

CONTACT STRESS ANALYSIS OF PLASTIC GEARS DUE TO MISALIGNMENTS OF SHAFTS USING FEM

A Project report submitted in partial fulfilment of the requirements for the Award of the
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**BACHELOR OF ENGINEERING
in
MECHANICAL ENGINEERING**

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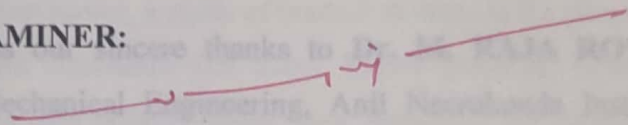
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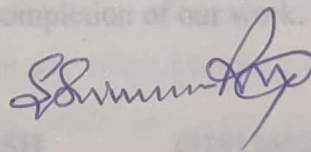


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ABSTRACT

Contact stresses analysis of gear drives has been a subject of intensive research since last few years. Contact stresses were first analyzed by Hertz and he developed analytical correlations for determining them. Contact stresses are strongly dependent on the types of gears, friction at the point of contact, velocities of rotation, loads, etc. It also depends on the accuracy of mounting, bearings, shaft deflection and workmanship of gears.

In the current endeavor, a study of contact stresses in the plasticspur gears under different misalignment conditions was taken. Plastic gears are widely utilized in the healthcare industry as they possess higher wear resistance and produce less noise. For quite a long time, the designers have begun to explore the benefits of using special compounds in Gears used for medical applications.

The proposed work is intended to derive the contact stresses under the combined effect of pitch and radial misalignments. These misalignments could be due to wear in the bearings or improper workmanship in the fabrication of gears and shafts or ineffective assembly of the gear systems or any other external causes. This work is accomplished using two types of composites, which are acetol co-polymer reinforced with 1.5% graphene and cast nylon. The gears are assumed to be fabricated with these composites. The stresses induced in the gears for these two different composites were analyzed and compared to identify the best material.

The contact stress analysis is conducted with the help of FEM software ANYSYS. Hertz contact stresses were also evaluated for ideal case using theoretical correlations, when there are no misalignments in the shafts, and the values were compared with the analytical results for validation of the model. The induced stresses exhibited an increasing trend with an increase of the combined effect of pitch and radial misalignments. Minimum values were observed when the two shafts are parallel and maintained at required central distance. Graphene reinforced Acetol co-copolymer was observed as the best material out of the selected materials. It was also observed that the stresses induced in this material was 27% to 30% less than that induced in cast nylon.

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NOMENCLATURE

A	Transverse pressure angle, deg
a	Addendum, mm
b	length of contact, mm
β	Helix angle, deg
C_P	Elastic coefficient Factor
D	Pitch diameter of pinion and gear, mm
E	Young's Modulus, GPa
F_t	Tangential Force on gear teeth, N
F_n	Normal force on gear teeth, N
i	Speed ratio
I	Geometry factor
K_V	Dynamic factor
K_O	Overload factor
K_M	Load distribution factor
K_S	Size factor
n_1	No. of teeth on pinion
n_2	No. of teeth on gear
σ_c	Contact stresses, MPa
R	Radial distance from Centre to pitch circle, mm
r_1	Pitch radius of pinion, mm
r_2	Pitch radius of gear, mm
μ	Coefficient of Friction
ν	Poisson's ratio
V	velocity of rotation, m/s
Z_R	surface condition factor

CHAPTER-1

INTRODUCTION

Gears are one of the oldest of humanity's inventions. Nearly all the devices we think of as machines utilize gearing of one type or another. Gear technology has been developed and expanded throughout the centuries. In many cases, gear design is considered as a specialty. Nevertheless, the design or specification of a gear is only part of the overall system design picture. From industry's standpoint, gear transmission systems are considered one of the critical aspects of vibration analysis.

The understanding of the behavior when gears are in mesh is extremely important if one wants to perform system monitoring and control of the gear transmission system. Although there is large amount of research studies about various topics of gear transmission, the basic understanding of gears in mesh still needs to be confirmed. When a pair of gears mesh, localized Hertzian contact stress are produced along with tooth bending and shearing. This is a non-linear problem, and it can be solved by applying different types of contact elements and algorithms in finite element codes.

However, due to the complicated contact conditions, acquiring results in the meshing cycle can be challenging since some solutions may not converge. In any case, using quadrilateral elements seem to be useful in solving gear contact problems with finite element analysis. Furthermore, meshing stiffness is often being discussed when a pair of gears are in mesh. Meshing stiffness can be separated into Torsional Mesh Stiffness and Linear Tooth Mesh Stiffness.

Gears are a critical component in the rotating machinery industry. Various research methods, such as theoretical, numerical, and experimental, have been done throughout the years regarding gears. One of the reasons why theoretical and numerical methods are preferred is because experimental testing can be particularly expensive. Thus, numerous mathematical models of gears have been developed for different purposes.

This chapter presents a brief review of papers recently published in the areas of gear design, transmission errors, vibration analysis, etc., also including brief information about the models, approximations, and assumptions made at the contact point. However, if there are more teeth in contact, the uncracked teeth would share the load, which unloads the cracked tooth and thus reduces the stiffness disturbance effect.

The main purpose of gearing is to transmit motion from one shaft to another. If there is any mistake or error on the gears, motion will not be transmitted correctly. Also, if the errors on the gears are crucial, it may destroy or heavily damage the components in a gearbox. Therefore, it becomes important to understand the subject of gearing. In order to gain better understanding of gearing, one should get some knowledge about the design of gear and the theory of gear tooth action.

1.1 Types of gears:

There are many different types of gears used by industry, but all these gears share the same purpose, which is to transmit motion from one shaft to another. Generally, gearing consists of a pair of gears with axes are either parallel or perpendicular. Among all the gears in the world, the four most commonly discussed gears are spur gear, helical gear, bevel gear, and worm gearing.

1.1.1 Spur gear:

It is considered as the simplest form of gearing, and they consist of teeth parallel to the axis of rotation. The common pressure angles used for spur gears are $14\frac{1}{2}$, 20, to and 25 degrees. One of the advantages of a low-pressure angle is smoother and quieter tooth action. In contrast, larger pressure angles have the advantages of better load carrying capacity.

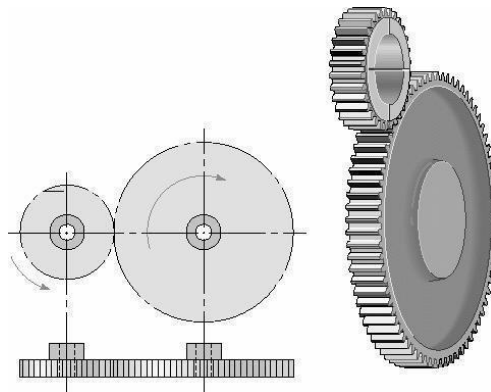


Fig.1.1: Spur Gear

1.1.2 Helical gears: It consists of teeth that are cut at an angle and inclined with the axis of rotation. Helical gears essentially have the same applications as spur gears. However, because of their gradual engagement of the teeth during meshing, helical gears tend to be less noisy. In addition, the inclined tooth develops thrust loads and bending couples, which are not present in the spur gear.

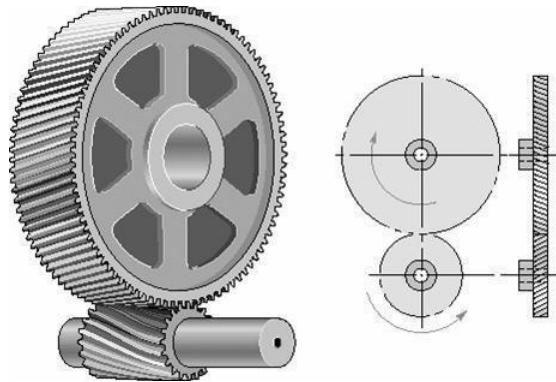


Fig.1.2: Helical Gear

1.1.3 Bevel gears: Its teeth are formed on conical surfaces and unlike spur and helical gears, bevel gears are used for transmitting motion between intersecting shafts not parallel shafts. There are different types of bevel gears, but all of them establish thrust, radial, and tangential loads on their support bearings.



Fig.1.3: Bevel Gear

1.1.4 Worm gearing: It consists of the worm and worm gear. Depend upon the rotation direction of the worm; the direction of rotation of the worm gear would be different. The direction of rotation also depends upon whether the worm teeth are cut left-hand or right-hand. In general, worm gear sets are more efficient when the speed ratios of the two shafts are high. Basically, in worm gearing, higher speed equals to better efficiency



Fig.1.4: Worm Gear

1.2 spur gear: Spur gears are simple, easily manufactured gears and are usually the first choice when exploring gear options. Transmitting power between parallel axes, the teeth project radially on the disc. Spur gears are regularly used for speed reduction or increase, torque multiplication, resolution and accuracy enhancement for positioning systems. The teeth run parallel to the gear axis and can only transfer motion between parallel-axis gear sets. Spur gears mate only one tooth at a time, resulting in high stress on the mating teeth and noisy operation.



Fig.1.5: Spur Gear

1.2.1 Gear Nomenclature:

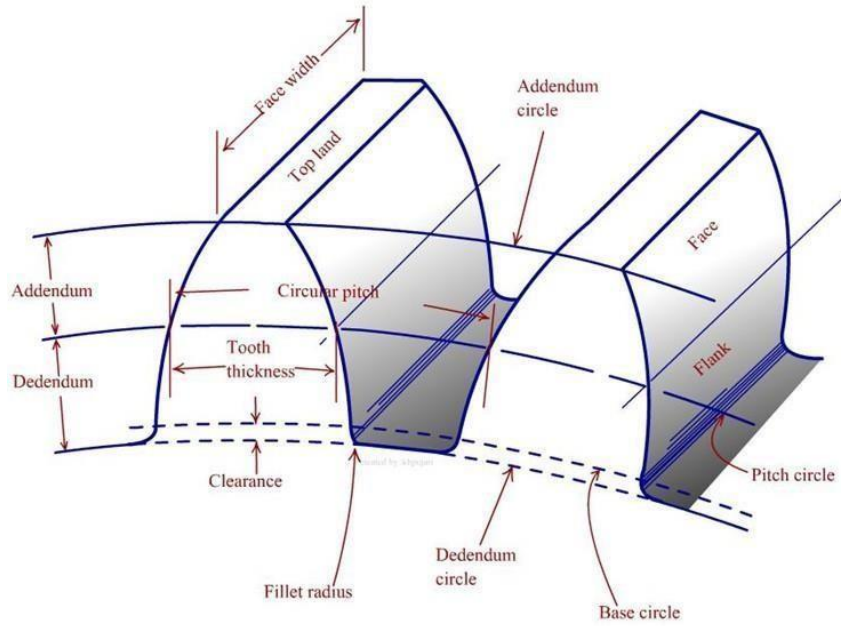


Fig.1.6. Gear Nomenclature

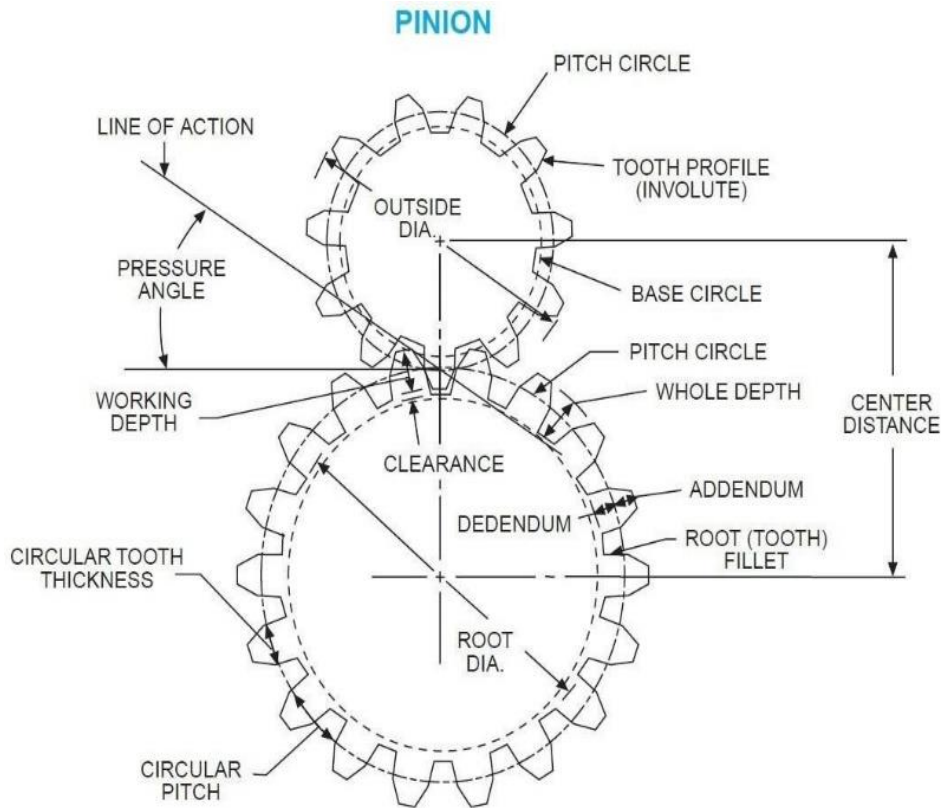


Fig.1.7: Gear Mating Nomenclature

In a pair of gears, the larger is often called the GEAR and, the smaller one is called the PINION. Helical gears can be used in a variety of applications since they can be mounted on either parallel or on 90° non-intersecting shafts. The close concentricity between the pitch diameter and outside diameter allow for smooth and quiet operation.

- **Pitch Surface:** The pitch surfaces of the gears are imaginary planes, cylinders or cones that roll together without slipping.
- **Pitch circle:** It is a theoretical circle upon which all calculations are usually based. It is an imaginary circle that rolls without slipping with the pitch circle of a mating gear. Further, pitch circles of a mating gear are tangent to each other.
- **Pitch circle diameter:** The pitch circle diameter is the diameter of pitch normally, the size of the gear is usually specified by pitch circle diameter. This is denoted by “d”
- **Top land:** The top land is the surface of the top of the gear tooth
- **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.
- **Addendum:** The Addendum is the radial distance between the pitch and addendum circles. Addendum indicates the height of tooth above the pitch circle
- **Dedendum:** The dedendum is the radial distance between pitch and the dedendum circles. Dedendum indicates the depth of the tooth below the pitch circle.
- **Whole Depth:** The whole depth is the total depth of the tooth space that is the sum of addendum and Dedendum.
- **Working depth:** The working depth is the depth of engagement of two gear teeth that is the sum of their addendums
- **Clearance:** The clearance is the amount by which the Dedendum of a given gear exceeds the addendum of its mating tooth.
- **Face:** The surface of the gear tooth between the pitch cylinder and the addendum cylinder is called face of the tooth.
- **Flank:** The surface of the gear tooth between the pitch cylinder and the root cylinder is called flank of the tooth.
- **Face Width:** It is the width of the tooth measured parallel to the axis.

- **Fillet radius:** The radius that connects the root circle to the profile of the tooth is called fillet radius.
- **Circular pitch:** is the distance measured on the pitch circle, from a point on one tooth to a corresponding point on an adjacent tooth.
- **Circular tooth thickness:** The length of the arc on pitch circle subtending a single gear tooth is called circular tooth thickness. Theoretically circular tooth thickness is half of circular pitch.
- **Width of space:** (tooth space) the width of the space between two adjacent teeth measured along the pitch circle. Theoretically, tooth space is equal to circular tooth thickness or half of circular pitch
- **Working depth:** The working depth is the depth of engagement of two gear teeth, that is the sum of their addendums.
- **Whole depth:** The whole depth is the total depth of the tooth space, that is the sum of addendum and dedendum and (this is also equal to whole depth + clearance)
- **Centre distance:** It is the distance between centres of pitch circles of mating gears. (it is also equal to the distance between centre's of base circles of mating gears)
- **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 teeth must be on this line to give smooth operation. The force is transmitted from the driving gear to the driven gear on this line.
- **Pressure angle:** It is the angle that the line of action makes with the common tangent to the pitch circles.
- **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.
- **Arc of approach:** It is the arc of the pitch circle through which a tooth moves from its beginning of contact until the point of contact arrives at the pitch point.
- **Arc of recess:** It is the arc of the pitch circle through which a tooth moves from the contact at the pitch point until the contact ends.
- **Velocity ratio:** If the ratio of angular velocity of the driving gear to the angular velocity of driven gear. It is also called the speed ratio.

