Contact Stress Analysis of Involute Spur Gear with varying Centre to Centre Distance

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Abstract— **Contact stress analysis of gear drives has been a subject of intensive research since last few years. Contact stresses were first analyzed by Hertz to develop analytical correlation 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 present attempt contact stresses in spur gear pairs have been taken for study. The analysis is conducted using Ansys software. The variation of contact stresses with varying centre to centre distance is the main objective of this work. The centre to centre distance can change due to shaft deflection or improper mounting for that the analysis is extended for three different types of materials and their combination. It is observed that the contact stresses do vary with centre to centre distance and the material of construction. The stresses are maximum for Grey Cast Iron and minimum for Structural Steel. The stresses show a decreasing trend with decrease in centre to centre distance and approach the minimum value when centre to centre is equal to the pitch circle radii. A further decrease in centre to centre distance exhibits an increasing trend in contact stresses. However it is further observed that the stresses are minimum when the centre to centre distance takes the minimum most possible value. Any further decrease in centre to centre distance will lead to underscoring.**

IndexTerms— Contact stress analysis, Gears, Friction, Visual Basics software, Ansys software, Structural Steel, centre distance

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I. 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 behaviour when gears are in mesh is extremely important if one wants to perform system monitoring and control of the gear transmission system. Although there are 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.

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.

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 are141/2, 20, and25 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.

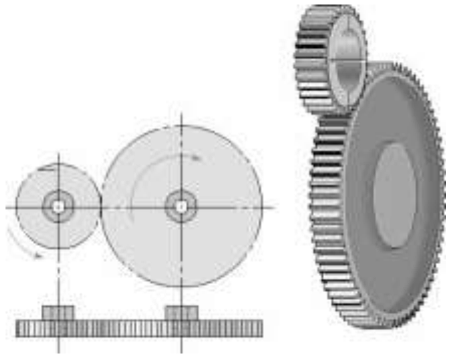


Fig.1.1: Spur Gear

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.

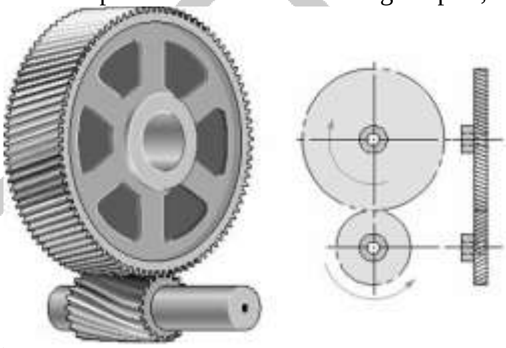


Fig.1.2: Helical Gear

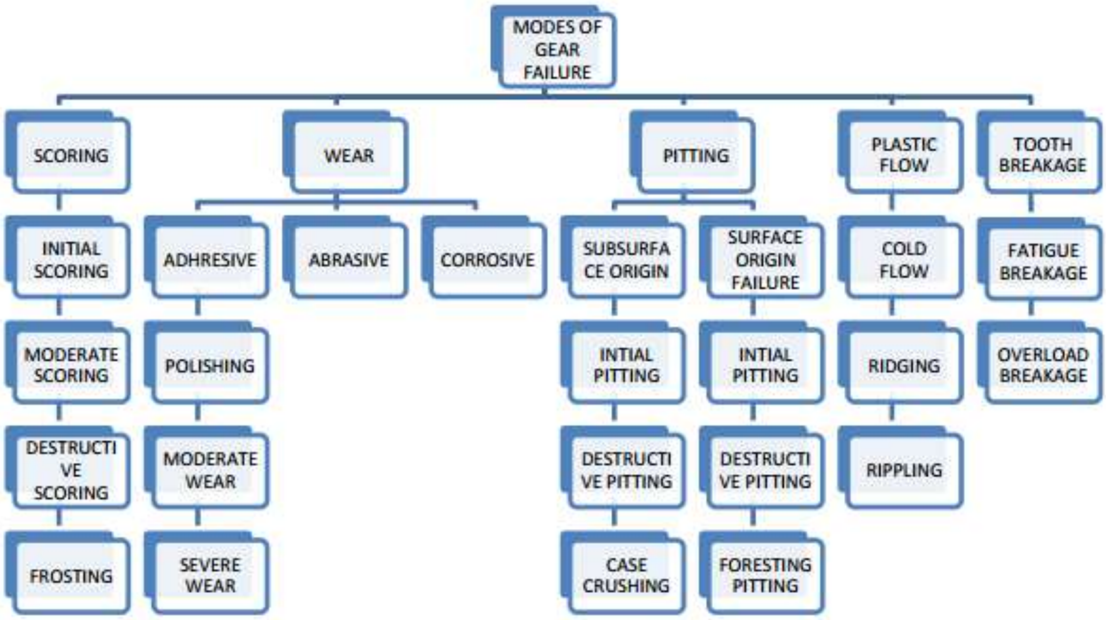
Bevel gears: Its teeth are formed on conical surfaces and unlike spur and helical gears, bevel gears are used for transmitting motion between intersecting shafts not parallel shafts. There are different types of bevel gears, but all of them establish thrust, radial, and tangential loads on their support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

Fig.1.3: Bevel Gear

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-hander 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

Modes of Gear failures :



