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Arjun N Prof.
Associate Professor Department of Humanities and Sciences, Malla Reddy Institute of Engineering and Technology, Hyderabad, Telangana, arjunmeister@gmail.com

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Computational Analysis of Contact Stresses In Involute Spur Gears

Arjun. N
Department of Humanities and Sciences
Malla Reddy Institute of Engineering and Technology
Hyderabad, India
arjunmeister@gmail.com

Abstract— **The present paper investigates computational analysis of the contact stresses involved in an involute gear system using ANSYS simulation software. Gearing is one of the most critical components in mechanical power transmission systems. The computational simulation of contact stresses of two cylinders is carried out to validate the methodology with available analytical method, Hertz contact stress equation, which are originally derived for contact between two cylinders. The simulation of contact problems with ANSYS is carried out with the stiffness relationship between the two contact areas is usually established through a contact element, spring type is placed between the two contacting areas. The results of the two dimensional FEM analyses using ANSYS, a FEM solver software are presented. The results simulated are in agree with the theoretical values calculated using Hertz contact stress equation.**

Keywords- Gears, contact stress.

I. INTRODUCTION

Gearing is one of the most critical components in a mechanical power transmission system, and in most industrial rotating machinery. It is possible that gears will predominate as the most effective means of transmitting power in future machines due to their high degree of reliability and compactness. The increasing demand for quiet power transmission in machines, vehicles, elevators and generators, has created a growing demand for a more precise analysis of the characteristics of gear systems. In the automobile industry, the largest manufacturer of gears, higher reliability and lighter weight gears are necessary as lighter automobiles continue to be in demand.

Designing highly loaded spur gears for power transmission systems that are both strong and quiet requires analysis methods that can easily be implemented and also provide information on contact and bending stresses, along with transmission errors. Gears analyses in the past are performed using analytical methods, which required a number of assumptions and simplifications. In general, gear analyses are multidisciplinary, including calculations related to the tooth stresses and to tri biological failures such as like wear or scoring.

There has been a great deal of research on gear analysis, and a large body of literature on gear modeling has been published. The gear stress analysis, the transmission errors, the prediction of gear dynamic loads, gear noise, and the optimal design for gear sets are always major concerns in gear design. Errichello [12] and Ozguven and Houser[13] survey a great deal of literature on the development of a variety of simulation models for both static and dynamic analysis of different types of gears. The first study of transmission error was done by Harris [14]. He showed that the behavior of spur gears at low speeds can be summarized in a set of static transmission error curves. In later years, Mark [15] and [16] analyzed the vibratory excitation of gear systems theoretically. He derived an expression for static transmission error and used it to predict the various components of the static transmission error spectrum from a set of measurements made on a mating pair of spur gears. Kohler and Regan [17] discussed the derivation of gear transmission error from pitch error transformed to the frequency domain. Kubo et al [18] estimated the transmission error of cylindrical involute gears using a tooth contact pattern.

In this research paper, preliminary investigation of static contact stresses between two cylinder to validate the methodology that can be implemented for gears in contact resulting in complicated mesh are performed using ANSYS.

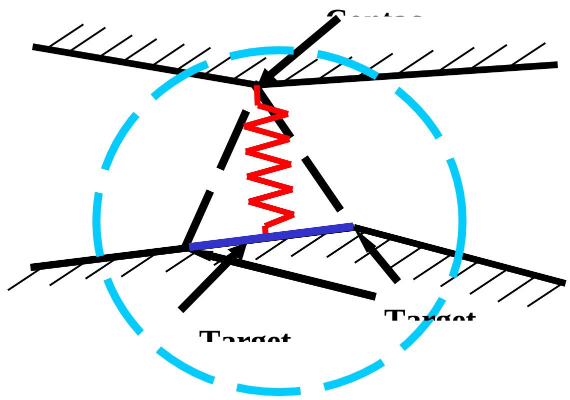
II. THEORY

Despite the importance of contact in the mechanics of solids and its engineering applications, contact effects are rarely seriously taken into account in conventional engineering analysis, because of the extreme complexity involved. Mechanical problems involving contacts are inherently nonlinear. Change in Status Nonlinearities (Contact) is considerable in the analysis of gears. Contact problems present many difficulties. First, the actual region of contact between deformable bodies in contact is not known until the solution has been obtained. Depending on the loads, materials, and boundary conditions, along with other factors, surfaces can come into and go out of contact with each other in a largely unpredictable manner. Secondly, most contact problems need to account for friction. The modeling of friction is very difficult as the friction depends on the surface smoothness, the physical and chemical properties of the material, the properties of any lubricant that might be present in the motion, and the temperature of the contacting surfaces. There are several friction laws and models to choose from, and all are nonlinear. Frictional response can be chaotic, making solution convergence difficult (ANSYS).