- **Module:** It is the ratio of pitch circle diameter in millimeters to the number of teeth. It is usually denoted by 'm'.
- **Back lash:** It is the difference between the tooth space and the tooth thickness as measured on the pitch circle.
- **Velocity Ratio:** Is the ratio of angular velocity of the driving gear to the angular velocity of driven gear. It is also called the speed ratio

1.3 .Modes of Gear failures:

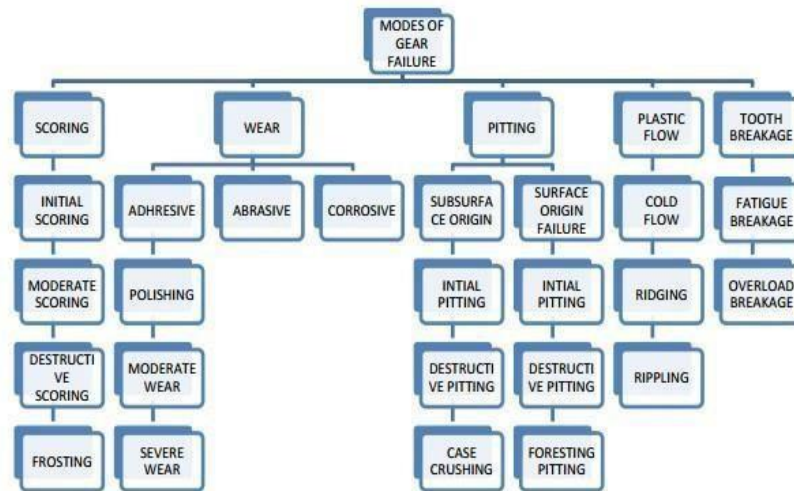


Fig.1.8: Modes of Failure

Gear failure can occur in various modes. If care is taken during the design stage itself to prevent each of these failures a sound gear design can be evolved. The gear failure is explained by means of flow diagram in Fig. 1.8.

1.3.1. SCORING:

Scoring is due to combination of two distinct activities: First, lubrication failure in the contact region and second, establishment of metal to metal contact. Later on, welding and tearing action resulting from metallic contact removes the metal rapidly and continuously so far the load, speed and oil temperature remain at the same level. The scoring is classified into initial, moderate and destructive



Fig.1.9: Scoring

1.3.2 WEAR:

As per gear engineer's point of view, the wear is a kind of tooth damage where in layers of metal are removed more or less uniformly from the surface. It is nothing but progressive removal of metal from the surface. Consequently tooth thins down and gets weakened. Three most common causes of gear tooth wear are metal-to-metal contact due to lack of oil film, ingress of abrasive particles in the oil and chemical wear due to the composition of oil and its additives. Wear is classified as adhesive, abrasive and chemical wear.



Fig.1.10: Wear

1.3.3 PITTING OF GEARS:

Pitting is a surface fatigue failure of the gear tooth. It occurs due to repeated loading of tooth surface and the contact stress exceeding the surface fatigue strength of the material. Material in the fatigue region gets removed and a pit is formed. The pit itself will cause stress concentration and soon the pitting spreads to adjacent region till the whole surface is covered. Subsequently, higher impact load resulting from pitting may cause fracture of already weakened tooth. However, the failure process takes place over millions of cycles of running. There are two types of pitting, initial and progressive.



Fig.1.11: Macro pitting on Gear Teeth

1.3.4 PLASTIC FLOW:

COLD FLOW Plastic flow of tooth surface results when it is subjected to high contact stress under rolling cum sliding action. Surface deformation takes place due to yielding of surface or subsurface material. Normally it occurs in softer gear materials. But it can occur even in heavily loaded case hardened gears. Cold flow material over the tooth tip can be seen clearly in the bevel gear shown in Fig.1.12.



Fig.1.12: Damage to a case-hardened gear resulting from overheating associated with insufficient lubrication

1.3.5. TOOTH FRACTURE:

Tooth fracture is the most dangerous kind of gear failure and leads to disablement of the drive and frequently to damage of other components (shafts, bearings, etc.) by pieces of the broken teeth. Tooth breakage may be the result of high overloads of either impact or static in nature, repeated overloads causing low- cycle fatigue, or multiple repeated loads leading to high cycle fatigue of the material.

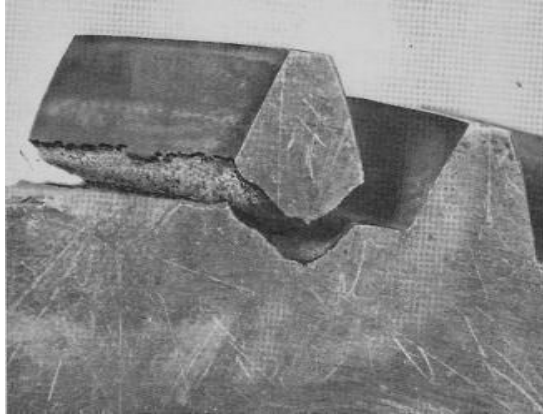


Fig.1.13: Tooth Breakage

1.3.6. GEAR NOISE:

The gear noise arises due to several reasons. At the contact point due to error in the gear profile, surface roughness, impact of tooth and sliding and rolling friction; bearings, churning of the lubricant, and windage. The principal methods of combating noise are: improving the tooth finishing operations, changing over to helical gearing, modifying the profile by flanking, increasing the contact ratio, equalizing the load along the face width of the tooth rim, using crowned gears, and improving the design of the covers and housings.

CHAPTER-2

LITERATURE REVIEW

- 1. S PhaniKuamr¹, KNS Suman², S. Ramanjaneyulu³ et.al:** The current work manages to assess the contact stresses in a spur gear pair of given input parameters with a deviation of gear shaft axis varying the center-to-center distance (axial misalignment) and varying the radial misalignment. For the period of running of the spur gears, there is a chance of wear and tear within the bearing houses and these bearings are supporting the shafts which might results in misaligning of shafts in different directions. In this manner, it is important to examine the combination of various misalignment effects on the running of plastic gears. This idea has been satisfied to some extent through this examination, and the following preliminary conclusions were made from the investigation.
- 2. Putti Srinivasa Rao and et.al:** This paper aims at the minimization of both contact stress as well as deformation to arrive at the best possible combination of driver and driven gear. In this process of spur gears mating, 3 different materials were selected and the software program was performed for 9 different combinations to get the best result possible. The results of the two-dimensional FEM analysis from ANSYS are presented. These stresses were compared with the theoretical Hertz's equation values. Both results agree very well. This indicates that the FEM model is accurate.
- 3. Seok-ChulHwang, Jin-Hwan Lee , Dong-Hyung Lee ,Kwon-Hee Lee [2011] :**This paper presents a contact stress analysis for a pair of mating gears during rotation. Contact stress analyses for spur and helical gears are performed between two gear teeth at different contact positions during rotation. Two examples of spur and helical gears are presented to investigate the respective variations of the contact stress in a pair of mating gears with the contact position. The variation of the contact stress during rotation is compared with the contact stress at the lowest point of single-tooth contact (LPSTC) and the AGMA (American Gear Manufacturers Association) equation for the contact stress. In this study, we can see that the gear design that considers the contact stress in a pair of mating gears is more severe than that of the AGMA standard.

4. **M.R.Lias, M.Awang ,M.N.Talib, A.R.Senawi , M.A.Samad** : In this paper, a finite-element method was used to investigate the effects of axial misalignments to strength properties of the spur gear. This model was constructed based on gear CAD model generated from an Autodesk Inventor gear design generator using quasi statics approach. The investigation leads to a significant finding that axial misalignment results showed that axial misalignments do impact upon strength the strength properties of the gears.
5. **S. Ramanjaneyulu, S.Phani Kumar , V.Suresh Babu** : In this paper , initially graphene filler material has been prepared through modified Hummers method on a bulk scale. After that composites with varying weight percentages of grapheme i.e 0.5,1, 1.5, 2% in Acetal copolymer has been fabricated through injection molding process. The prepared composites have been tested for their mechanical and tribological properties based on the obtained results 1.5% filler composition has been found to be optimum when compared with other various combinations of the composition. So 1.5% graphene filled Acetal copolymer composites were used to design spur gears and analysis has been carried out through FEM. These analyses concludes that 1.5% graphene reinforced Acetal copolymer gears were found to be much more superior in case of performance, weight and cost compared to the both metallic and other polymer gears .

CHATER-3

CONTACT STRESS

3.1 Introduction:

Contact stress is generally the deciding factor for the determination of the requisite dimensions of gears. Research on gear action has confirmed fact that beside contact pressure, sliding velocity, viscosity of lubricant as well as other factors such as frictional forces, contact stresses also influence the formation of pits on the tooth surface. So thorough study of contact stress developed between the different mating gears are mostly important for the gear design. Gearing is one of the most critical components in mechanical power transmission systems.

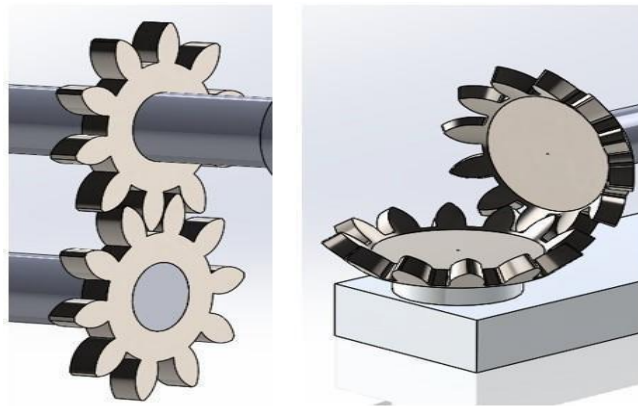


Fig. 3.1: Gear mating

Current Analytical methods of calculating gear contact stresses use Hertz's equations, which were originally derived for contact between two cylinders. So for CONTACT STRESSES it's necessary to develop and to determine appropriate models of contact elements, and to calculate contact stresses using ANSYS and compare the results with Hertzian theory.

3.2 Contact Mechanics:

Study of deformation of solids under contact is called contact mechanics,

comprising of mechanics of material and continuum mechanics. Contact mechanics provides the information for safe and energy efficient design of mechanical elements in contact, while continuum mechanics provides for analysis of the kinematics and the mechanical behavior of materials modeled as a continuous mass rather than as discrete particles.

Contact between two continuous, non-conforming solids is initially a point or line. Under the action of a load the solids deform and a contact area is formed as shown in Figure 3.2. Hertz contact stress theory allows for the prediction of the resulting contact area, contact pressure, compression of the bodies, and the induced stress in the bodies. In 1880 Heinrich Hertz developed his theory for contact stress after studying Newton's rings with two glass lenses. He became concerned about the effect of contact pressure between the two lenses and set out to analyze the effects. The result was the first satisfactory theory for contact mechanics and is still in use today.

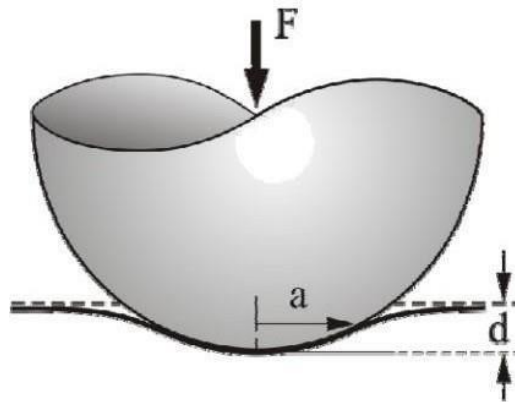


Fig.3.2: Depiction of contact area under applied load

In the course of developing his theory Hertz made some simplifying assumptions which are summarized as follows:

- a) Surfaces are continuous and non-conforming.
- b) Strains are small.
- c) Solids are elastic.
- d) Surfaces are frictionless.

3.3 Hertz contact stress (Involute Gear Tooth Contact Stress Analysis):

One of the main gear tooth failure is pitting which is a surface fatigue failure due to many repetition of high contact stresses occurring in the gear tooth surface while a pair of teeth is transmitting power.

Contact failure in gears is currently predicted by comparing the calculated Hertz contact stress to experimentally determined allowable values for the given material.

The method of calculating gear contact stress by Hertz's equation originally derived for contact between two cylinders.

$$\sigma_c = c_p \left[\frac{F_t}{dDl} \right]^{\left(\frac{1}{2}\right)}$$
$$C_p = \left[\frac{1}{\pi \left(\frac{1 - \nu_1^2}{E_1} + \frac{1 - \nu_2^2}{E_2} \right)} \right]^{\left(\frac{1}{2}\right)}$$

$$F_t = F_n X \cos \theta$$

$$I = \frac{\cos \alpha \sin \alpha}{2} \frac{m_g + 1}{m_g}$$

The AGMA Contact stress equation is given by,

$$\sigma_H = C_P \left[\frac{F_t}{bdl} \left(\frac{\cos \beta}{0.95CR} \right) K_v K_o (0.93K_m) \right]^{\left(\frac{1}{2}\right)}$$

CHAPTER-4

ANALYTICAL CONTACT STRESS OF INVOLUTE SPUR GEAR

4.1 Hertz Contact Stress:

Contact pressure for two mating spur gears is calculated using hertz equation. Those two mating gears are taken from lathe gear box as shown in figure. The details of these gears are shown below.

Table 4.1: Specifications of Gear

S. No	PINION	GEAR
Module (mm)	5	5
No. of Teeth	20	20
Pitch circle dia.(mm)	100	100

4.2 Nomenclature of parameters in Hertz theory:

- A is the transverse pressure angle, deg
- a is the addendum, mm
- b is the length of contact, mm
- β is the helical angle, deg
- C_p is the elastic coefficient factor
- D is the pitch diameter of pinion and gear, mm
- E is the Young's modulus, GPa
- F_t is tangential force on gear tooth, N
- F_n is normal force on gear tooth, N
- i is the speed ratio
- I is geometry factor
- K_v is dynamic factor

- K_O is overload factor
- K_M is load distribution factor
- K_S is size factor
- n_1 and n_2 number of teeth, suffix 1 for the pinion and 2 for the gear
- σ_c is Contact stresses, Mpa
- R is the radial distance from Centre to pitch circle, mm
- r_1 and r_2 are the pitch radii of pinion and gear, mm
- μ is the coefficient of friction
- ν is the Poisson's ratio
- V is the velocity of rotation, m/s
- Z_R is surface condition factor

The **AGMA** Contact stress equation is given by,

$$\sigma_H = C_F \left[\frac{F_t}{bdl} \left(\frac{\cos \beta}{0.95CR} \right) K_v K_o (0.93K_m) \right]^{\left(\frac{1}{2}\right)}$$

Bending Stress is given by,

$$\sigma_b = \frac{F_t}{bm_j} K_v K_o (0.93K_m)$$

Where J is Geometry Factor

$$\sigma_b = w^t K_o K_v K_s \frac{1}{bm_c} \frac{K_H K_B}{Y_j}$$

Hertz Contact Equation is given by,

$$\sigma_c = C_P \left[W^T K_v K_o K_s \frac{K_m C_f}{d_p F_I} \right]^{\left(\frac{1}{2}\right)}$$

Hertz contact stresses for different parameters like module, power, are easily calculated by making an interface in VISUAL BASIC (VB).

VB makes an easy way to calculate much iteration for calculating contact pressure and stresses.

CHAPTER-5

VISUAL BASICS

5.1 Introduction:

What is Visual Basic?