II. LITERATURE REVIEW

1. ***Santosh S Patil, Saravanan Karuppanan, Ivana Atanasovska, Azmi Abdul Wahab:*** In this paper an attempt has been made to study the contact stresses among the helical gear pairs, under static conditions, by using a 3D finite element method. The helical gear pairs on which the analysis was carried out were 01, 51, 151, 251 helical gear sets. The effect of friction was varied at the point of contact which made the problem nonlinear and complicated. To simplify the problem, a range of average static coefficients of friction, from 0 to 0.3, has been considered as the scope of study. The variation of contact stresses with helix angle and also with friction coefficients has been discussed. The commercial finite element software used was ANSYS and the results were compared with analytical calculations.
2. ***Putti Srinivasa Rao, Nadipalli Sriraj, Mohammad Farookh:*** 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 programme 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. ***Ivana Atanasovska, Vera Nikolic-Stanojlovic, Dejan Dimitrijevic, Dejan Momcilovic***: This paper tells us that one of the main goals during load calculations and load capacity check for gears is determination of deformation and stress state in teeth contact zones and teeth fillets. This paper describes development of the finite element model for simultaneously monitoring the deformation and stress state of teeth flanks, teeth fillets and parts of helical gears during the tooth pair meshing period. The paper also describes the Finite Element Method simulation of contact conditions for helical gears teeth with an involute profile. A suitable analysis is performed in order to select a meshed gears model which is sufficiently economic and in same time sufficiently geometrically accurate. The special algorithm for the tooth involute profile drawing is developed and built in currently available software for Finite Element Analysis to assure drawing of real flanks contact geometry.
4. ***Zheng Li and Ken Mao:*** The present paper concentrates on the investigations regarding the situations of frictional shear stress of gear teeth and the relevant frictional effects on bending stresses and transmission error in gear meshing. Sliding friction is one of the major reasons causing gear failure and vibration; the adequate consideration of frictional effects is essential for understanding gear contact behaviour accurately. An analysis of tooth frictional effect on gear performance in spur gear is presented using finite element method. Nonlinear finite element model for gear tooth contact with rolling/sliding is then developed. The contact zones for multiple tooth pairs are identified and the associated integration situation is derived. The illustrated bending stress and transmission error results with static and dynamic boundary conditions indicate the significant effects due to the sliding friction between the surfaces of contacted gear teeth, and the friction effect cannot be ignored. To understand the particular static and dynamic frictional effects on gear tooth contact analysis, some significant phenomena of gained results will also be discussed. The potentially significant contribution of tooth frictional shear stress is presented, particularly in the case of gear tooth contact analysis with both static and dynamic boundary conditions.
5. ***S.Sai Anusha, P.Satish Reddy, P.Bhaskar, M.Manoj:*** This paper presents that Gears are one of the most critical components in mechanical power transmission systems. The gears are generally used to transmit power and torque. The efficiency of power transmission is very high when compared to other kind of transmission. In the gear design the bending stress and surface strength of the gear tooth are considered to be one of the main contributors for failure of the gears in gear set. The analysis of stresses has become popular as an area of research on gears to minimize and reduce the failures. The present investigation is carried out to make use of helical gear, by analyzing the contact stresses for different Pressure angles

(14.5˚,16˚,18˚,20˚) Helix angles(15˚,20˚,25˚,30˚) and (80mm,90mm, 100mm,110mm,120mm) Face width. A Three-dimensional solid model is generated by Pro-E that which is powerful and modern solid modeling software .The numerical solution is done by Ansys, which is a finite element analysis package. The analytical approach is based on contact stress equation, to determine the contact stresses between two mating gears. The results obtained from Ansys values are compared with theoretical values are in close agreement. The present analysis is useful in quantifying the above said parameters that helps in safe and efficient design of the helical gear.

6. ***Dr M S Murthy, Pushpendra kumar Mishra:*** This paper presents the stress analysis of helical gear. Gears are the most important members of mechanical power transmission systems. For power transmission helical gears have become the subject of attention. The main factors responsible for the failure of a gear set are bending stress and surface strength of a gear tooth..Therefore stress analysis becomes an important area of research which deals with minimization or reduction of the stresses and also with optimal design of gears. This paper presents a detailed study of different techniques proposed and used by various researchers to optimize and to calculate the stresses involved in the helical gear design. Several three dimensional solid models of gears of different specification have been developed by various researchers for analysis. They then used various analysis tools like Ansys, Computer aided FEM; Pro-e software etc for analysis. In this work various parameters that can affect the gear tooth, i.e. variation in face width, helix angle etc. for the complex design problems is also discussed. After going through the literature it is suggested that Matlab Simulink could be used as one of the tools to effectively solve the problems which are taken up as future work.
7. ***Mr. Kishor N. Naik, Prof. Dhananjay Dolas:*** Gears have wide variety of application. Their application varies from watches to very large mechanical units like the lifting devices and automotives. Gear generally fails when the working stress exceeds the maximum permissible stress. The main objective of this paper is to analyze the bending stresses occur on the gear tooth profile of gear used in gear box of special purpose machine also effect on bending stress by variation of the gear parameters. Face width and root radius are taken gear parameters, how stress redistribution are taken place by varying this parameter studied. The stresses are calculated with the help of the FEA this result are compared with the stresses calculated by Lewis equation. For this work parametric modelling is done using Pro-e WF 5.0 and for analysis ANSYS 12.0 workbench is used. This work helpful to conclude effect of bending stress on gear tooth profile by variation of gear parameters also give the comparison of FEM method with analytical calculation.

