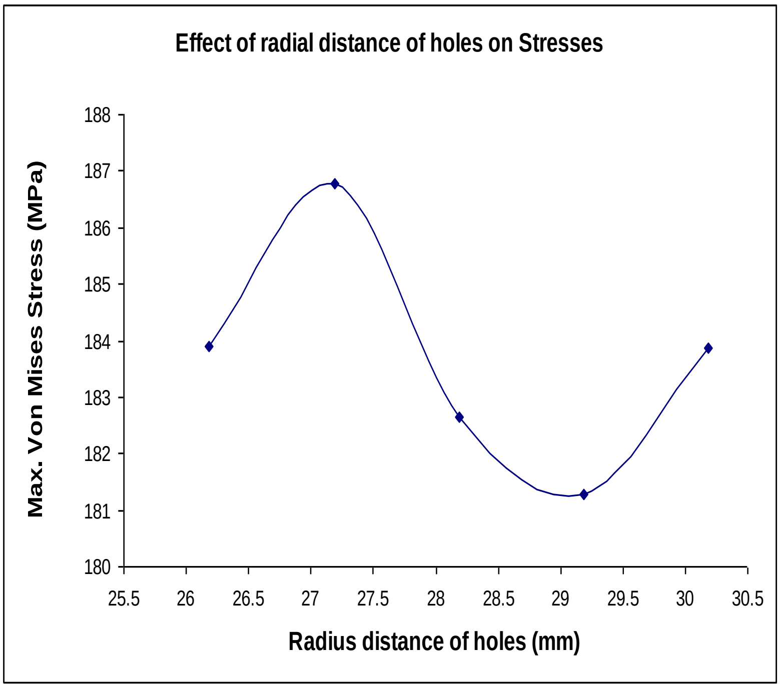


**Figure 8.2** *Effect of radial distance of holes on stresses for hole radius = 0.5 mm & offset distance = 1.25 mm*

From the above graph it is seen that as the radial distance of holes goes on increasing initially, there is increase in the Von Mises stresses but afterwards it decreases & again increases as radial distance increases.

<u>Effect of angular position of holes for offset = 1.25 mm & radius of hole = 0.5 mm</u>

Effect of angular position of hole is studied as in the earlier case. The results are as below.

**Table 8.3** *Effect of angular position of holes for offset = 1.25 mm*

| Hole Number | Angle of radial centerline of hole with centerline of teeth ($^0$) | Max. Von Mises Stress (MPa) |
|---|---|---|
| 23 | 7.5 | 184.558 |
| 27 | 8.5 | 183.433 |
| 31 | 9.5 | 177.4 |
| 35 | 10.5 | 183.968 |
| 39 | 11.5 | 183.146 |
| 43 | 12.5 | 183.073 |

A graph shown ahead is representing the above data.

From the below graph it is seen that as the angle of radial centerline of hole increases, there is reduction in the stresses up to a value of $9.5^0$ but afterwards stresses again start increasing & they remain constant at the end.

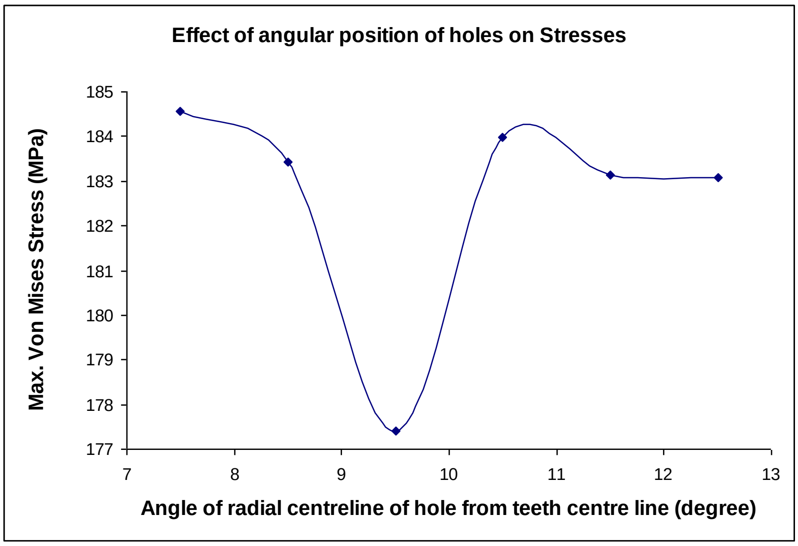


**Figure 8.3** *Effect of angular position of holes on stresses for hole radius = 0.5 mm & Offset distance = 0.75 mm*

## 9. EFFECT OF HOLE RADIUS OF 1MM FOR HOLES LYING AT 1.25 MM OF OFFSET

As the stresses obtained for the two offset distances 0.75 mm & 1.25 mm & hole radius 0.5 mm are higher than those obtained in the model without any stress relieving feature, few holes are tried with radius of holes as 1 mm. As the values of Stresses & Deformation are very much higher for many cases of holes lying at offset distance 0.75 mm, the holes are selected are having offset distance 1.25 mm.