- Visual Basic is a tool that allows you to develop Windows (Graphic User Interface - GUI) applications. The applications have a familiar appearance to the user.
- Visual Basic is event-driven, meaning code remains idle until called upon to respond to some event (button pressing, menu selection...). Visual Basic is governed by an event processor. Nothing happens until an event is detected. Once an event is detected, the code corresponding to that event (event Procedure) is executed. Program control is then returned to the event processor.

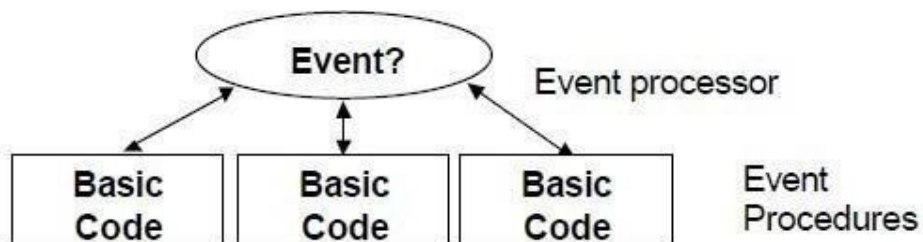


Fig. 5.1: Event Processor

- Some Features of Visual Basics
 - Full set of objects - you 'draw' the application
 - Lots of icons and pictures for your use
 - Response to mouse and keyboard actions
 - Clipboard and printer access
 - Full array of mathematical, string handling, and graphics functions
 - Can handle fixed and dynamic variable and control arrays
 - Sequential and random access file support
 - Useful debugger and error-handling facilities
 - Powerful database access tools
 - ActiveX support
 - Package & Deployment Wizard makes distributing your applications

5.2 Drawing the User Interface and Setting Properties:

- Visual Basic operates in three modes.
 - Design mode - used to build application
 - Run mode - used to run the application
 - Break mode - application halted and debugger is available
- We focus here on the design mode.
- Six windows appear when you start Visual Basic.
- The Main Window consists of the title bar, menu bar, and toolbar. The title bar indicates the project name, the current Visual Basic operating mode, and the current form. The menu bar has drop-down menus from which you control the operation of the Visual Basic environment. The toolbar has buttons that provide shortcuts to some of the menu options. The main window also shows the location of the current form relative to the upper left corner of the screen (measured in tips) and the width and length of the current form.

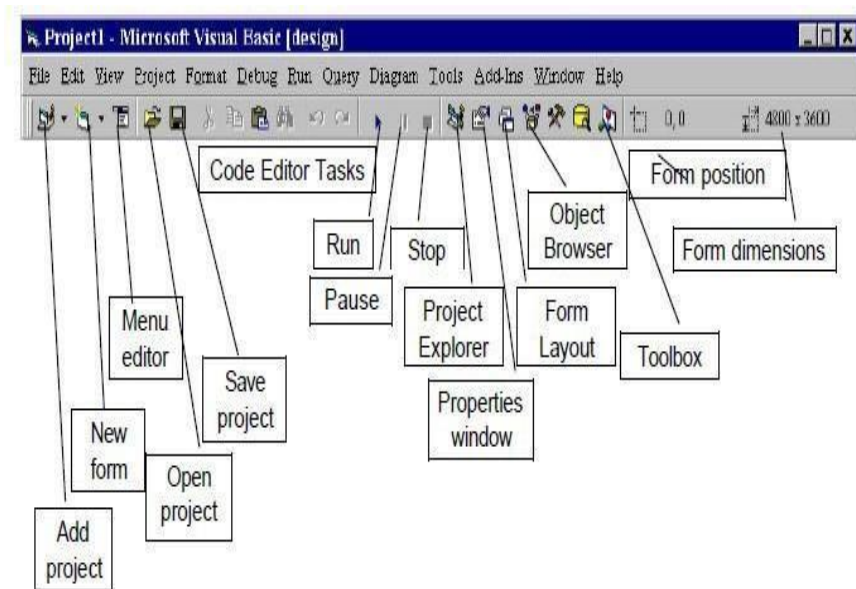


Fig. 5.2: Main window

- The Form Window is central to developing Visual Basic applications. It is where you draw your application.

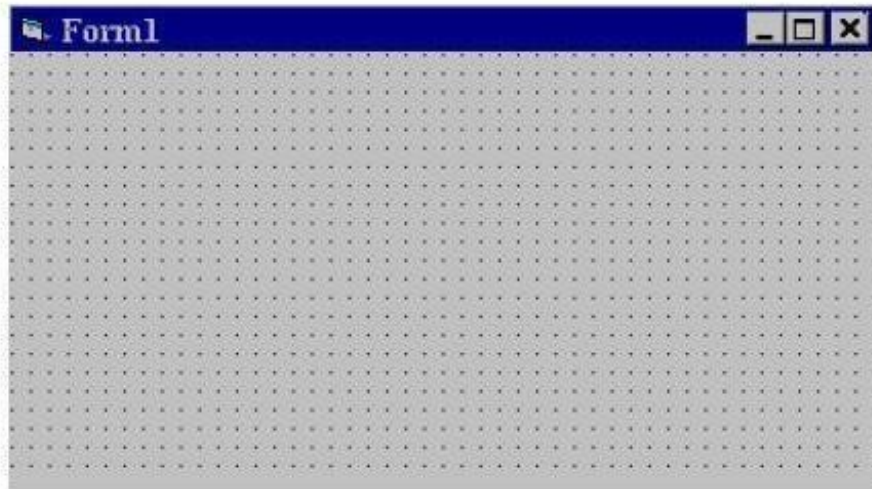


Fig. 5.3: Form window

The Toolbox is the selection menu for controls used in your application.

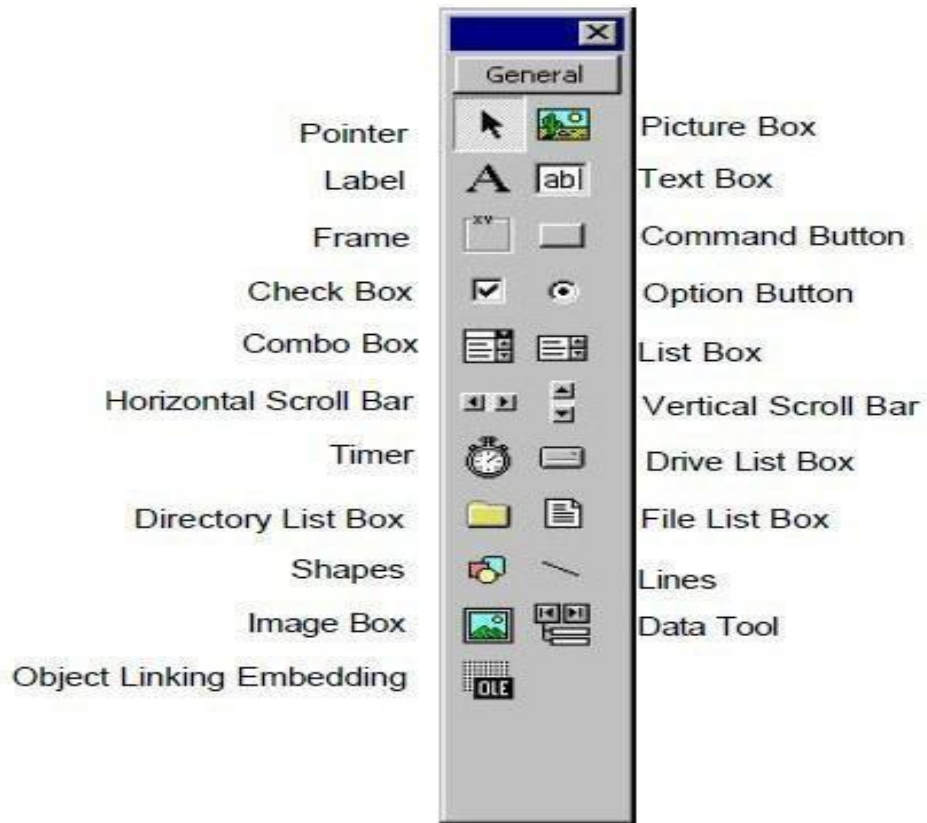


Fig. 5.4: Tool box

- The Properties Window is used to establish initial property values for objects. The drop-down box at the top of the window lists all objects in the current form. Two views are available: Alphabetic and Categorized. Under this box are the available properties for the currently selected object

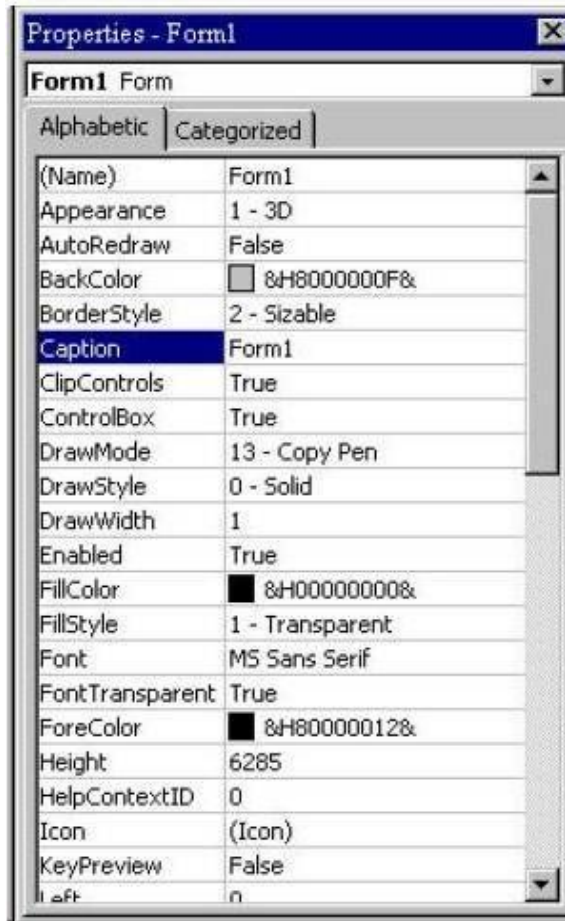


Fig. 5.5: Properties window

- The Form Layout Window shows where (upon program execution) Your form will be displayed relative to your monitor's screen:

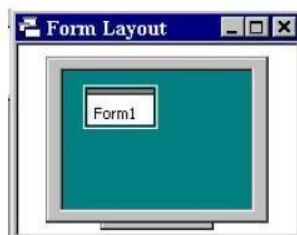


Fig. 5.6: Form layout window

The **project window** displays a list of all forms and modules making up your application. You can also obtain a view of the **Form** or **Code** windows (window containing the actual basic coding) from the Project window.

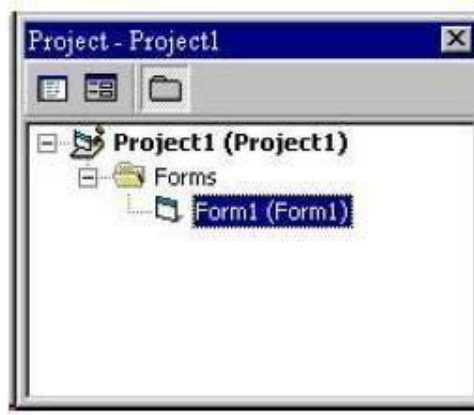


Fig. 5.7: Project window

➤ As mentioned, the user interface is 'drawn' in the form window. There are two Ways to place controls on a form.

1. Double-click the tool in the toolbox and it is created with a default size on the form. You can then move it or resize it.

2. Click the tool in the toolbox, and then move the mouse pointer to the form window. The cursor changes to a crosshair. Place the crosshair at the upper left corner of where you want the control to be, press the left mouse button and hold it down while dragging the cursor toward the lower right corner. When you release the mouse button, the control is drawn.

➤ To **move** a control you have drawn, click the object in the form window and drag into the new location. Release the mouse button.

➤ To **resize** a control, click the object so that it is select and sizing handles appear. Use these handles to resize the object.

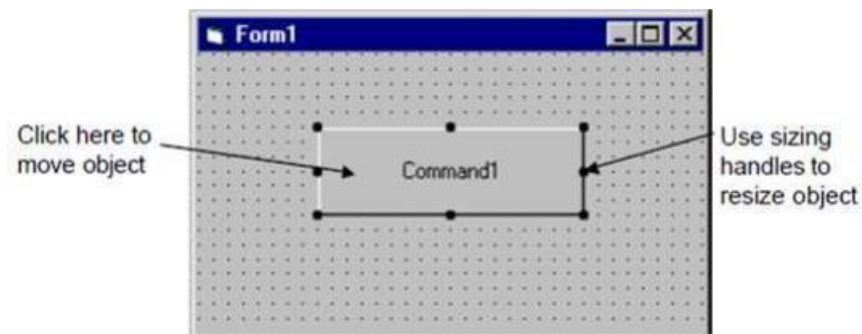


Fig. 5.8: Form window

5.3 Interface:

With the help of visual basics an interface is prepared for calculating Hertzian contact stresses.