III. CONTACT STRESS

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 matting gears are mostly important for the gear design. Gearing is one of the most critical components in mechanical power transmission systems.

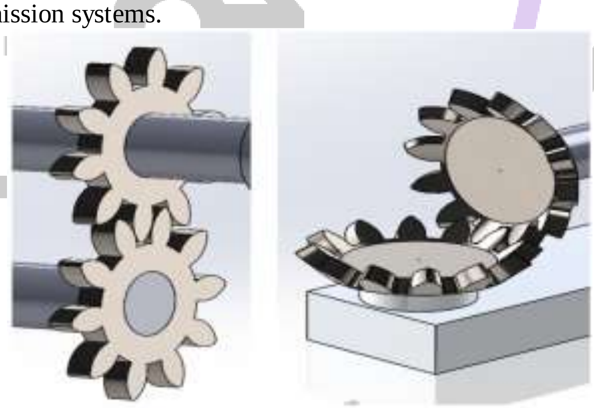


Fig.3.1: Gear mating

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.

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 behaviour of materials modelled as acontinuous 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.

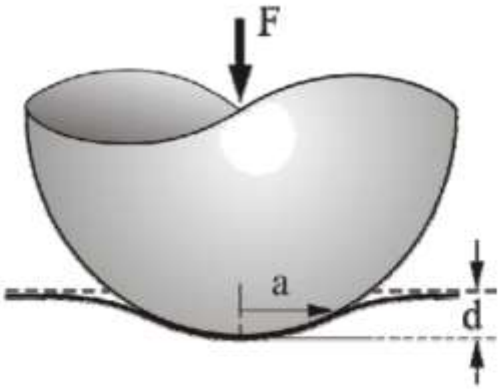


Fig.3.2: Depiction of contact area under applied load

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

a) Surfaces are continuous and non-conforming (i.e. initial contact is a point or a line)
b) Strains are small
c) Solids are elastic
d) Surfaces are frictionless

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.

$$\sigma_c = C_p \left[\frac{F_t}{dLI}\right]^{\left(\frac{1}{2}\right)}$$

$$C_p = \left[\frac{1}{\pi\left(\frac{1-v_1^2}{E_1}+\frac{1-v_2^2}{E_2}\right)}\right]^{\left(\frac{1}{2}\right)}$$

$$F_t = F_n \times \cos \emptyset$$

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

The AGMA Contact stress equation is given by,

$$\sigma_H = C_P\sqrt{\frac{F_t}{bdI}\left(\frac{\cos\beta}{0.95CR}\right)K_V K_O(0.93K_M)}$$

Analytical Contact Stress Of Involute Spur Gear

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 1: Specifications of Gear

S. No	Particulars	PINION	GEAR
1	Module(mm)	4.5	4.5
2	No. of Teeth	20	20
3	Pitch circle dia.(mm)	90	90

The AGMA Contact stress equation is given by,

$$\sigma_H = C_P\sqrt{\frac{F_t}{bdI}\left(\frac{\cos\beta}{0.95CR}\right)K_V K_O(0.93K_M)}$$

Bending Stress is given by,

$$\sigma_b = \frac{F_t}{bmJ} k_v k_o (0.93 k_m)$$

Where J is Geometry Factor

$$\sigma_b = w^t k_o k_v k_s \frac{1}{bm_t} \frac{k_H k_B}{Y_J}$$

Hertz Contact Equation is given by,

$$\sigma_c = C_p \sqrt{w^t k_o k_v k_s \frac{k_m}{d_p F} \frac{c_f}{I}}$$

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.

IV. VISUAL BASICS

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 theuser.
- Visual Basic is **event-driven**, meaning code remains idle until called upon torespond 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.

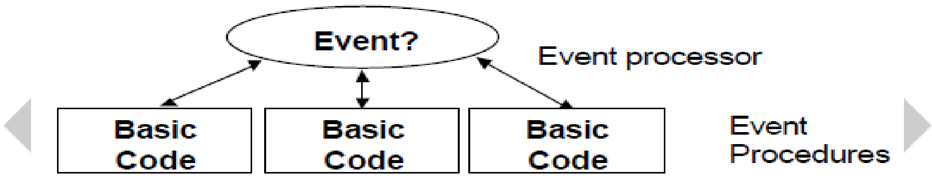


Fig. 4.1: Event Processor

Some Features of Visual Basic

- 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
- Package & Deployment Wizard makes distributing your applications simple

The **Main Window** consists of the title bar, menu bar, and toolbar. Thetitle bar indicates the project name, the current Visual Basic operatingmode, and the current form. The menu bar has drop-down menus fromwhich you control the operation of the Visual Basic environment. Thetoolbar has buttons that provide shortcuts to some of the menu options.The main window also shows the location of the current form relative tothe upper left corner of the screen (measured in tips) and the widthand length of the current form.