With the rapid development of computational mechanics, however, great progress has been made in computational analysis of the problem. Using the finite element method, many contact problems, ranging from relatively simple ones to quite complicated ones, can be solved with high accuracy. The Finite Element Method can be considered the favorite method to treat contact problems, because of its proven success in treating a wide range of engineering problem in areas of solid mechanics, fluid flow, heat transfer, and for electromagnetic field and coupled field problems.

The contact problem is addressed using a special contact element. A number of contact elements were available (two and three dimensional, spring and damper combinations). For the problem in hand, the element to be used is a two-dimensional, the three nodes, and point-to-surface contact element. In the input file, the CONTAC48 element from the ANSYS element library as the contact elements between the two contact bodies shown as Figure 3.1 is chosen. It is applicable to 2-D geometry, plane strain, plane stress, or axisymmetric situations. The area of contact between two or more bodies is generally not known in advance. It may be applied to the contact of solid bodies for static or dynamic analyses, to problems with or without friction, and to flexible-to-flexible or rigid-to-flexible body contact.

P o i n t -to-surface contact element



Specifications of spur gears used

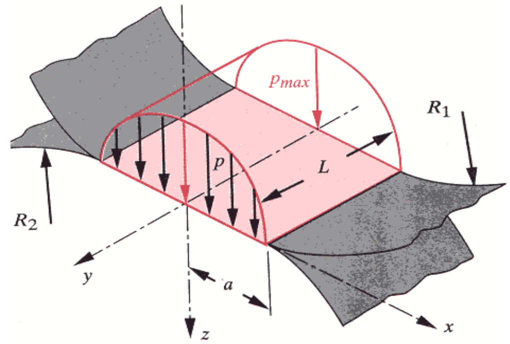
Number of teeth	25
Normal Module (M)	6 mm
Addendum Modification	0
Normal Pressure Angle	20 degrees
Face Width (mm)	0.015 M
Addendum (mm)	1.00 M
Dedendum (mm)	1.25 M

III. EASE OF USE

A. *Hertz Contact Stress Equation*

Usually, the current methods of calculating gear contact stresses use Hertz's equations, which were originally derived for contact between two cylinders. Contact stresses between two cylinders were shown in Figure 3.4. An ellipsoidal-prism pressure distribution is generated between the two contact areas.

Ellipsoidal-prism pressure distribution



From Figure 3.4 the width of the contact zone is 2a. If total contact force is F and contact pressure is p(x), there is a formula [5], which shows the relationship between the force F and the pressure p(x): $F = 2L\int_0^a p(x)dx$ (3.4)

contact width $a = \sqrt{\dfrac{2F(1-\nu_1^2)/E_1 + (1-\nu_2^2)/E_2}{\pi L(1/d_1 + 1/d_2)}}$ (3.5)

The maximum contact stress $P_{\max} = \dfrac{2F}{\pi a L}$

d_1 and d_2 represent the pinion and gear pitch diameters.

The maximum surface (Hertz) stress:

$$P_{\max} = \sigma_H 0.564\sqrt{\frac{F\left(\dfrac{1}{R_1}+\dfrac{1}{R_2}\right)}{\dfrac{(1-\nu_1^2)}{E_1}+\dfrac{(1-\nu_2^2)}{E_2}}}$$

F is the load per unit width,

R_i is the radius of cylinder i, $R_i = d_i \sin\varphi / 2$ for the gear teeth,

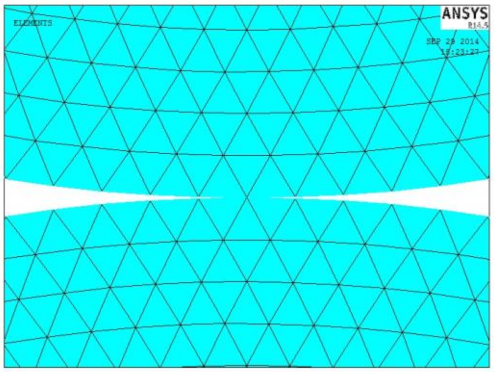
φ is pressure angle, ν_i is Poisson's ratio for cylinder i,

E_i is Young's modulus for cylinder i.

B. *Result of the Contact Stress Analysis*

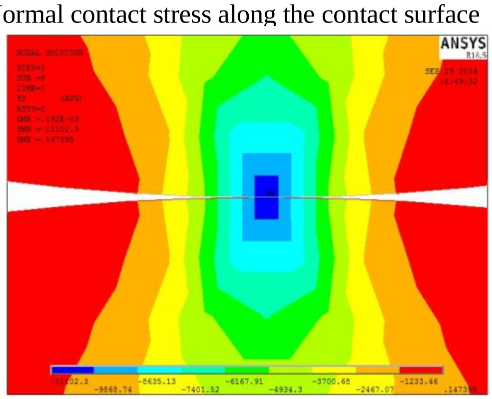
The objective of the contact stress analyses is to gain an understanding of the modeling and solution difficulties in contact problems and examine the contact stresses in the gears. In order to verify the FEM contact model procedure, contact between two cylinders is modeled. To reduce computer time, only half cylinders were meshed in the model as shown in Figure 3.5(a). The fine meshed rectangular shaped elements were generated near contact areas shown as 3.5 (b). The dimensions of the elements are based on the half width of the contact area. The contact conditions are sensitive to the geometry of the contacting surfaces, which means that the finite element mesh near the contact zone needs to be highly refined. Finer meshing generally leads to a more accurate solution, but requires more time and system resources. It is recommended not to have a fine mesh everywhere in the model to reduce the computational requirements.