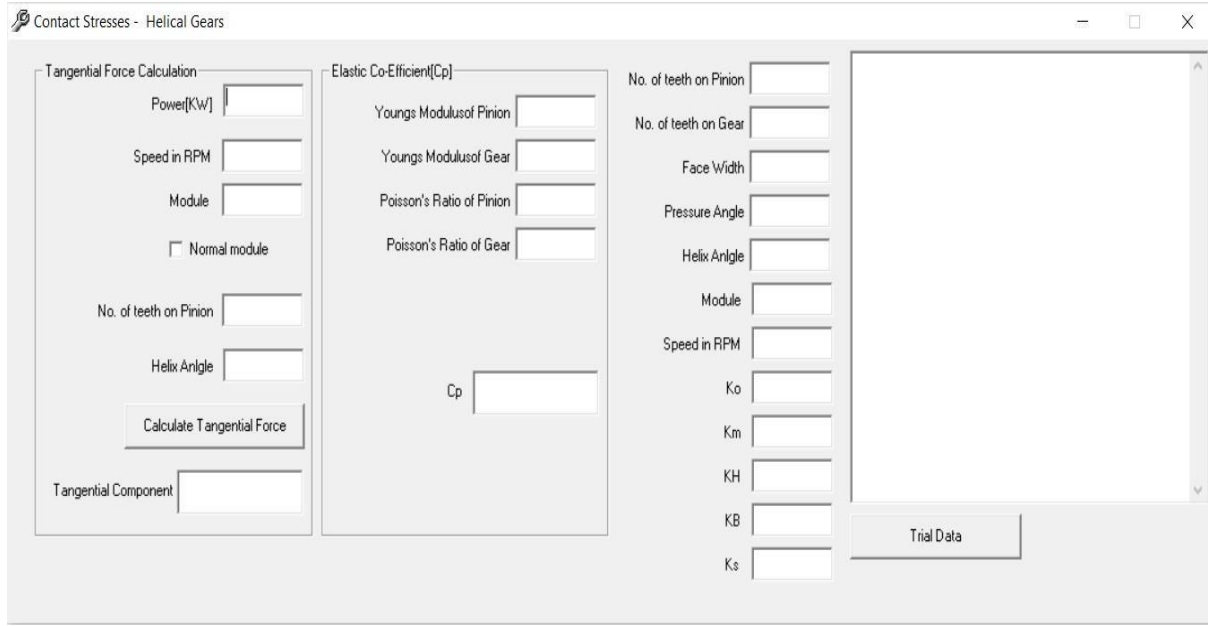


Fig. 5.9: Main interface

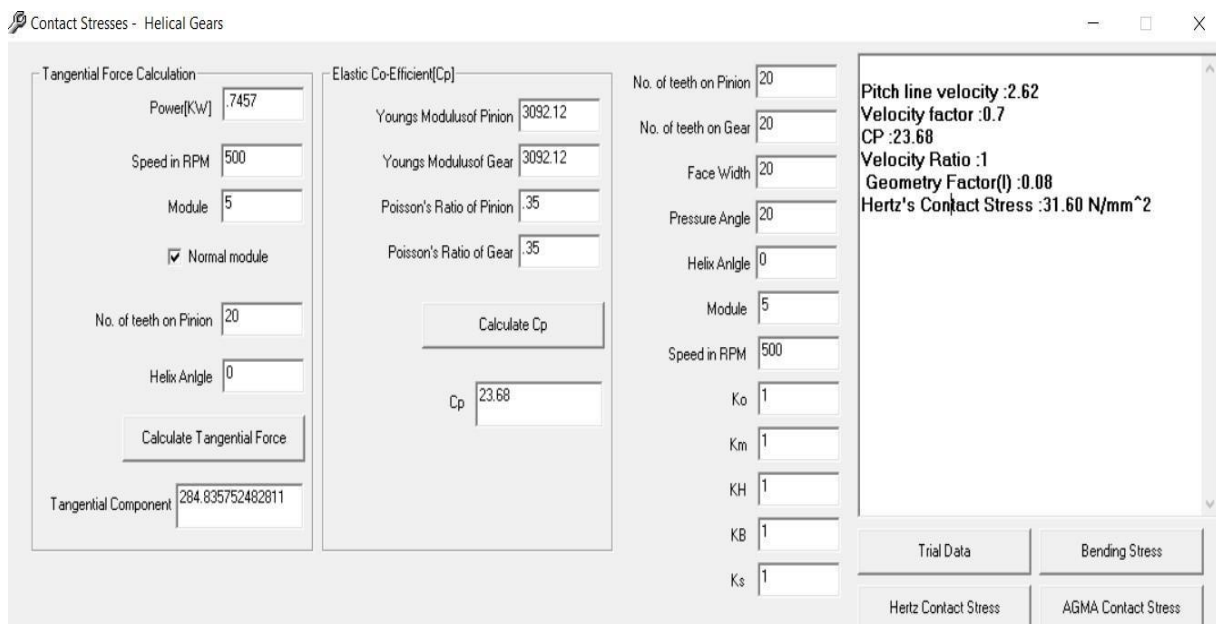


Fig. 5.10: Hertz's contact stresses of Acetal copolymer

5.4 Program Code for Spur Gear Interface:

```
Private Sub Command1_Click ()
```

```
Dim ft As Double
```

```
Dim zp AsDouble
```

```
Dim zg AsDouble
```

```
Dim b As Double
```

```
Dim pa AsDouble
```

```
Dim ha AsDouble
```

```
Dim e1 AsDouble
```

```
Dim e2 AsDouble
```

```
Dim v1 AsDouble
```

```
Dim v2 AsDouble
```

```
Dim m As Double
```

```
Dim n As Double
```

```
DimkoAsDouble
```

```
Dim kmAsDouble
```

```
Dim kh AsDouble
```

```
Dim kb AsDouble
```

```
Dim kvAsDouble
```

```
Dim v As Double
```

```
txtresult.Text = ""
```

```
ft = txtpt.Text
```

```
m = txtm.Text
```

```
zp =txtzp.Text
```

```
zg =txtzg.Text
```

```
b = txtb.Text
```

```
Pa =txtpa.Text
```

```
h =txtha.Text
```

```

n = txtn.Text
KO = txtko.Text
Km=txtkm.Text
Kh = txtkh.Text
Kb = txtkb.Text
Ks = txtks.Text
m_temp txtm.Text
ha = txtha.Text
har = (3.1416 * ha) / (180)

If chkmodule.Value = 1 Then m = (m_temp) / (Cos(har))
Else
m = m_temp
End If

dp = Val(m) * zp n = txtn.Text
txtresult.Text = txtresult.Text & vbCrLf & "Pitch circle Diameter :" & dp v1 = (3.1416)
* (Val(dp)) * (Val(n))

v = FormatNumber((v1 / 60000), 2)

txtresult.Text = txtresult.Text & vbCrLf & "Pitch line velocity :" & v kv =
FormatNumber((3 / (3 + v)), 2)
txtresult.Text = txtresult.Text & vbCrLf & "Velocity factor :" & kv yp =
FormatNumber((0.154 - (0.912 / zp)), 2)
txtresult.Text = txtresult.Text & vbCrLf & "Lewis form factor :" & yp

BS = FormatNumber(((ft * ko * kv * kb * kh * kb * ks) / (m * b * 3.1416 *yp)),2)
txtresult.Text = txtresult.Text & vbCrLf & "Bending Stress:" & BS &: N/mm^2")
End Sub

```

```
Private Sub Command2_Click ()
Dim ft As Double
Dim zp As Double
Dim zg As Double
Dim b As Double
Dim pa As Double
Dim ha As Double
Dim e1 As Doubl
Dim e2 As Double
Dim v1 As Double
Dim v2 AsDouble
Dim m as Double
Dim n as Double
Dim KO AsDouble
Dim km Double
Dim kh asDouble
Dim ko asDouble
Dim kv AsDouble
Dim v As Double
txtresult.Text = ""
ft = txtpt.Text
m = txtm.Text
zp =txtzp.Text
zg = txtzg.Text
b = txtb.Text
n = txtn.Text
ko = txtko.Text km=txtkm.Text
kh = txtkh.Text
```

```

Kb = txtkb.Text
m_temp = txtm.Text
Ha = txthad.Text
har = (3.1416 * ha) / (180)
If chkmodule.Value=1 Then m = (m_temp)/(Cos(har))
Else
m = m_temp
End If
dp = Val(m) * zp n = txtn.Text
txtresult.Text = txtresult.Text & vbCrLf & "Pitch circle Diameter :" & dp v_t = (3.1416)
* (Val(dp)) * (Val(n))
v = FormatNumber ((v_t / 60000), 2)
txtresult.Text = txtresult.Text & vbCrLf & "Pitch line velocity :" & v
kv = FormatNumber((6 / (6 + v)), 2)
txtresult.Text = txtresult.Text & vbCrLf & "Velocity factor :" & kv yp =
FormatNumber((0.154 - (0.912 / zp)), 2)
cp = txtcp.Text
txtresult.Text = txtresult.Text & vbCrLf & "CP :" & txtcp.Text
i = zg / zp
txtresult.Text = txtresult.Text & vbCrLf & "Velocity Ratio :" & i i_t1 = (Cos(par)) *
(Sin(par)) / 2
i_t2 = i / (i + 1)
i_t = i_t1 * i_t2
i_t = FormatNumber(i_t, 2)
txtresult.Text = txtresult.Text & vbCrLf & "Geometry Factor(I) :" & i_t h_c_s = ((ft) /
(dp * b * i_t))
h_c_s = Sqr(h_c_s)
h_c_s = cp * h_c_s
h_c_s = FormatNumber (h_c_s, 2)
txtresult.Text = txtresult.Text & vbCrLf & "Hertz's Contact Stress:" & h_c_s & "
N/mm^2"
End Sub

```



```
Private Sub
Command3_Click()
Dim ft As Double
Dim zpAsDouble
Dim zgAsDouble
Dim b As Double
Dim paAsDouble
Dim ha AsDouble
Dim e1 AsDouble
Dim e2 AsDouble
Dim v1 AsDouble
Dim v2 AsDouble
Dim m as Double
Dim n as Double
Dim KO AsDouble
Dim kmAsDouble
Dim kh AsDouble
Dim kb AsDouble
Dim kv AsDouble
Dim v As Double txtresult.Text = ""
Ft = txtpt.Text
m = txtm.Text
Zp =txtzp.Text
Zg =txtzg.Text
b = txtb.Text
Pa = txtpa.Text
```

```

Par = (3.1416 * pa) / (180)
ha = txtha.Text
har = (3.1416 * ha) / (180)
n = txtn.Text
ko = txtko.Text km=txtkm.Text
kh = txtkh.Text
kb = txtkb.Text
m_temp txtm.Text
ha = txthad.Text
har = (3.1416 * ha) / (180)

If chkmodule.Value = 1 Then m = (m_temp) / (Cos(har))
Else

m = m_temp

End If

dp = Val(m) * zp n = txtn.Text
txtresult.Text = txtresult.Text & vbCrLf & "Pitch circle Diameter :" & dp v_t = (3.1416)
* (Val(dp)) * (Val(n))
v = FormatNumber((v_t / 60000), 2)
txtresult.Text = txtresult.Text & vbCrLf & "Pitch line velocity :" & v kv1 = 78 + ((200 *
v) ^ 0.5)
kv = kv1 / 78
kv = kv ^ 0.5
kv = FormatNumber(kv, 2)
txtresult.Text = txtresult.Text & vbCrLf & "Velocity factor:" & kv yp =
FormatNumber((0.154 - (0.912 / zp)), 2)
txtresult.Text = txtresult.Text & vbCrLf & "CP :" & txtcp.Text i = zg / zp

```

```

txtresult.Text = txtresult.Text & vbCrLf & "Velocity Ratio :" & i_t1 = (Cos(par)) *
(Sin(par)) / 2
i_t2 = i / (i + 1)
i_t = i_t1 * i_t2
i_t = FormatNumber(i_t, 2)
txtresult.Text = txtresult.Text & vbCrLf & " Geometry factor(I) :" & i_t r1 = (m * zp) / 2
r2 = (m * zg)/2 a = m
rb1 = r1 - (1.25 * m)
rb2 = r2 - (1.25 * m)
cr1 = ((r1 + m) ^ 2) - (rb1 ^ 2)
cr1 = Sqr(cr1)
'txtresult.Text = txtresult.Text & vbCrLf & "CR1 :" & cr1 cr2 = ((r2 + m) ^ 2) - (rb2 ^ 2)
cr2 = Sqr(cr2)
'txtresult.Text = txtresult.Text & vbCrLf & "CR2 :" & cr2 cr3 = (r1 + r2) * (Sin(par))
'txtresult.Text = txtresult.Text & vbCrLf & "CR3 :" & cr3 cr4 = (3.1416 * m * (Cos(par)))
cr = (cr1 + cr2 - cr3) / (cr4)
txtresult.Text = txtresult.Text & vbCrLf & "CR :" & cr
agma1 = (Cos(har)) / (0.95 * cr)
agma2 = (ft) / (b * m * zp * i_t)
agma3 = agma1 * agma2 * kv * ko * 0.93 * km
agma3 = Sqr(agma3)
cp = txtcp.Text
agma = Val(cp) * agma3
agma = FormatNumber(agma, 2)
txtresult.Text = txtresult.Text & vbCrLf & "AGMA Contact Stress :" & agma & "
N/mm^2"
End Sub

Private Sub Command4_Click () txtpower.Text = "31.6"
txtzp1.Text = "20"
txtzg.Text = "20"

```

```

txtb.Text = "20"
txtpa.Text = "20" txthad.Text = "5" txte11.Text = "210000" txte21.Text = "210000"
txtv11.Text = "0.3" txtv21.Text = "0.3" txtm1.Text = "4.5" txtrpm1.Text = "1000"
txtko.Text = "1" txtkm.Text = "1" txtkh.Text = "1" txtkb.Text = "1"
txtks.Text = "1"
End Sub

```

```

Private Sub Command5_Click ()
Dim mt As Double
Dim pt As Double
Dim n1 As Double
Dim zp1 As Double m_temp=txtm1.Text
ha = txthad.Text
har = (3.1416 * ha) / (180)
If chkmodule.Value=1Thenm = (m_temp) / (Cos(har))
Else
m = m_temp
End If
zp1=txtzp1.Text dp = m * zp1
Power=txtpower.Text
n1 = txtrpm1.Text
mt = (power * 60 * 10 ^ 6) / (2 * 3.1416 * n1)
pt = (2 * mt) / (dp) txtpt.Text = pt Command6.Visible=True
End Sub

```

```

Private Sub Command6_Click ()
Dim e1 As Double
Dim e2 As Double
Dim v1 As Double
Dimv2AsDouble Dimcp1As Double
Dim cp2AsDouble
Dim cp As Double

```

```

e1 = txtel1.Text
e2= txte21.Text
v1=txtv11.Text v2=txtv21.Text
cp1=(1 - (v1 ^ 2)) / (Val(e1)) cp2=(1 - (v2 ^ 2)) / (Val(e2)) cp = (3.1416 * (cp1 + cp2))
cp = 1 / cp
cp = cp ^ 0.5
cp =FormatNumber(cp,2) txtcp.Text
cp Command1.Visible=True Command2.Visible=True Command3.Visible=True
End Sub
Private Sub Form_Load ()
End Sub
Private Sub txthad_Change ()
txtha.Text = txthad.Text
End Sub
Private Sub txtm1_Change ()
txtm.Text = txtm1.Text
End Sub
Private Sub txtrpm1_Change ()
txtn.Text = txtrpm1.Text
End Sub
Private Sub txtrpm1_Click () txtn.Text = txtrpm1.Text
End Sub
Private Sub txtzp1_Change ()
txtzp.Text = txtzp1.Text
End Sub

```

CHAPTER-6

SOLIDWORKS

6.1 Introduction:

SolidWorks is design automation software. In Solid Works, you sketch ideas and experiment with different designs to create 3D models. Solid Works is used by students, designers, engineers, and other professionals to produce simple and complex parts, assemblies, and drawings.

Solid works works the way engineers design and think and that is why it has become successful so quickly. Engineers and drafters say that it is easy to learn and gives them a model that they have complete confidence in manufacturing and know that it will work, just by using the tools provided with this one piece of software. Solid works is powerful. The figure 6.1 shows the solid works interface.



Fig. 6.1: Solidworks interface

- To start our projects select the sheet of paper shown. Notice there is another sheet of paper on the right side panel and you can use that one as well.
- Select PART from the dialog box shown and SELECT OK. We are going to make our FIRST part or drawing in Solid works. Basically solid works model is made up of PART, ASSEMBLY & DRAWING.

- Next, we are asked for a PLANE to begin our sketch. According to our drawing requirement, select the plane (as shown in the figure 6.2) in which the part must be drawn. It should HIGHLIGHT and use your left mouse button to SELECT it.

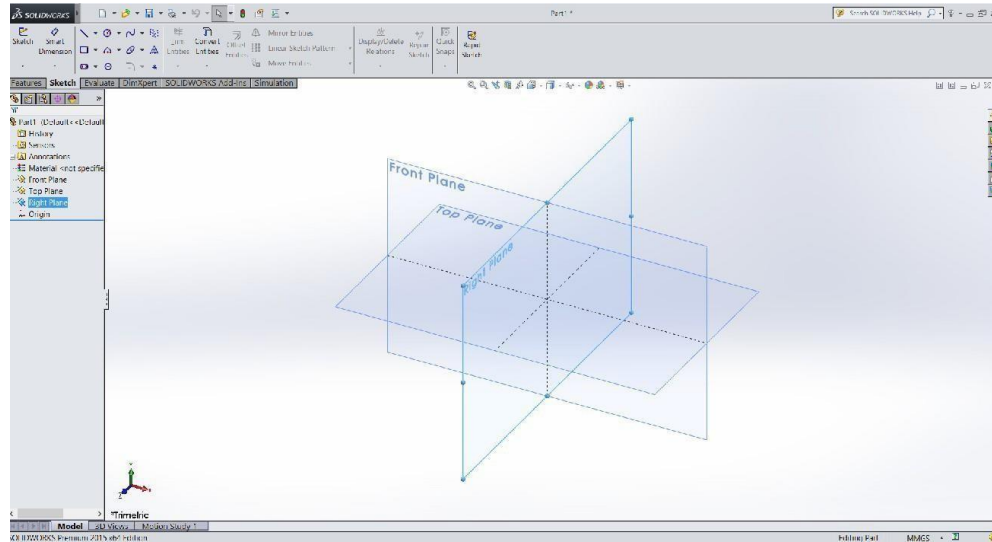


Fig.6.2: Plane Selection

- Construct the drawing of required part and by giving the dimensions by smart dimensioning.
- After constructing all the parts of the component, assemble the parts to obtain the component. Further obtain the drawing of the component.

6.2 Intended Audience:

This document is for new Solid Works users. In this document, you are introduced to concepts and design processes in a high-level approach. Solid Works Help contains a comprehensive set of tutorials that provide step-by-step instruction on many of the features of Solid Works.

6.3 Terminology:

These terms appear throughout the Solid Works software and documentation as shown in the figure 6.3.