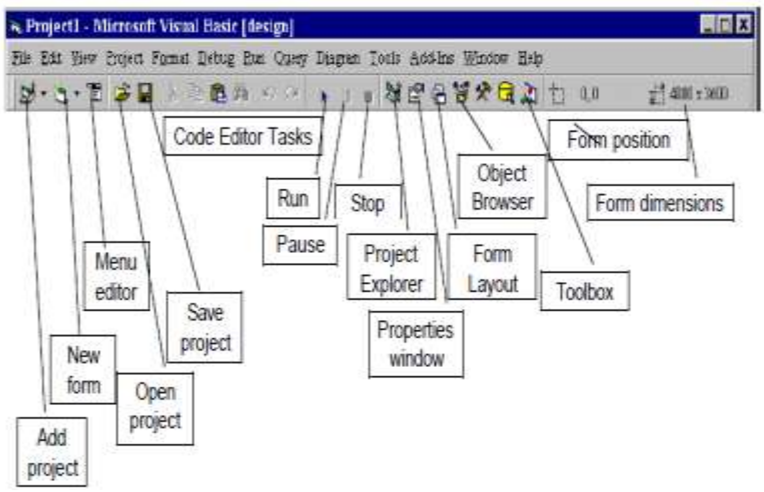


Fig. 4.2: Main window

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

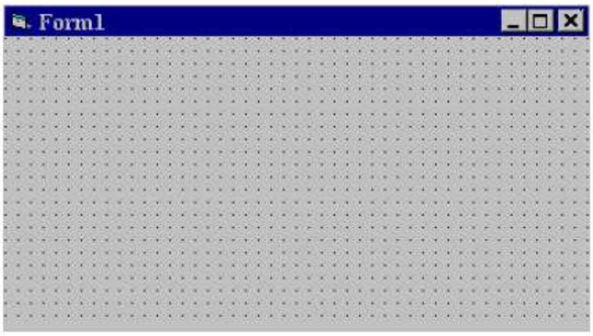


Fig. 4.3: Form window

V. SOLID WORKS

Introduction

Solid Works 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. 5.1: Solid works 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.

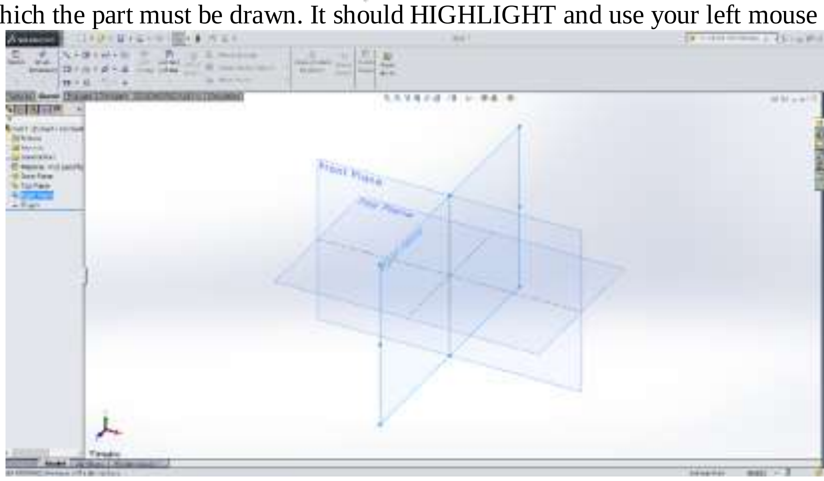


Fig.5.2: plane selection

- 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.

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.

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.

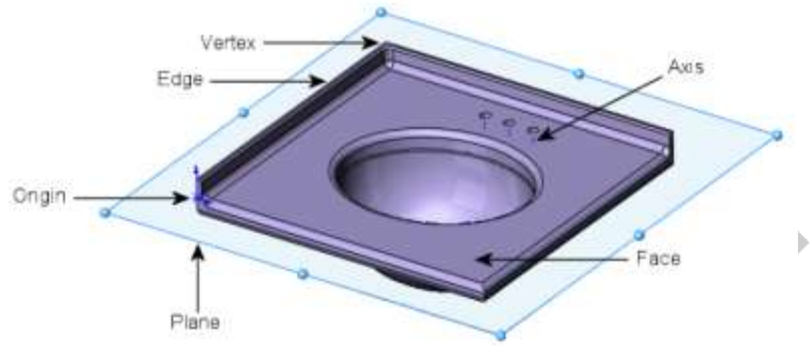
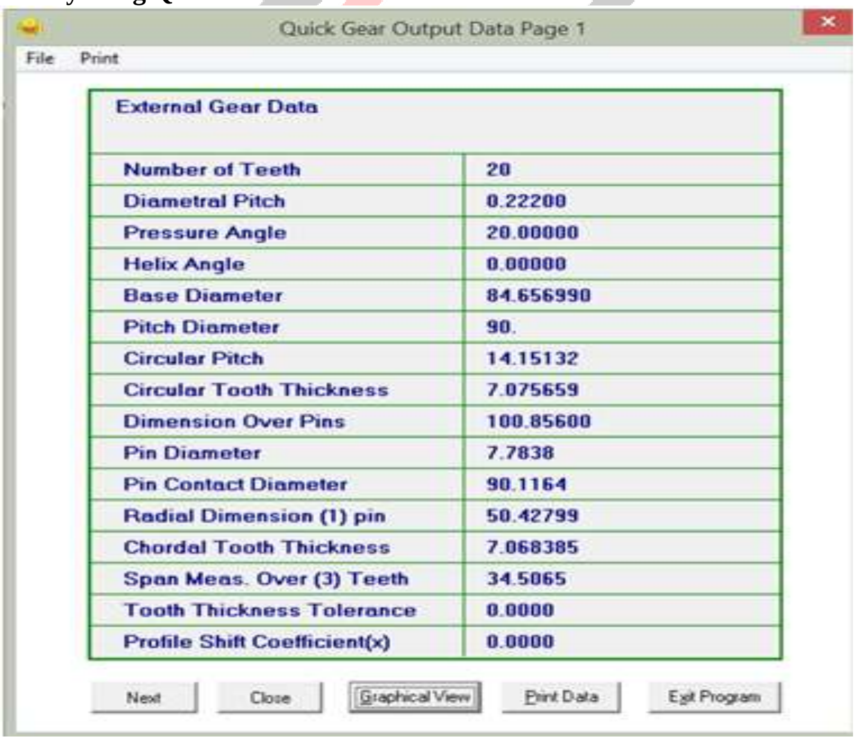