T r i a n g u l a r shaped elements were generated near contact areas

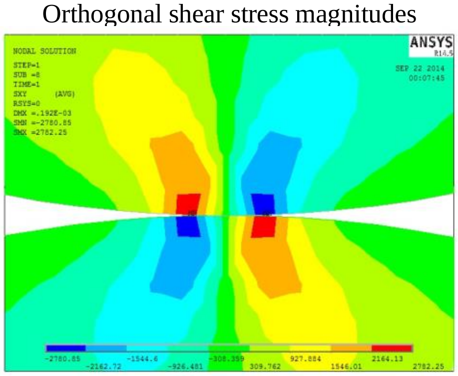


The comparison of results from FEM and the Hertzian theoretical formula are shown in Figure 3.7 in which the two distributions lie very close. The red color line represents the value from the theoretical Hertz equation and the blue color points represent the results from ANSYS. They match very well. It is easy to see the blue color points are on the red curve.

Normal contact stress along the contact surface



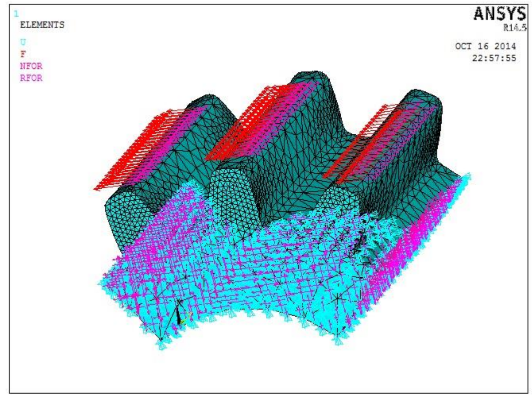
Orthogonal shear stress magnitudes



THE THREE DIMENSIONAL MODEL

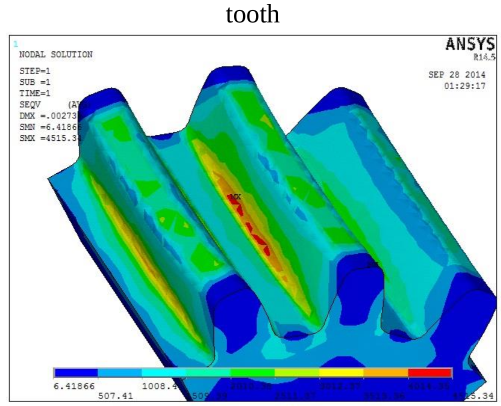
Figure 4.1 shows how to mesh the 3D model and how to apply the load on the model. The element type "SOLID TETRAHEDRAL 10 NODES 187" was chosen. Because "SMART SET" was chosen on the tool bar there are many more elements near the root of the tooth than in other places. There are middle side nodes on the each side of each element. So a large number of degrees of freedom in this 3D model take a longer time to finish running.

Figure 4-1 FEM bending model with meshing



From the stress distributions on the model, the large concentrated stresses are at the root of the tooth. Figure 4.2 shows large Von Misses stresses at the root of the tooth. They are equal to the tensile stresses. The tensile stresses are the main cause of crack failure, if they are large enough. That is why cracks usually start from the tensile side. From the Lewis equation if the diameters of the pinion and gear are always kept the same and the number of teeth was changed, the diametral pitch will be changed or the module of gear will be changed. That means that there are different bending strengths between the different teeth numbers. Different Maximum Von Misses with different numbers of teeth are shown.

Figure 4-2 Von Misses stresses with 28 teeth on the root of tooth



TORSIONAL MESH STIFFNESS AND STATIC TRANSMISSION ERROR

Getting and predicting the static transmission error (TE) is a necessary condition for reduction of the noise radiated from the gearbox. In the previous literature to obtain TE the contact problem was seldom included because the nonlinear problem made the model too complicated. This chapter deals with estimation of static transmission error including the contact problem and the mesh stiffness variations of spur gears. For this purpose, an FEA numerical modelling system has been developed. For spur gears a two dimensional model can be used instead of a three dimensional model to reduce the total number of the elements and the total number of the nodes in order to save computer memory. This is based on a two dimensional finite element analysis of tooth deflections. Two models were adopted to obtain a more accurate static transmission error, for a set of successive positions of the driving gear and driven gear. Two different models of a generic gear pair have been built to analyze the effects of gear body deformation and the interactions between adjacent loaded teeth. Results are from each of the two models' average values.

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