- **Origin:** Appears as two blue arrows and represents the (0, 0, 0) coordinate of the model. When a sketch is active, a sketch origin appears in red and represents the (0, 0,0) coordinate of the sketch. You can add dimensions and relations to a model origin, but not to a sketch origin

- **Plane:** Flat construction geometry. You can use planes for adding a 2D sketch, section view of a model, or a neutral plane in a draft feature, for example.
- **Axis:** Straight line used to create model geometry, features, or patterns. You can create an axis in different ways, including intersecting two planes. The Solid Works application
- **Face:** Boundaries that help define the shape of a model or a surface. A face is a Selectable area (planar or non-planar) of a model or surface. For example, a rectangular solid has six faces.
- **Edge:** Location where two or more faces intersect and are joined together. You can select edges for sketching and dimensioning, for example.
- **Vertex:** Point at which two or more lines or edges intersect. You can select vertices sketching and dimensioning, for example.

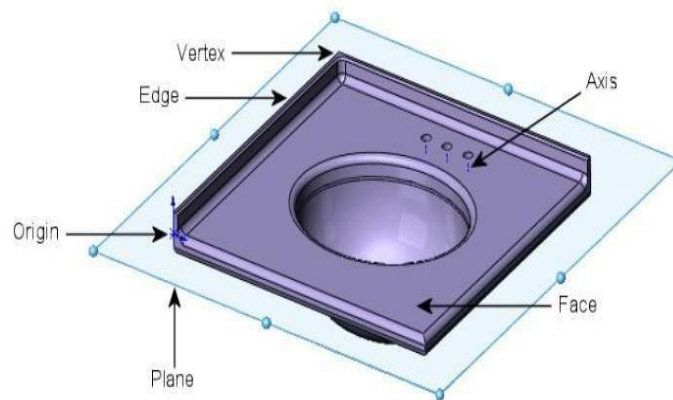


Fig.6.3: Model Terminology

6.4 Procedure for generating Involute Spur gear in Solid works:

- Select a new document of 3D arrangement of parts as shown
- In sketch select front plane
- Draw pitch circle having diameter 100mm

TABLE 6.1: Data used for Gear Construction:

Parameter	Spur gear	
	Pinion	Gear
No. of teeth	20	20
module	5	5
Pressure angle (deg)	20	20
Pitch diameter(mm)	100	100
Face Width (mm)	20	20
Contact ratio	1	1

- Using offset command and taking offset distance 5mm, offset the pitch circle outwards to get addendum circle.
- Using offset command and taking offset distance 6.25mm, offset the pitch circle inside to get dedendum circle.
- Draw base circle having diameter 90mm.

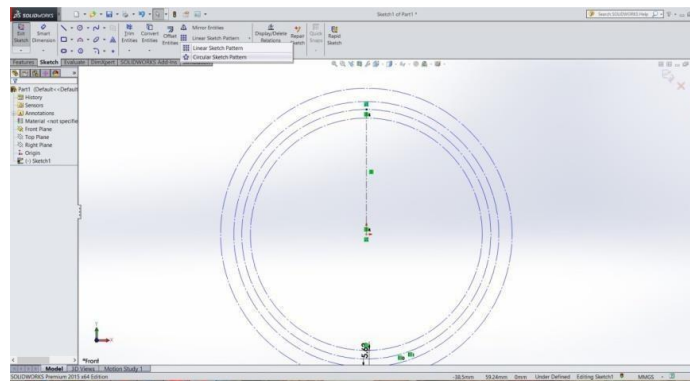


Fig. 6.4: Pitch circle and Base circle

- Divide $1/4^{\text{th}}$ of the circle into 10 equal parts using circular pattern command giving angle 45° and 10 number of lines.
- Draw tangents to the base circle from these projection lines leaving the first projection line and the tangents are started from the adjacent line.

- The length of first tangent will be equal to arc length between any two projection lines, and the second tangent length will be equal to two times the arc length and so on.
- Using spline command draw a curve touching the edges of the tangents, trimout the extra curve exceeding the addendum circle. Hence the curve generated is the required involute curve.
- Now delete all the projection lines leaving the first line and also delete the tangents of the base circle.

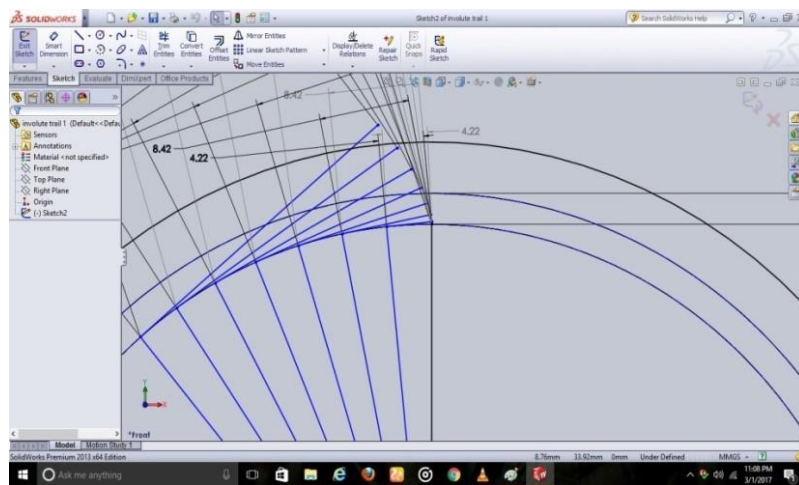


Fig. 6.5: Involute tooth Profile construction



Fig. 6.6: Involute tooth Profile

- Now using line command project a line from centre point to involute curve where the curve meets the pitch circle.
- From this point taking angular distance 7.186mm draw another line to the pitch circle, this angular distance defines the tooth thickness.
- Draw a line which is centre to these two lines and mirror the involute curve taking mirror plane as the centre line.
- Now from the copied involute curve taking angular distance 15.707mm draw a line touching the pitch circle that distance defines the circular pitch.
- Now generate another involute curve from this point by copying it from already generated involute curve. At bottom draw lines from base circle to dedendum circle and provide sufficient fillet radius.

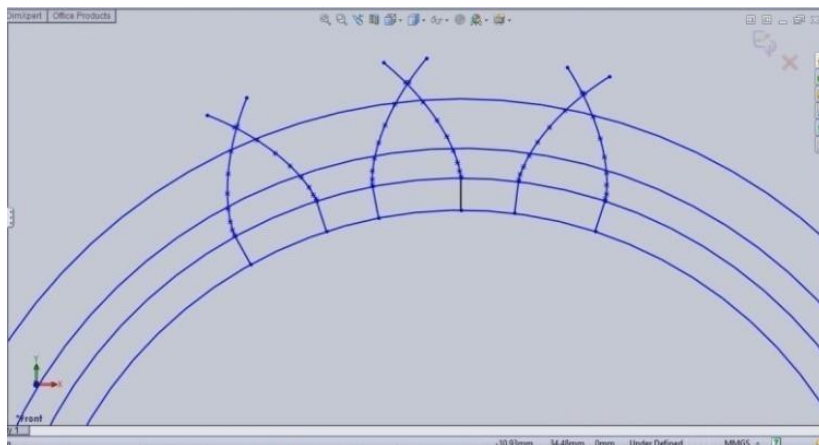


Fig.6.7. Three Involute teeth

- Now trim out all the unwanted profiles leaving the closed boundary. Now exit from the sketch and in front view again take sketch and draw the addendum circle and extrude it according to the required face width.
- Now go to sweep cut in features menu bar and select the boundary region to sweep cut and select the spiral curve for direction of cut and press ok the cut the profile and required profile is generated.
- In features menu go to circular pattern and select the swept portion and in parameters select the circumference and give pattern 20 to generate 20 teeth spur gear.

- To get the profile of three teeth select the face and go to sketch and draw lines such that only three teeth are excluded and now go to extruded cut and select the plane to remove the unwanted part.
- Hence required spur gear with 3 teeth is generated.
- Now the construction lines to represent the removed part of the gear for the Shaft hole are to be drawn.
- Select the required face of the gear, opt sketch and drawn lines in order to

6.5 Assembly of Gears:

- Open a new window in solid works and click assembly of components and
Insert gear components that were already prepared.
- Now draw a sketch in top plane representing the axes of the respective gears separated by distance 100 mm between them.

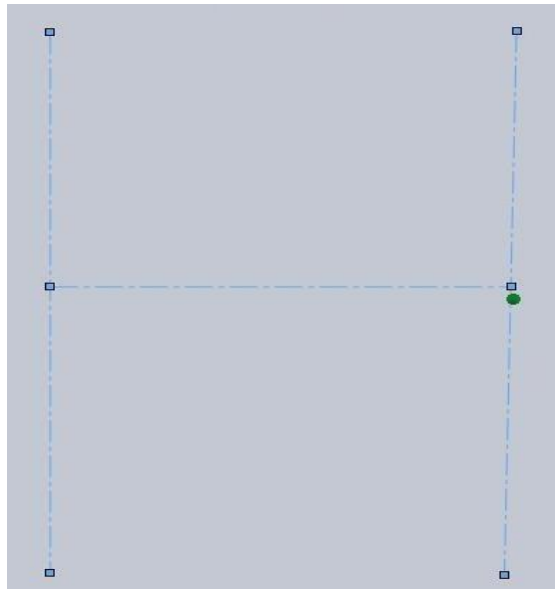


Fig.6.8. Axes of the gears with centre distance

- To draw inclined axis, draw another line with inclination to the original axis from the Centre and extend it to the required length, trim the un-necessary lines.
- Go to the mate option in menu ribbon and select the line and the concave face of the gear and keep them in concentric.
- Now select the Centre point of the gear and line, keep them coincidental.
- Select the move components option and opt for collision detection, keep in stop at collision condition and mate accordingly.
- Again, go to the mate in menu and select the concave faces of the two gears.
- Go to mechanical mates and select gear component and exit from it.
- The final assembly of the gear looks as shown below.

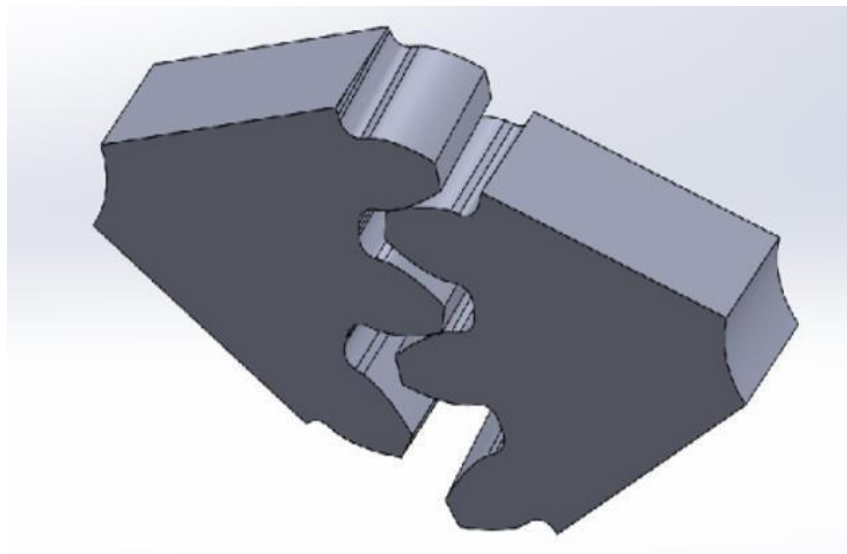


Fig. 6.9: Final Gear Profile assembly

- For the inclined gear sets, the top view of the mate looks as shown below

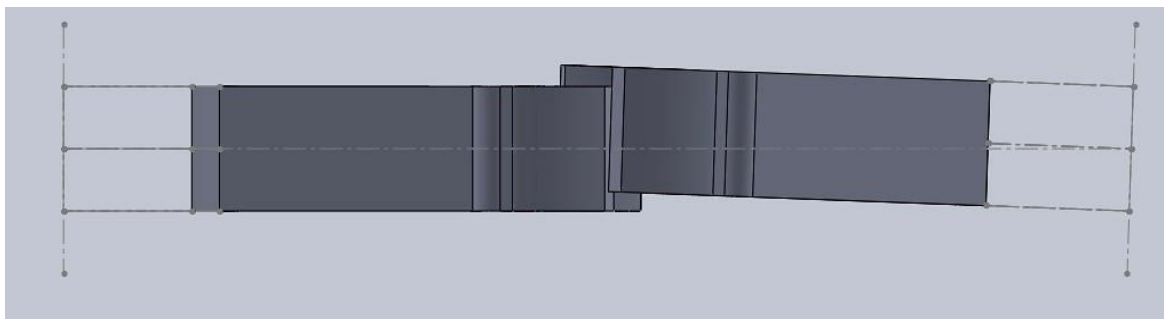


Fig. 6.10: Top view of the inclined gear assembly

CHAPTER-7

ANSYS WORKBENCH

7.1 Introduction:

ANSYS Workbench is a new-generation solution from ANSYS that provides powerful methods for interacting with the ANSYS solver functionality. This environment provides a unique integration with CAD systems, and your design process, enabling the best CAE results.

ANSYS Workbench is comprised of five modules:

- Simulation for performing structural and thermal analyses using the ANSYS solver
- CFX-Mesh for generating a CFX-Pre mesh for the CFX-5 solver
- Design Modeller for creating and modifying CAD geometry to prepare the solid model for use in Simulation or CFX-Mesh
- Design Explorer and Design Explorer VT for investigating the effect of
- Variations input to the response of the system
- FE Modeller for translating a Nastran mesh for use in ANSYS
- Every analysis involves four main steps:
 - Preliminary Decisions
 - What type of analysis: Static, modal,
 - What to model: Part or Assembly?
 - Which elements: Surface or Solid
 - Pre-processing
 - Attach the model geometry
 - Define and assign material
 - Mesh the geometry
 - Apply loads and supports
 - Request results
 - Solve the Model
 - Post processing
 - Review results

- Check the validity of the solution

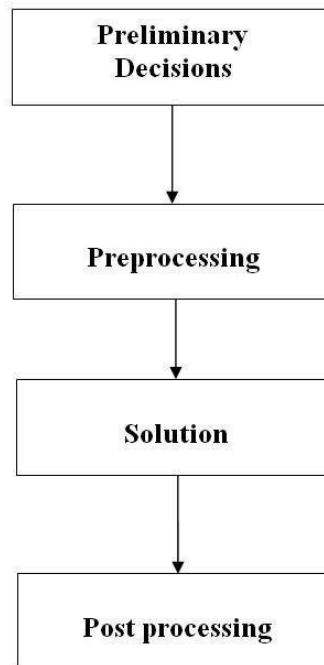


Fig. 7.1: ANSYS Procedure

7.2 ANSYS Work Bench Analysis:

The ANSYS Workbench platform is the framework upon which the industry's broadest and deepest suite of advanced engineering simulation technology is built. An innovative project schematic view ties together the entire simulation process, guiding the user through even complex metaphysics analyses with drag-and-drop simplicity. With bi- directional CAD connectivity, powerful highly-automated meshing, a project-level update mechanism, pervasive parameter management and integrated optimization tools, the ANSYS Workbench platform delivers unprecedented productivity, enabling Simulation Driven Product Development.

The Workbench environment allows you to solve much more complex analyses, including (as of ANSYS 8.0):

- Multi-part assemblies
- 3-D solid elements, shell elements, and shell-solid assemblies

- Nonlinear contact with or with or without friction
- Small-displacement and large-displacement static analyses
- Modal, harmonic, and Eigen value buckling analyses
- Steady-state thermal analysis, including temperature-dependent material properties and thermal contact.

Some of the features available in Ansys.

7.3 Graphics:

- To rotate the model, “drag” the middle mouse button.
(Drag = move the mouse while pressing down on a mouse button)
- To zoom in, drag the right mouse button and create a box around the region of interest.
- To fit the image, right-click anywhere in the graphics “Fit” on the pop-up menu. Or click on the Fit icon in the toolbar.
- To pan the model, drag the middle mouse button while pressing the Shift key.