Fig.5.3: Model Terminology

Design of Gear:

Specifications of gear calculated by using Quick Gear-2 Interface



Number of Teeth	20
Diametral Pitch	0.22200
Pressure Angle	20.00000
Helix Angle	0.00000
Base Diameter	84.656990
Pitch Diameter	90.
Circular Pitch	14.15132
Circular Tooth Thickness	7.075659
Dimension Over Pins	100.85600
Pin Diameter	7.7838
Pin Contact Diameter	90.1164
Radial Dimension (1) pin	50.42799
Chordal Tooth Thickness	7.068385
Span Meas. Over (3) Teeth	34.5065
Tooth Thickness Tolerance	0.0000
Profile Shift Coefficient(x)	0.0000

Fig. 5.4: Trial Data used for gear construction

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 90mm.

TABLE 2: Data used for Gear Construction

Parameter	Spur gear	
	Pinion	Gear
No. of teeth	20	20
Normal module	4.5	4.5
Normal pressure (deg)	20	20
Helix angle (deg)	0	0
Pitch Diameter (mm)	90	90
Face Width (mm)	20	20
Contact ratio	1	1
Torque (N-m)	302	
Power (K-w)	31.075	31.075
Speed (rpm)	1000	
Contact force (N)	6102.5	

- Using offset command and taking offset distance 4.5mm, offset the pitch circle outwards to get addendum circle.
- Using offset command and taking offset distance 5.625mm, offset the pitch circle inside to get dedendum circle.
- Draw base circle having diameter 84.5723mm.

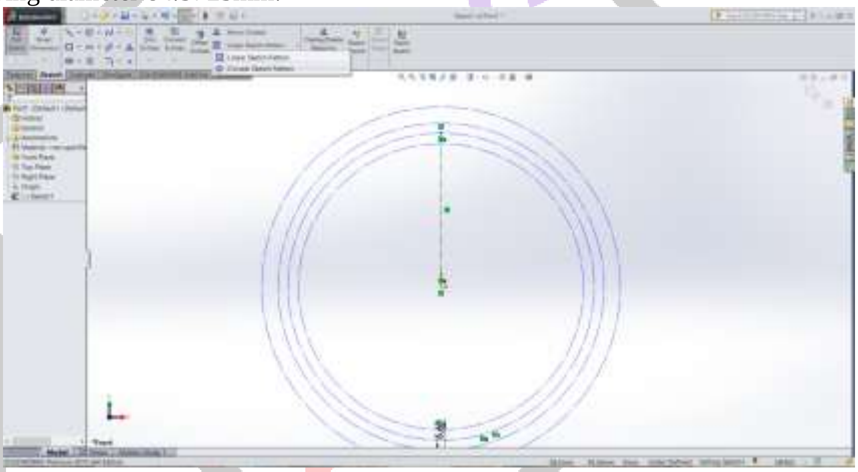


Fig. 5.5: Pitch circle and Base circle

- Divide 1/4th of the circle into 10 equal parts using circular pattern command giving angle 45^{0}and 10 number of lines.
- Draw tangents to the base circle from these projection lines leaving the first projection line and 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, trim out the extra curve exceeding the addendum circle. Hence the curve generated is the required involute curve.
- 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 7.0686mm draw another line, 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.

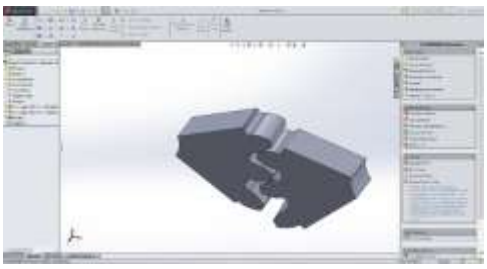


Fig. 5.6: Final Three teeth Gear Profile assembly

VI. ANSYS WORKBENCH

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 Xplorer and Design Xplorer 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, etc.?
- What to model: Part or Assembly?
- Which elements: Surface or Solid Bodies?
  - Pre-processing
- Attach the model geometry
- Define and assign material properties to parts
- Mesh the geometry
- Check the validity of the solution

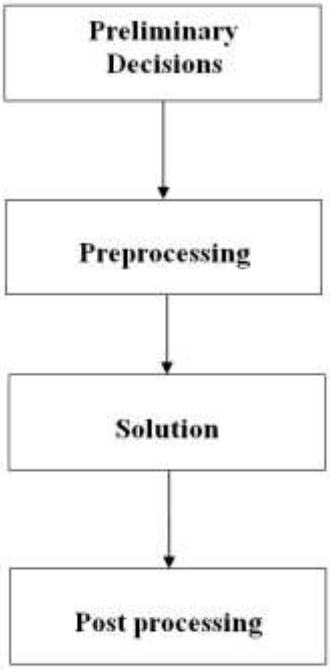


Fig. 6.1: ANSYS Procedure

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.