**Table 9.1** *Effect of holes of 1 mm radius & offset distance 1.25 mm.*

| Hole Number | Von Mises Stress (Max.) | Deflection (Max.) |
|---|---|---|
| 31 | 297.678 | 0.012935 |
| 35 | 217.386 | 0.012463 |
| 39 | 231.384 | 0.012341 |
| 43 | 179.364 | 0.012118 |

## 10. EFFECT OF HOLE CHOSEN ARBITRARILY

As the stress & deformation are not reducing, few more holes are tried which are chosen arbitrarily. The coordinates of centre of hole are given with reference to the centre of the pinion which lies at a distance of 75 mm in X direction & 25 mm in Y direction from origin. Following are the results obtained during the above study.

**Table 10.1** *Effect of holes chosen arbitrarily*

| Sr. No. | X Coordinate (in mm) | Y Coordinate (in mm) | Radius of hole (in mm) | Max. Von Mises Stress (in MPa) | Max. Deflection (in mm) |
|---|---|---|---|---|---|
| 1 | 74.05 | 51.2 | 0.8 | 175.951 | 0.011389 |
| 2 | 74.05 | 51.2 | 1 | 184.058 | 0.011489 |
| 3 | 74.05 | 51.2 | 0.7 | 183.681 | 0.011361 |
| 4 | 75 | 51 | 0.8 | 184.507 | 0.011317 |
| 5 | 73 | 51 | 0.8 | 184.581 | 0.011770 |
| 6 | 74 | 52 | 0.8 | 184.548 | 0.011528 |
| 7 | 74 | 50 | 0.8 | 181.541 | 0.011333 |
| 8 | 77 | 51 | 0.8 | 227.879 | 0.012255 |
| 9 | 74.15 | 51.2 | 0.8 | 180.175 | 0.011377 |
| 10 | 73.95 | 51.2 | 0.8 | 178.052 | 0.011401 |
| 11 | 73.85 | 51.2 | 0.8 | 179.843 | 0.011431 |
| 12 | 74 | 51.2 | 0.8 | 177.9 | 0.011404 |
| 13 | 74.05 | 51 | 0.8 | 181.903 | 0.011363 |

During this study, it is seen that there is only slight reduction in maximum stress for a hole lying at a X = 74.05 mm & Y = 51.2 mm & having a radius = 0.8 mm.

The reduce value of Maximum Von Mises stress is 175.951 MPa.

% Reduction in stress = (176.12 – 175.951) *100 / 176.12 = 0.09595

Figure 5.4 shows the Von Mises stresses in the above case.

As this reduction is very less, more than one circular hole as a stress reducing feature is tried. For all other values of X & Y coordinate of centre of hole, the stresses are higher than those present in the model without any stress relieving feature.

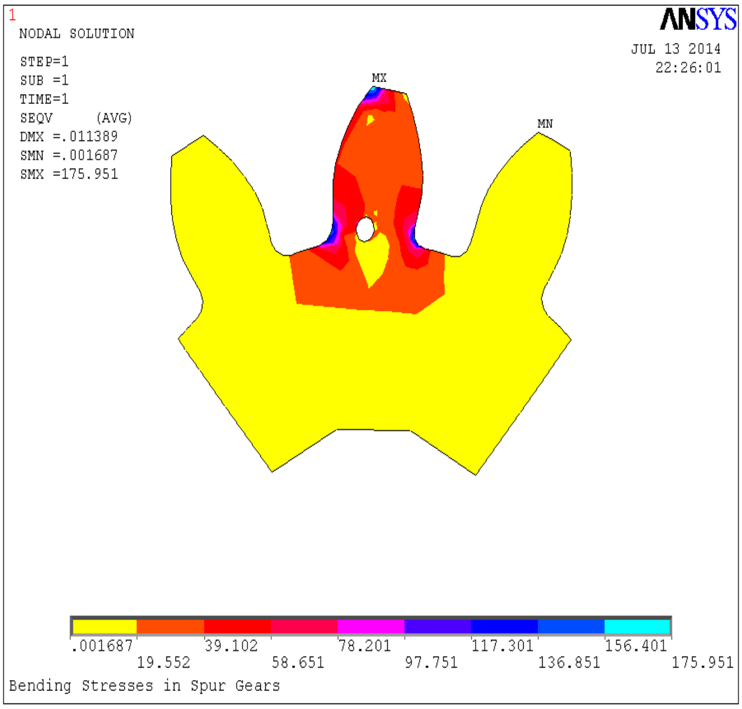


**Figure 10.1** *Von Mises stresses for Single Circular hole as a stress reducing feature*

## 11. CONCLUSION

In this paper stress relieving features for spur gear have been studied with single hole stress relieving technique by using FEA.

Results of this study provides how to minimize stresses developed in spur gear.

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