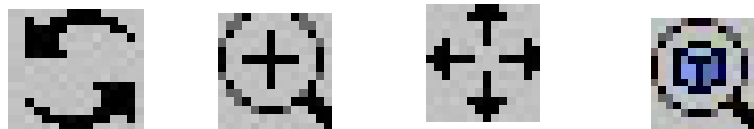


Fig. 7.2: Graphics

Picking:

- Face picking is active by default (as indicated by the Face icon attached to the mouse cursor). To pick a face, left-click anywhere on the face.
- Use the Vertex, Edge, Face, and Body icons in the toolbar to pick the respective entities. Or right-click to activate the pop-up menu and choose Cursor Mode.
- To pick multiple entities, hold down the Ctrl key while picking the additional entities. The status region at the bottom shows the number of entities currently picked.

Experiment with various contour options (available on the Context toolbar):

- “Geometry” icon: Switch to “Iso surfaces.” Zoom in and rotate the Iso surfaces plot as desired, then switch back to “Exterior.”
- “Contours” icon: Switch to “Smooth Contours.”
- “Edges” icon: Switch to “Show Elements.” Zoom in and rotate as desired, then switch back to “No Wireframe.”



Fig. 7.3: Various Contours

Animate the results:

- Click the “Animate” icon in the Context toolbar to animate the results. You may pan, zoom, and rotate the model during animation.
- To save the animation, click the “Export Animation File” icon.



Fig. 7.4: Export Animation File

7.4 Types of Contact Behaviors in Work Bench

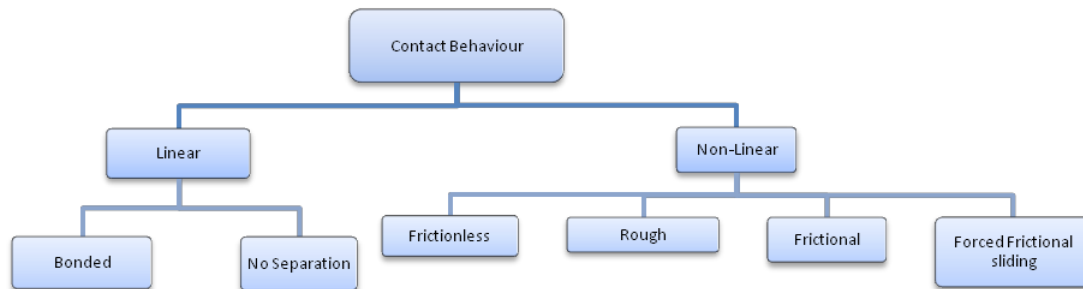


Fig. 7.5: Types of Contact Behaviours

- **Bonded Contact:** No, penetration, separation, or sliding between faces or edges.
- **No separation Contact:** Similar to bonded, except frictionless sliding can occur along contacting faces

- **Frictionless Contact:** No penetration allowed, but surfaces are free to slide and separate without resistance.
- **Rough contact:** Similar to the frictionless setting except no sliding allowed (i.e., friction coefficient=infinite).
- **Frictional Contact:** Allows sliding with resistance proportional to user-defined coefficient of frictional free to separate without resistance.

Type	Frictional
<input type="checkbox"/> Friction Coefficient	Bonded
Scope Mode	No Separation
Behavior	Frictionless
Trim Contact	Rough
Suppressed	Frictional
	Forced Frictional Sliding
	NO

Fig. 7.6: Types of contacts in Ansys Work Bench

7.5 Step by step procedure analysis in Ansys work bench:

- Open Ansys workbench analysis version 14
- Ansys work bench environment will be seen. Then assembled part in SOLIDWORKS is imported in (.step) format to Ansys by going through this step Ansysworkbench environment will be seen like this.

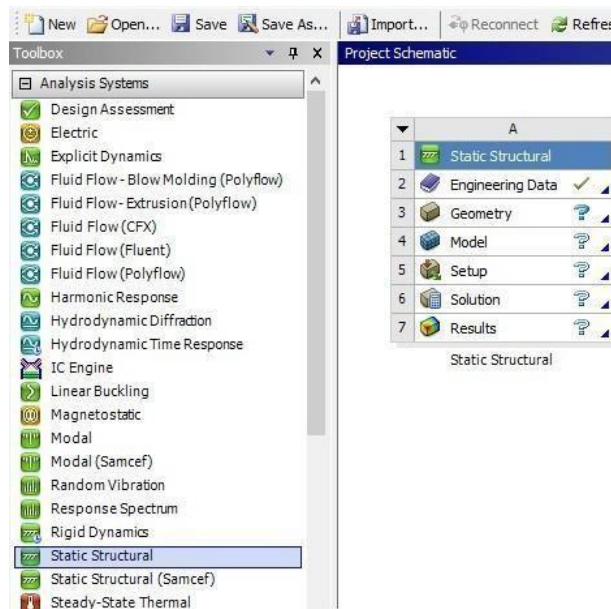


Fig. 7.7: Work bench environmen

- Engineering data such as young's modulus, density, poissons ratio is given to the material Structural Steel (AISI1040).

	A	B	C	D	E
1	Property	Value	Unit		
2	Density	7850	kg m ⁻³		
3	Isotropic Secant Coefficient of Thermal Expansion				
6	Isotropic Elasticity				
7	Derive from	Young's Mo...			
8	Young's Modulus	2E+11	Pa		
9	Poisson's Ratio	0.3			
10	Bulk Modulus	1.6667E+11	Pa		
11	Shear Modulus	7.6923E+10	Pa		
12	Alternating Stress Mean Stress	Tabular			
16	Strain-Life Parameters				

Fig. 7.8: Selection of material

S. No	Material	Density (g/cc)	Tensile Strength (MPa)	Modulus Of Elasticity (MPa)	Poisson Ratio
1	Acetal copolymer	1.41	70	3092.12	0.35
2	Cast Nylon	1.15	83.3565	3138.128	0.25

Table 7.1 Mechanical Properties of the Material

After importing the geometry, the environment will be seen like this.

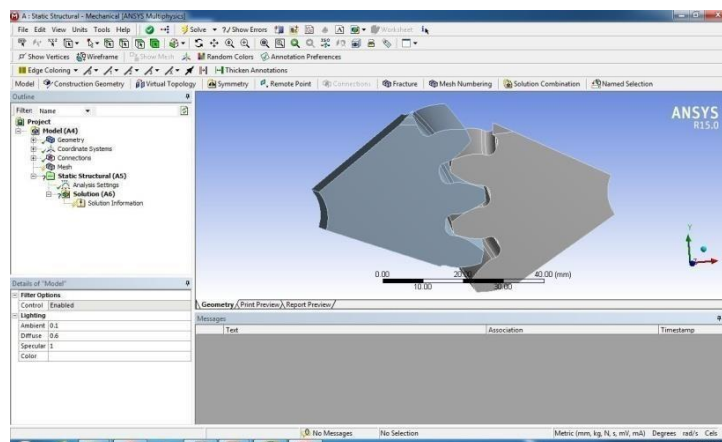


Fig. 7.9: Contact region

Meshing is done by using mesh tool and relevant mesh size is given and SOLID186 element is used in Meshing.

Lid element that exhibits quadratic displacement behaviour. The element is defined by 20 nodes having three degrees of freedom per node: translations in the nodal x, y, and z directions. The element support Plasticity, hyper elasticity, creep, stress stiffening, large deflection, and large strain capabilities. It also has mixed formulation capability for simulating deformations of nearly incompressible elastic plastic materials, and fully incompressible hyper elastic materials.

In this analysis the following no of Nodes and elements are used.

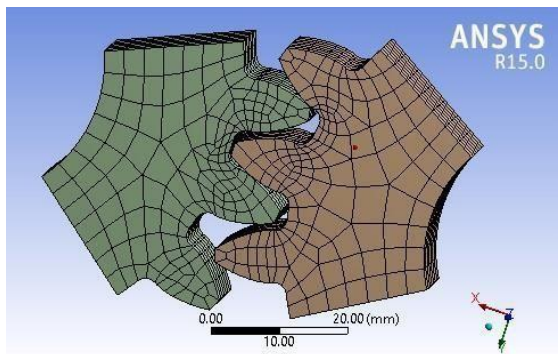


Fig.7.10: Meshing

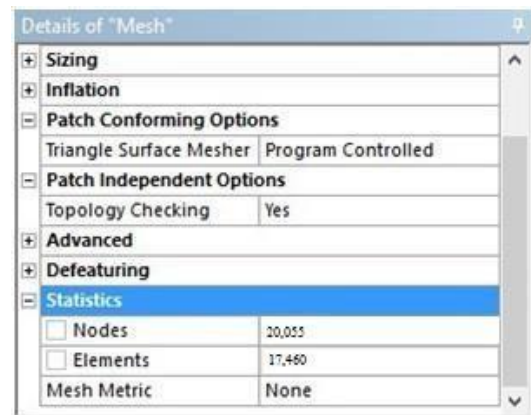


Fig.7.11: Details of Mesh

Table No 7.2 No. of Nodes, Elements:

Axes set	No. Of nodes	No. Of elements
0-0	20,055	17,460
0-0.5	22,386	19,454
0-1	25,727	21,679
0-1.5	28,450	23,953

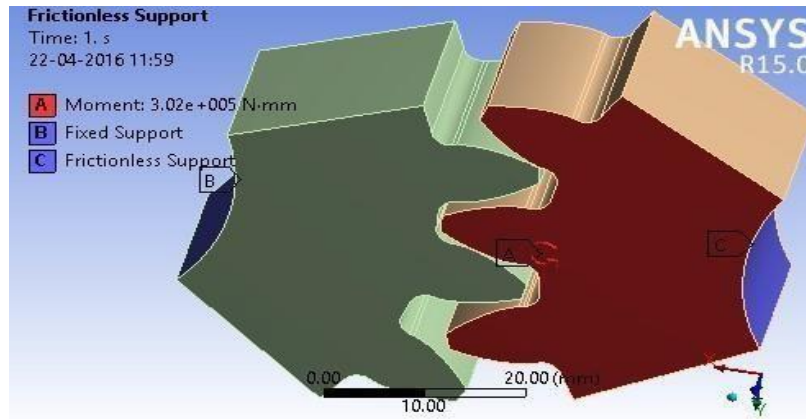


Fig. 7.12: Boundary Conditions

- To find out the deformation by inserting the deformation tool.

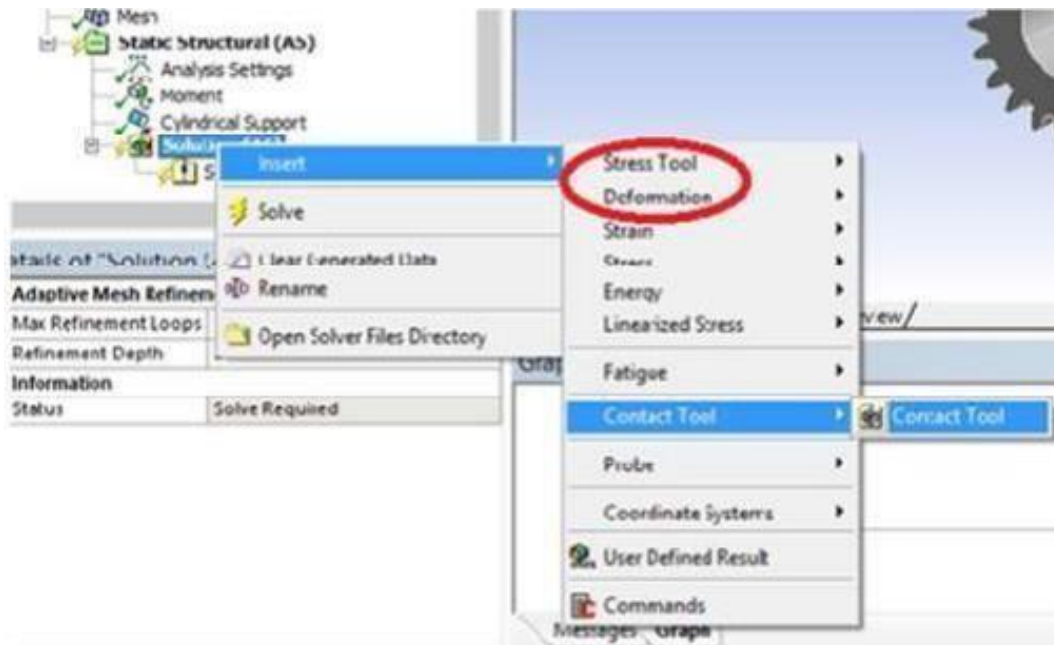


Fig. 7.13: Insert Deformation

CHAPTER-8

ANSYS RESULTS

The results below represent the von Mises stresses acting on involute spur gears with inclined axes pairs such as $0^\circ-0^\circ$; $0^\circ-0.5^\circ$; $0^\circ-1.0^\circ$; $0^\circ-1.5^\circ$.

The results summarize that the von Mises stresses of ACETAL COPOLYMER shows an increasing tendency with increase in inclination of the axes.