- **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.

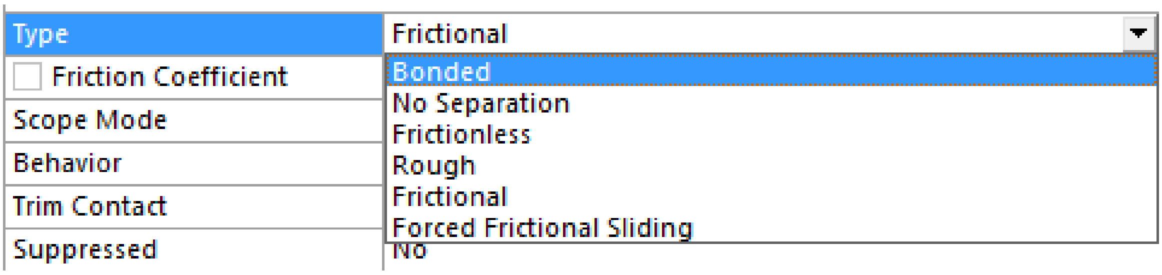


Fig. 7.6: Types of contacts in Ansys Work Bench

Table 3 Mechanical Properties of the Materials

S. No	Material	Density (g/cc)	Hardness (BHN)	Tensile Strength (MPa)	Modulus Of Elasticity (GPa)	Poisson Ratio
1	Structural Steel AISI1040 Steel	7.845	170	585	200	0.29
2	Aluminium Aluminium8090	2.54	91	340	77	0.33
3	Gray Cast Iron ASTM Class 60	7.15	302	431	152	0.29

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

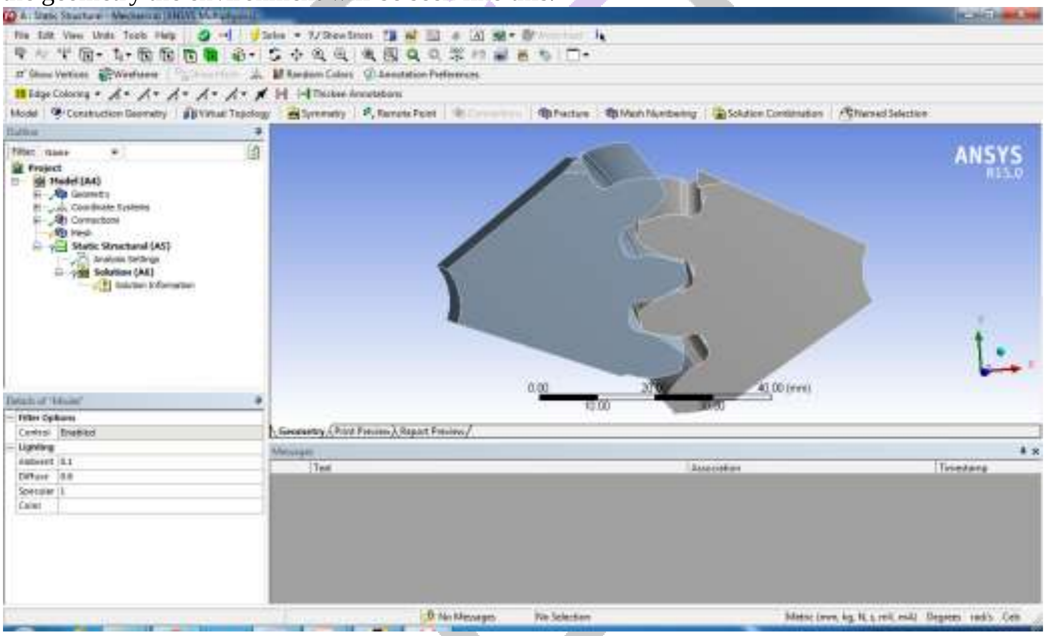


Fig. 6.2: Contact region

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

SOLID186 is a higher order 3-D 20-node solid 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 supports plasticity, hyper elasticity, creep, stress stiffening, large deflection, and large strain capabilities. It also has mixed formulation capability for simulating deformations of nearly incompressible elasto plastic materials, and fully incompressible hyper elastic materials.

Table: 4. No of nodes and elements

Centre Distance(mm)	Nodes	Elements
91.5	11745	2261
91	11745	2261
90.5	11745	2261
90	11745	2261
89.5	11745	2261
89	11745	2261

VII. ANSYS RESULTS

The results below represent the von Mises Stresses acting on involute spur gears with pressure angle 20°and varying Centre distances of 91.5, 91, 90.5, 90, 89.5, 89(mm) for Structural Steel, Aluminum Alloy and Gray Cast Iron.