8.1 Results of von Mises stresses when central distance is 100.00mm :

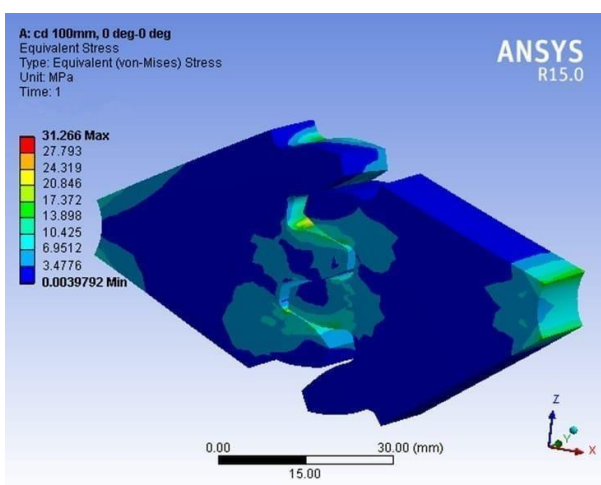


Fig.8.1. a gear 1 at 0° , gear 2 axis at 0°

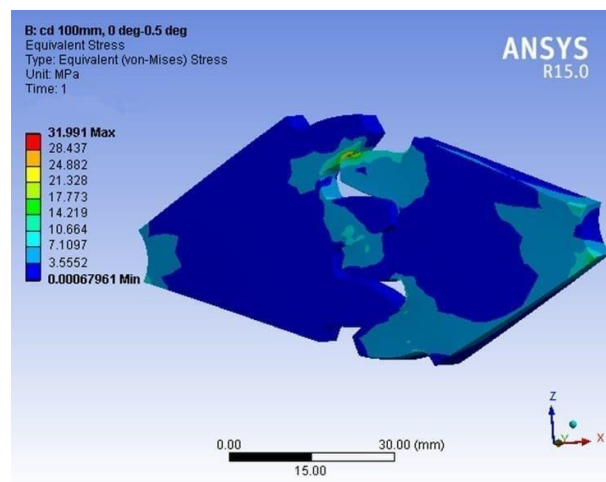


Fig.8.1. b gear 1 at 0° , gear 2 axis at 0.5°

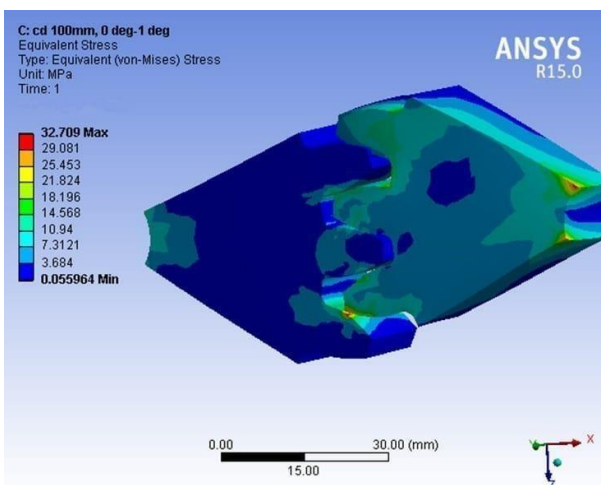


Fig.8.1.c gear 1 axis at 0° , gear 2 axis at 1°

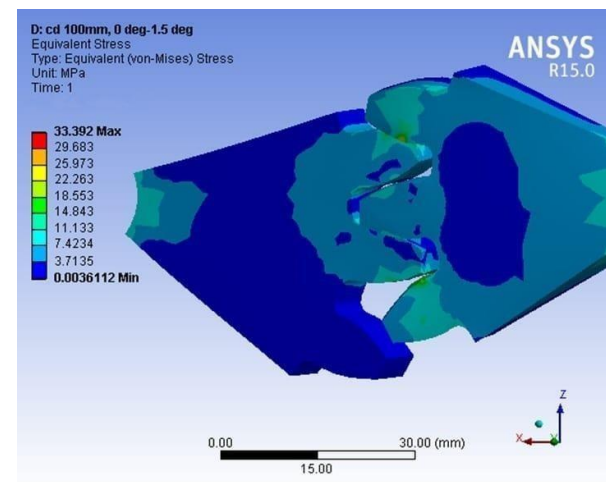


Fig.8.1.d gear 1 axis at 0° , gear 2 axis at 1.5°

8.2 Results of von Mises stresses when central distance is 100.2mm :

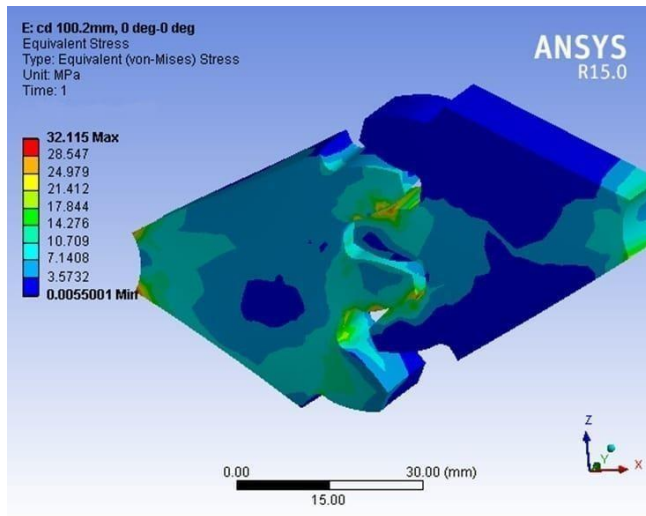


Fig.8.2. a gear 1 axis at 0° , gear 2 axis at 0°

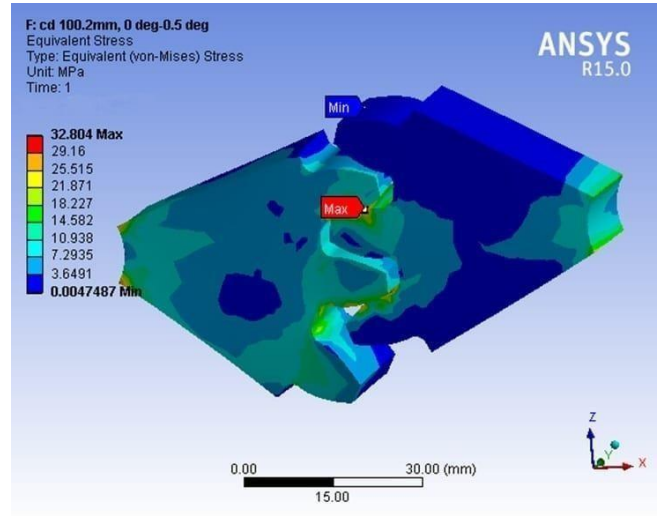


Fig.8.2. b gear 1 axis at 0° , gear 2 axis at 0.5°

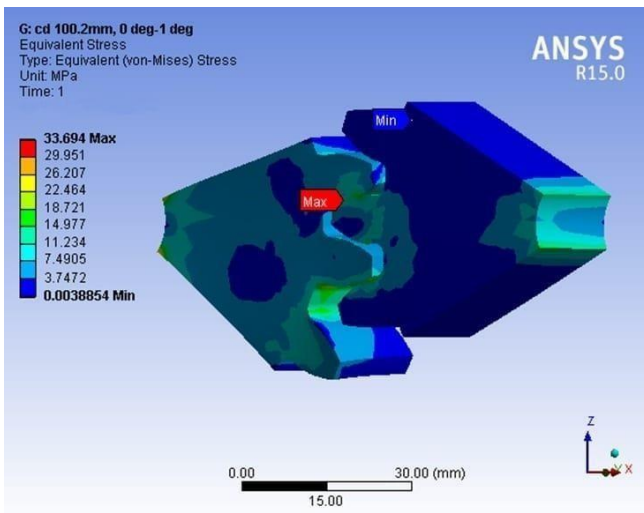


Fig.8.2.c gear 1 axis at 0° , gear 2 axis at 1°

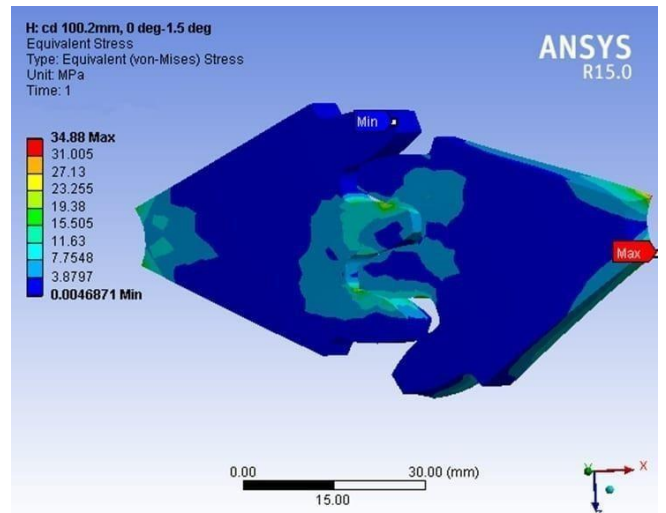


Fig.8.2. d gear 1 axis at 0° , gear 2 axis at 1.5°

8.3 Results of von Mises stresses when central distance is 100.4mm :

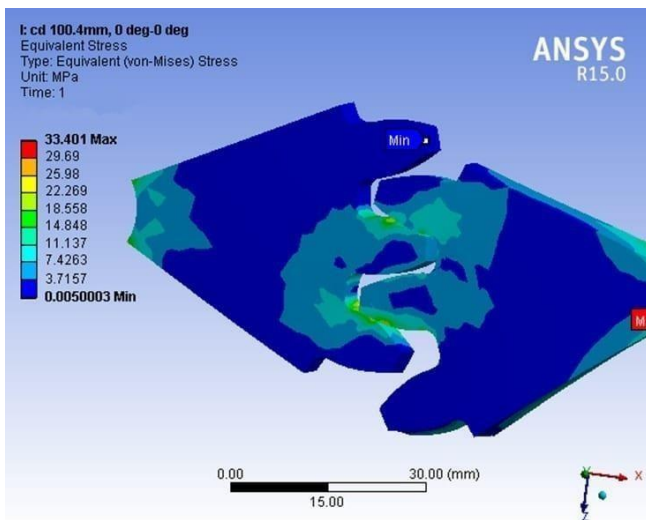


Fig.8.3. a gear 1 axis at 0°, gear 2 axis at 0°

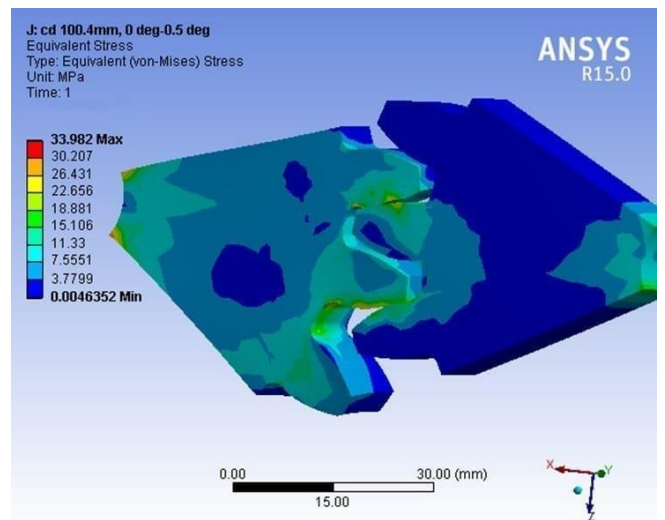


Fig.8.3. b gear 1 axis at 0°, gear 2 axis at 0.5°

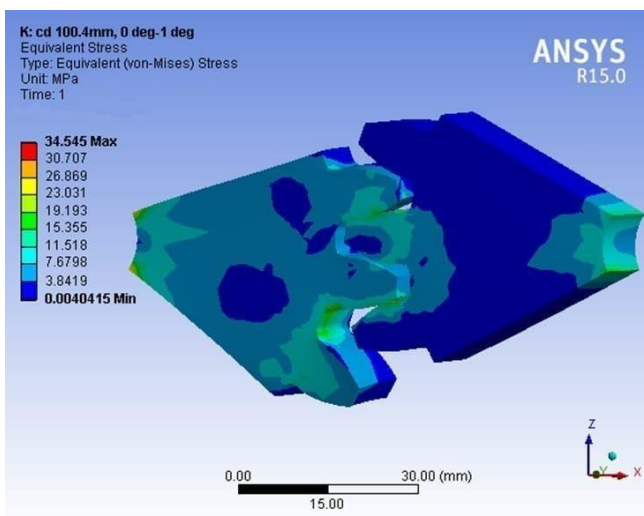


Fig.8.3.c gear 1 axis at 0°, gear 2 axis at 1°

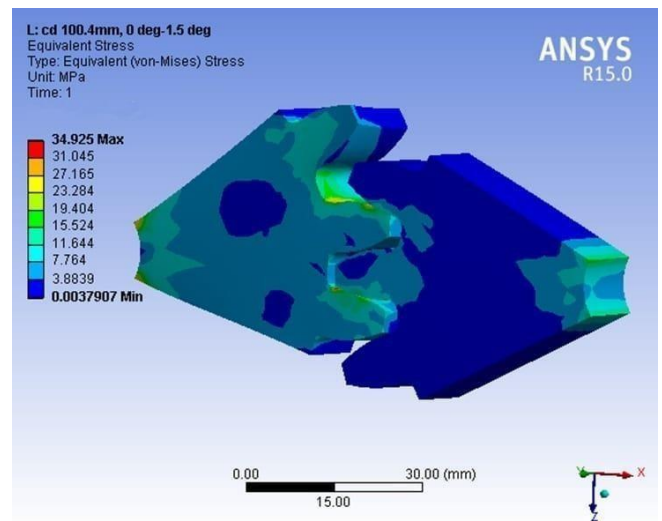


Fig.8.3. d gear 1 axis at 0°, gear 2 axis at 1.5°

The von Mises stresses of CAST NYLON shows an increasing tendency with increase in inclination of the axes.