The results summarize that the von Mises stresses show a declining variation with decrease in centre to centre distance

Results of von Mises stresses when both the Gear and Pinion are Structural Steel:

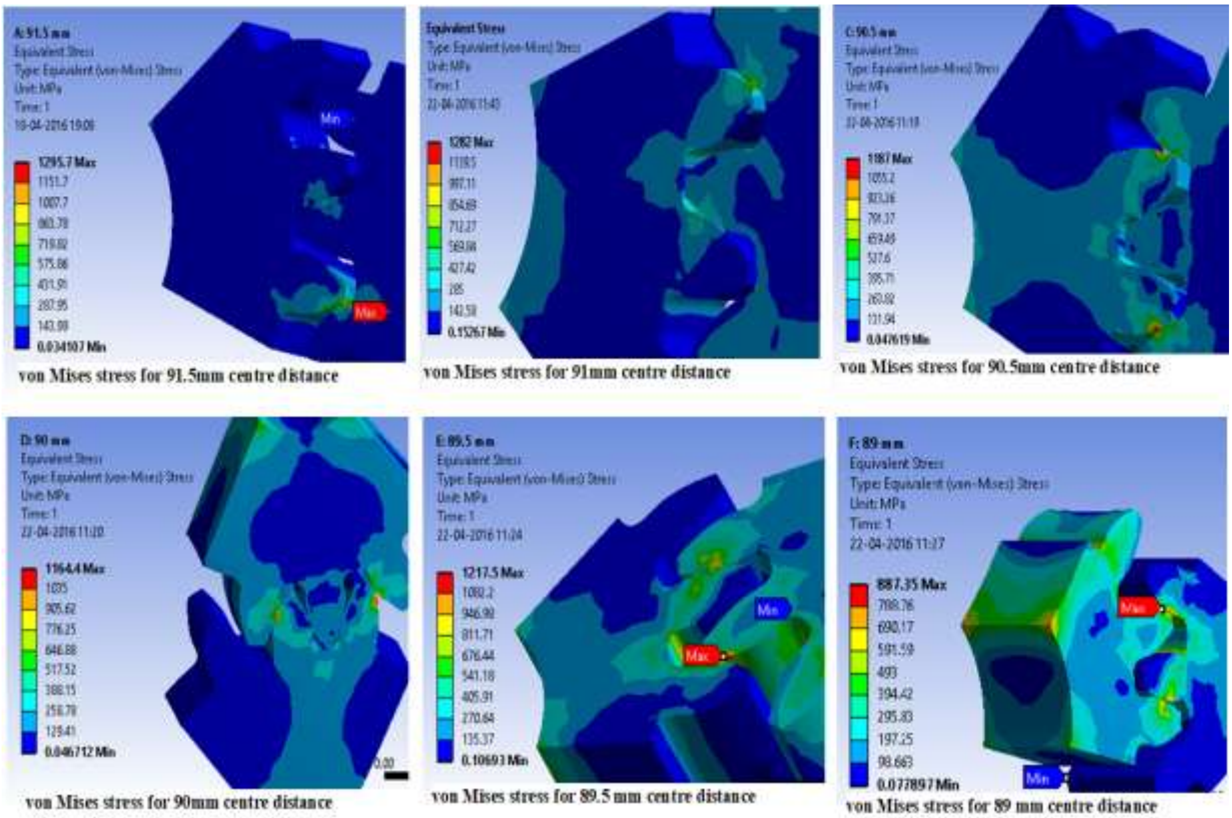


Fig. 7.1: Mises stresses when both the Gear and Pinion are Structural Steel

I. RESULT SUMMARY

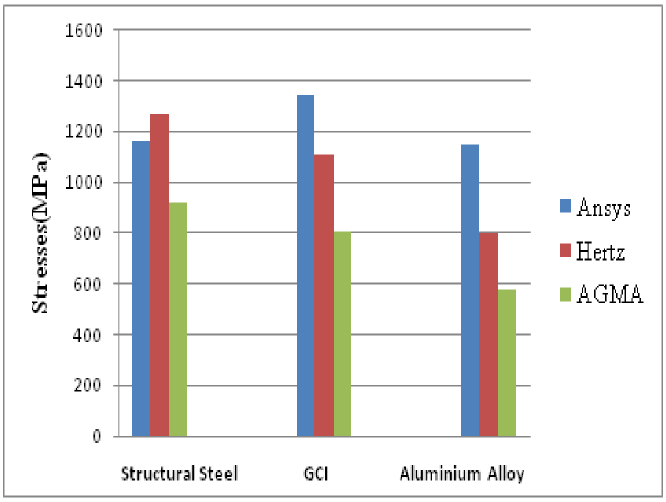
Von Mises stresses, Deformation and Contact Pressure for a pair of mating involute Spur gears are calculated for different centre distances. Those involute Spur Gears are drawn in SOLIDWORKS software. Contact stresses obtained in Analytical calculations and analysis in ANSYS for different centre distances are plotted below.

TABLE: 5 RESULTS OF VON MISES STRESSES, DEFORMATION AND CONTACT PRESSURE

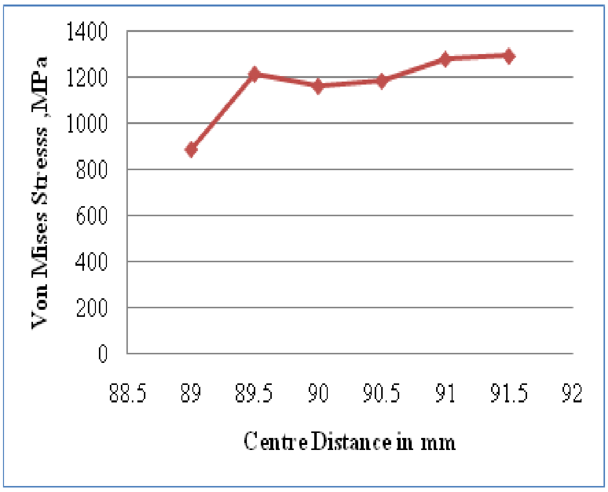
Centre Distance		G:SS P:SS	G:Al Alloy P:Al Alloy	G:GCI P:GCI	G:GCI P:SS	G:SSP: GCI	G:Al AlloyP: SS	G:SS P:Al Alloy	G:Al Alloy P:GCI	G:GCIP: Al Alloy
91.5	vonMises Stress (MPa)	1295.7	1269.8	1318.5	1308	1303	1325	674.62	1323	1263.3
	Deformation (mm)	0.1110	0.3106	0.2027	0.1635	0.149	0.226	0.1513	0.26501	0.2471
	Contact Pressure (mm)	150.64	149.64	152.76	177.68	137.7	192.8	319.71	166.5	137.66
91	Von Mises Stress (MPa)	1282	1259.9	562.31	1335	1215	1381	750.98	1350.8	1201.8
	Deformation (mm)	0.1056	0.2971	0.1923	0.1408	0.155	0.183	0.1943	0.2367	0.2504
	Contact Pressure (mm)	906.89	905.98	907.62	1021.4	803.0	1071	661.15	966.28	853.55

TABLE: 6 Results Of Contact Stresses Calculated Through Analytical Method And Using Ansys Software

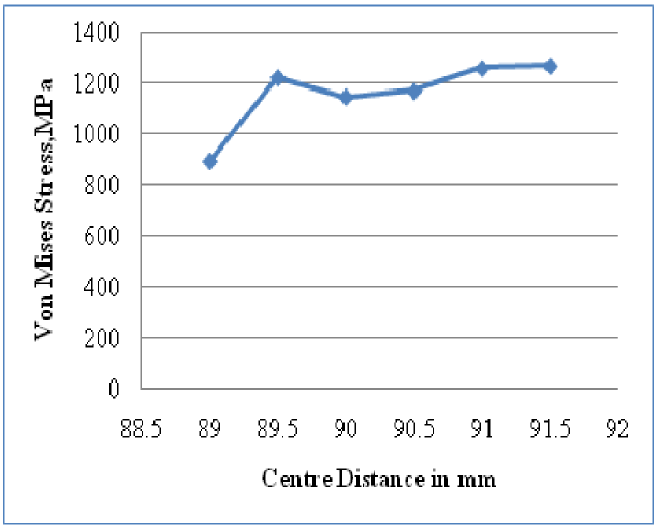
S. No	Centre distance (mm)	Stresses(M Pa)		
		Ansys	Analytical	
			Hertz	AGMA
Structural Steel AISI1040 Steel	90	1164.4	1272.14	922.01
Gray Cast Iron ASTM Class 60	90	1346.5	1109.04	803.81
Aluminium Aluminium8090	90	1148	800.25	580.01



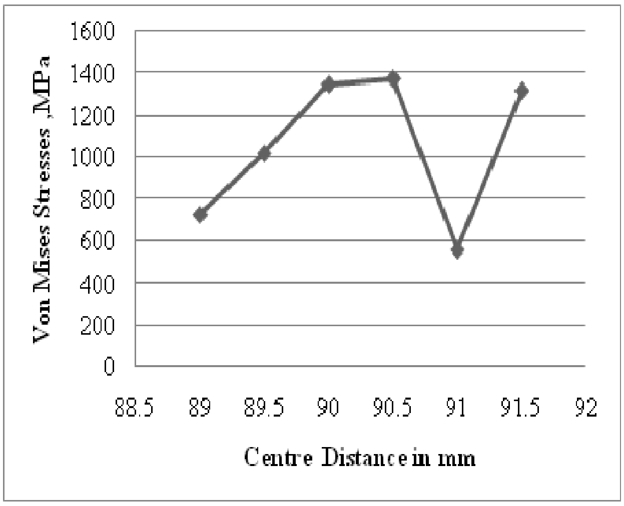
von Mises stress, Hertz stress and AGMA stress for different materials at 90mm centre distance



Analytical von Mises stress for different centre distances Gear sets of Structural Steel



Analytical von Mises stress for different centre distances Gear sets of Aluminium Alloy



Analytical von Mises stress for different centre distances Gear sets of Gray Cast Iron

Fig 7. 1Results Of Contact Stresses Calculated Using Ansys Software

VIII. CONCLUSION

The present work deals with evaluating the contact stresses in a pair of spur gears of given parameters further the effect of centre to centre distance the contact stresses where also analyzed. The work further includes study in the variation of contact stresses with different materials of construction. The following conclusions can be drawn from the study.

(1) The contact stresses are found to be maximum in Grey Cast Iron and minimum in Structural Steel. The study was also conducted for different combination of materials for gear and pinion.
(2) The variation of contact stresses with change in materials is not very marked.
(3) The effect of centre to centre distances on contact stresses shows an interesting trend. The stresses show a declining variation with decrease in centre to centre distance.

(4) The minimum stresses are observed when centre to centre distance is equal to the sum of the pitch circle radii.
(5) This can be attributed to the reasons that, with an increase in contact area with shorter centre to centre distances.
(6) The contact stresses were further evaluated at the minimum most possible centre to centre distance. It is obvious that the stresses take the least value for this distance.
(7) Hertz and AGMA stresses are calculated and found that Hertz contact stresses are much preferable than AGMA stresses.

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