8.4 Results of von Mises stresses when central distance is 100.00mm :

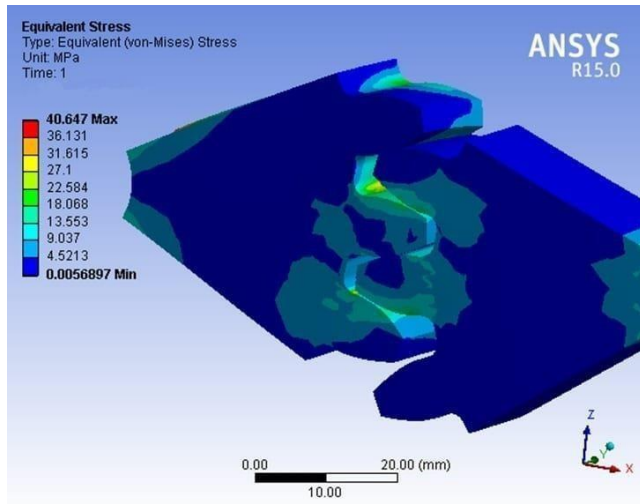


Fig.8.4.a gear 1 axis at 0° , gear 2 axis at 0°

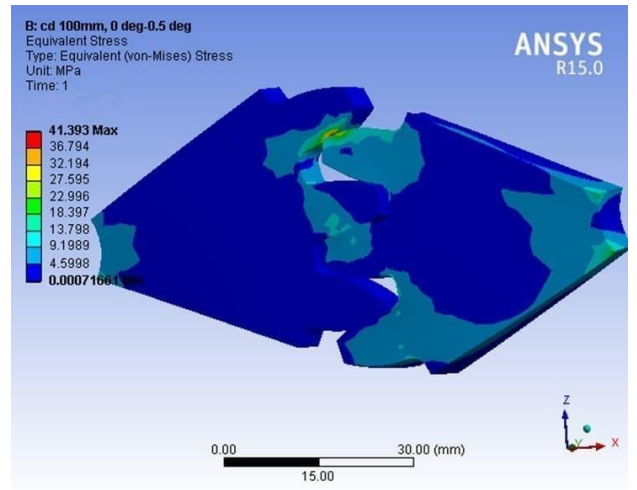


Fig.8.4.b gear 1 axis at 0° , gear 2 axis at 0.5°

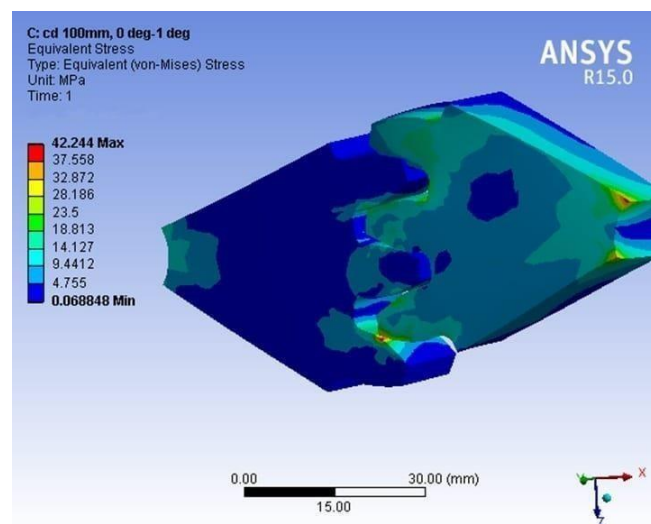


Fig.8.4.c gear 1 axis at 0° , gear 2 axis at 1°

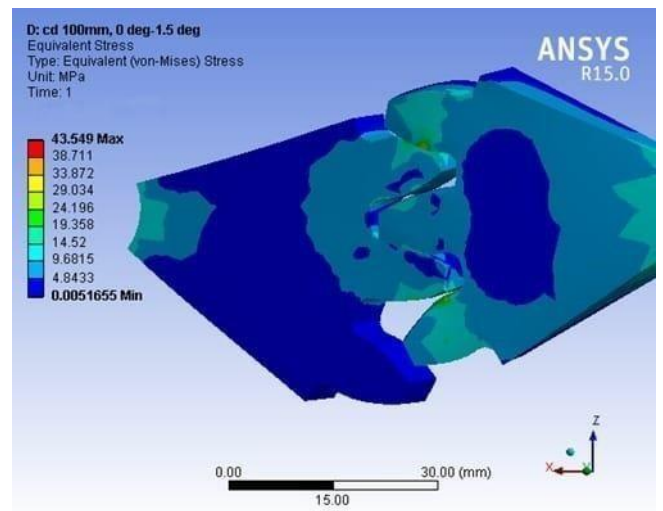


Fig.8.4.d gear 1 axis at 0° , gear 2 axis at 1.5°

8.5 Results of von Mises stresses when central distance is 100.2mm:

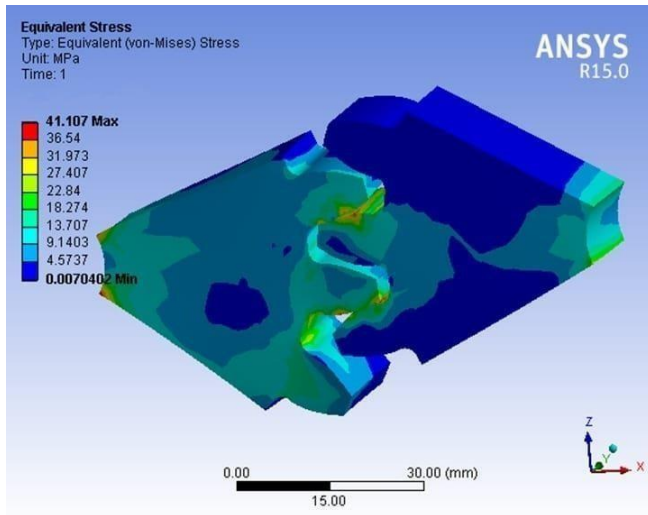


Fig.8.5.a gear 1 axis at 0° , gear 2 axis at 0°

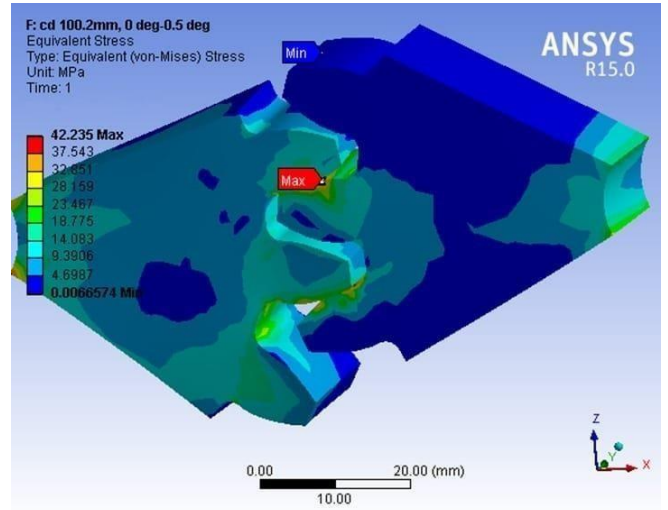


Fig.8.5.b gear 1 axis at 0° , gear 2 axis at 0.5°

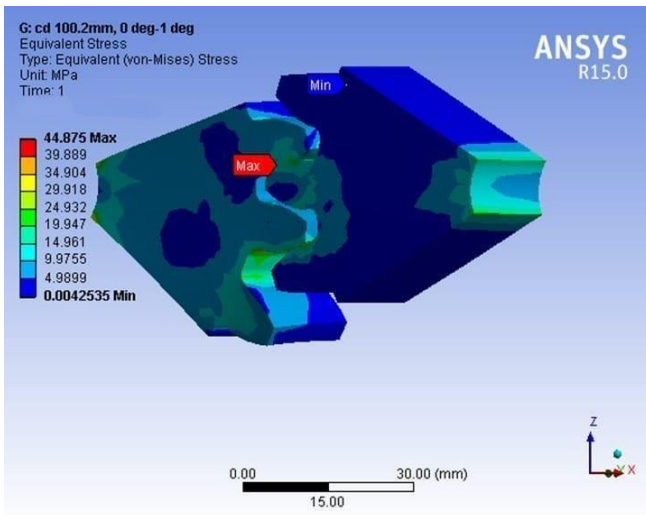


Fig.8.5.c gear 1 axis at 0° , gear 2 axis at 1°

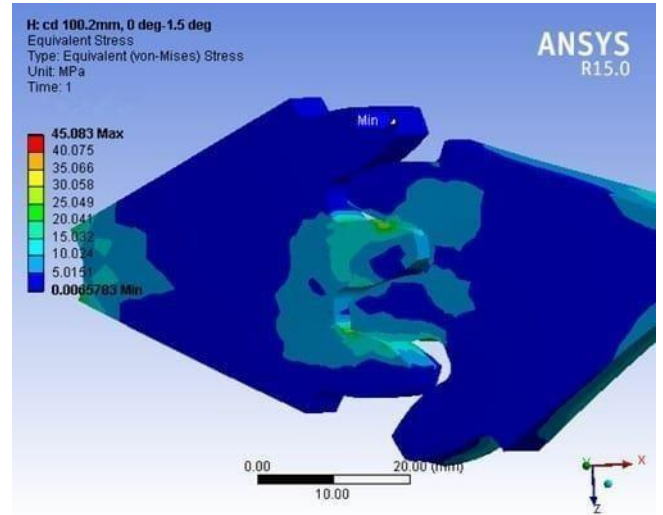


Fig.8.5. d gear 1 axis at 0° , gear 2 axis at 1.5°

8.6 Results of von Mises stresses when central distance is 100.4mm :

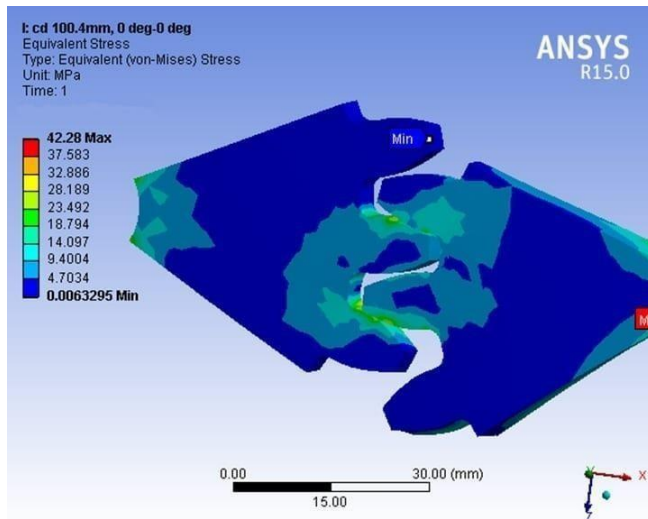


Fig.8.6. a gear 1 axis at 0° , gear 2 axis at 0°

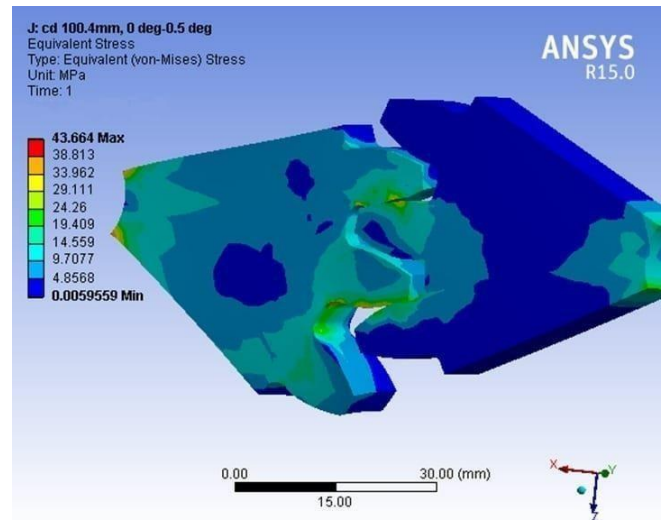


Fig.8.6. b gear 1 axis at 0° , gear 2 axis at 0.5°

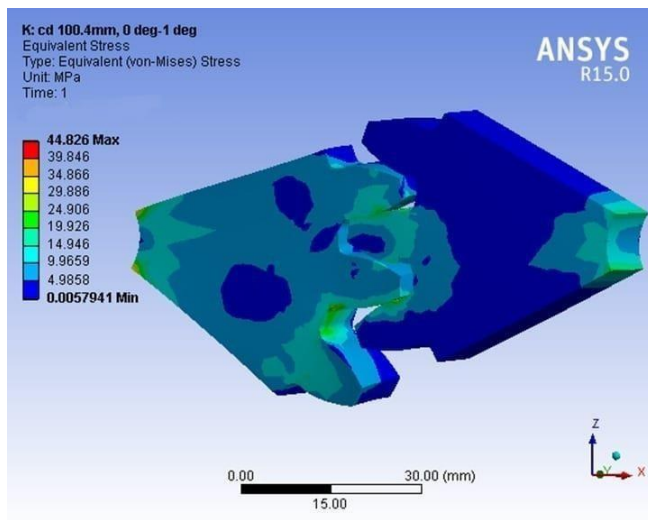


Fig.8.6.c gear 1 axis at 0° , gear 2 axis at 1°

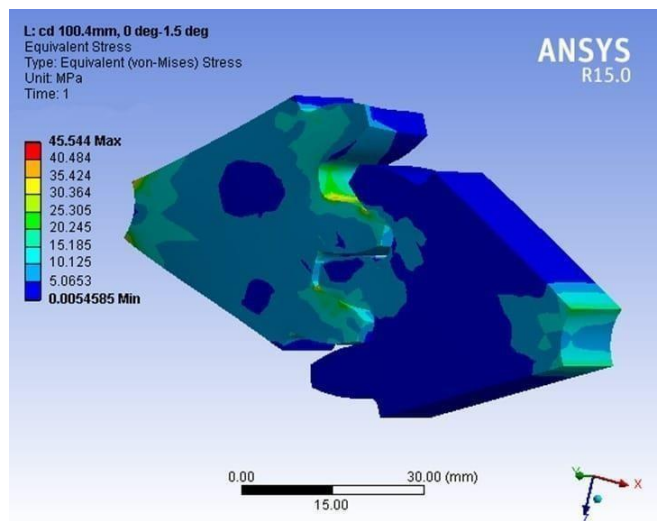


Fig.8.6. d gear 1 axis at 0° , gear 2 axis at 1.5°

CHAPTER-9

RESULTS SUMMARY

von Mises stresses of a pair of mating involute Spur gears are calculated for different sets of inclined axes. Those involute Spur Gears are drawn in SOLIDWORKS software. Contact stresses obtained in Analytical calculations and analysis in ANSYS for different sets of inclined axes plotted below.

TABLE: 9.1 RESULTS OF VON MISES STRESSES (MPa) IN ACETAL COPOLYMER FOR ALL PAIRS:

Central distance	100.00 mm	100.2 mm	100.4 mm
Axis inclination	-	-	-
0°	31.27	32.16	33.4
0.5°	31.99	32.8	33.98
1.0°	32.71	33.69	34.54
1.5°	33.39	34.88	34.925

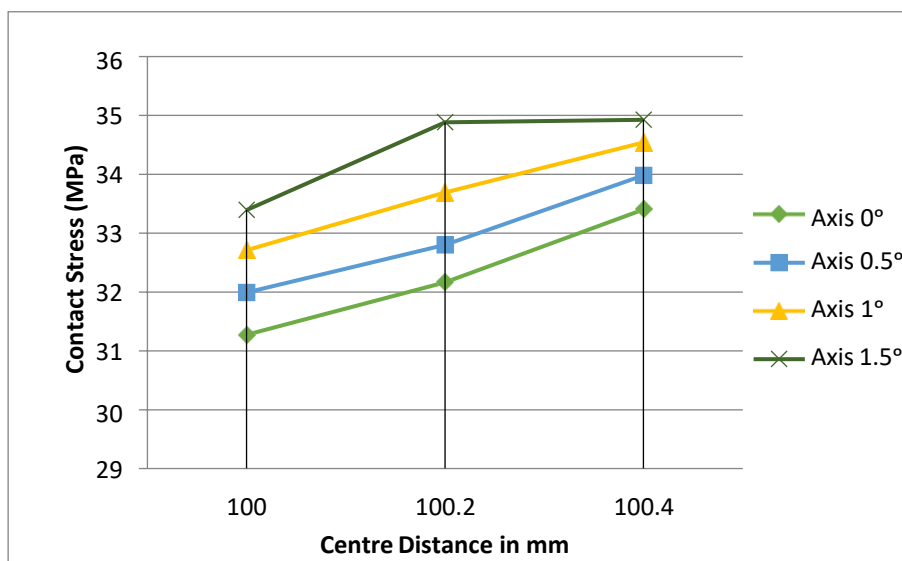
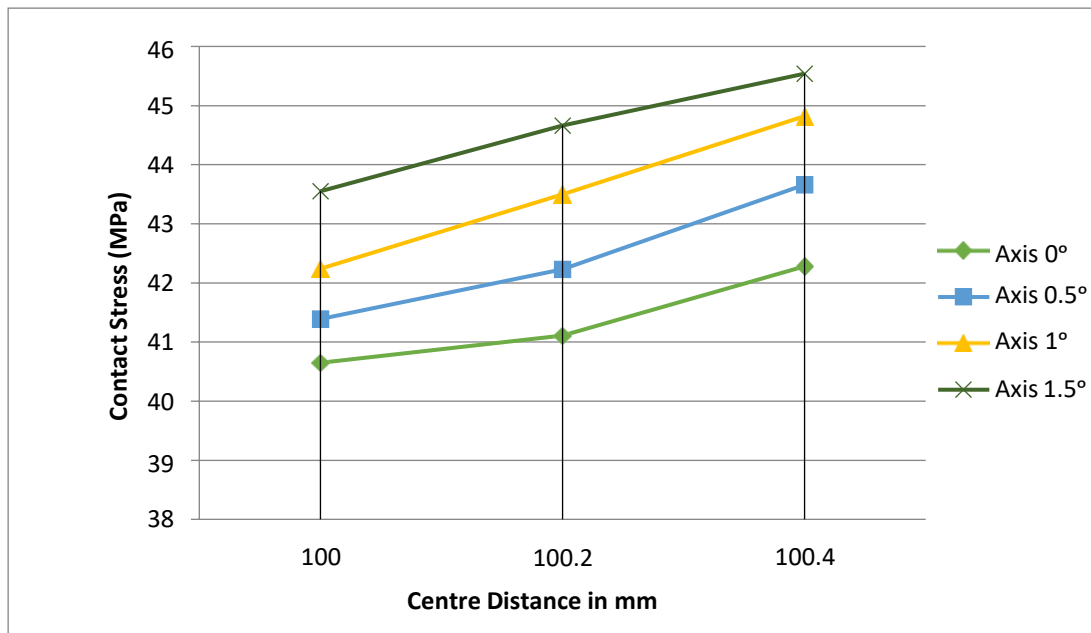


TABLE: 9.2 RESULTS OF VON MISES STRESSES (M Pa) IN FOR ALL PAIRS:

Central distance	100.00 mm	100.2 mm	100.4 mm
Axis inclination	-	-	-
0°	40.65	41.11	42.28
0.5°	41.39	42.23	43.66
1.0°	42.24	43.5	44.82
1.5°	43.55	44.66	45.54



CHAPTER-10

CONCLUSIONS

The present work deals with evaluating the contact stresses in a pair of spur gears of given parameters with varying inclination of axes of the gear shafts and central distance. During the operation of the gears, there is always a possibility of wear and tear in the bearings supporting the shafts which can probably result in misalignment of shafts. It is therefore necessary to study the effect of misalignment on the working of gears. This study was carried out for two different types of composites a) Graphene reinforced Acetol co-polymer and b) Cast nylon using ANSYS software. The following conclusions were drawn from the study:

(1) The contact stresses were found to be increasing with increase in misalignment of the shafts. The minimum was observed when the axes are parallel at theoretical central distance. This can be attributed to the reason that, with an increase in misalignment, the contact area decreases and therefore the stresses increase.

(2) The contact stresses were also determined theoretically for the ideal case by using the semi-empirical equations given by Hertz and AGMA. The contact stresses determined from the FEM analysis were close to the stresses given by hertz.

(3) The contact stresses were higher in the CAST NYLON gear when compared with the contact stresses developed in the ACETAL COPOLYMER gear for all misalignments. The variation was in the range of 27% to 30%.

FUTURE SCOPE

Research on gears was being done from a long time and also it goes on in future as the gears are one of the best mechanical components to transmit motion and power. Research work on gears is much diversified; people can research on different aspects which would affect the efficiency and the working of the machine. The future scope of the present work can be vividly explained as below:

- 1) In the present work, we limited our work to the inclinations in one plane; it can be further developed to the inclinations along different planes.
- 2) The gear can be designed according to the requirements of strength and the stresses will change according to the strength and workmanship of gears.

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