Load Sharing Based Analysis of Helical Gear using Finite Element Analysis Method

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Abstract
A gear is a rotating machine part having cut teeth, which mesh with another toothed part in order to transmit torque. Gears may be spur, helical, bevel or worm in which Helical Gear is most common type of gear used in engineering applications. The increased in performance requirement such as high load carrying capacity, high speed, high reliability and long life leads to new design of gear. These gears should be strong; corrosion resistance light weighted and should be durable for a long time. The bending and surface strength of the gear tooth are considered to be one of the main contributors for the failure of the gear in a gear set. Thus, the analysis of stresses on gears is to minimize or to reduce the failures and for optimal design of gears. This work investigates the characteristics of a helical gear system mainly focused on bending and contact stresses using analytical and finite element analysis. Mating gears of both gear and pinion is taken to be same materials with same size. The model for helical gear and pinion was done in Pro-E and stress analysis was carried out using ANSYS. The results obtained from ANSYS are presented in this work.

Keywords — *Load Sharing, Gear, Stress, ANSYS.*

I. INTRODUCTION

Gears are most commonly used for power transmission in all the modern devices. These toothed wheels are used to change the speed or power between input and output. The design and manufacture of precision cut gears, made from materials of high strength, have made it possible to produce gears which are capable of transmitting extremely large loads at extremely high circumferential speeds with very little noise, vibration and other undesirable aspects of gear drives. The rapid development of heavy industries such as vehicle, shipbuilding and air craft industries require advanced application of gear technology.

Rama mohana rao and Mwthuveerappan [1] developed the geometry of helical gears by simple mathematical equations. The load distribution for various positions of the contact line and the stress analysis of helical gears are carried out using the three-dimensional finite element method. A computer program has been developed for the stress analysis of the gears. Root stresses are evaluated for different positions of the contact line when it moves from the root to the tip.

Also parametric study is made by varying the face width and the helix angle to study their effect on the root stresses of helical gears. But they have not included the contact effect and also the load on the adjacent teeth. Vijayarangan and Ganesan [2] elaborated that the study of contact stress in a pair of mating gear teeth, under static conditions, by using a three-dimensional finite element method.

Shreyash [3] developed an analytical approach and modeling procedure to evaluate stress distribution under velocity and moment would provide a useful tool to improve spur gear design with high efficiency and low cost. The purpose of this work is to analyze and validate the stress distribution in spur and helical gears using contemporary commercial FEM program ANSYS coupled with the Pro/E solid modeler. Raynald Guilbault, et al [4] calculated the performance and durability of a gear pair which depend largely on manufacturing and assembly precision. The AGMA/DIN stress equations include factors to account for inaccuracies in meshing gear teeth, and to reflect non-uniform distribution of load on the contact line, but their simplified formulation is not always sufficient. Hence they developed an Express model to determine load sharing, fillet stresses, and pressure distribution along contacting surfaces between meshing helical gear teeth.

Baud and Velex [5] developed specific finite element code aimed at stimulating dynamic tooth loading in geared rotor systems. Experiments have been conducted on a high-precision single stage spur and helical gear reducer with flexible shafts mounted on hydrostatic or hydrodynamic bearings. The numerical model is based on classical elements (shaft, lumped stiffness's) and on a gear element which accounts for non-linear time-varying mesh stiffness, gear errors and tooth shape modifications. External and parametric excitations are derived from the instantaneous contact conditions between the mating flanks by using an iterative contact algorithm inserted in a time- step integration scheme. First, experimental and numerical results at low speeds are compared and confirmed that the proposed tooth mesh interface model is valid. Comparisons were then extended to dynamic fillet stresses on both spur and helical gears between 50–6000 rpm on pinion shaft. Mishra and Murthy [6] elaborated that the study of complex design problem of helical gear requires superior software skills for modelling and analysis and used Pro-E wild fire, ansys and matlab.

II. HELICAL GEAR MODELING

The word “Relation” and “Parameters” itself gives the idea about relating the feature with the help of equations. Relations are used to express dependencies between the dimensions of a feature.

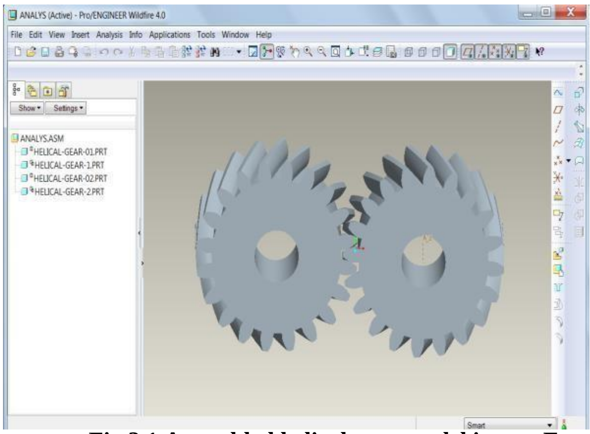

Fig 2.1 Assembled helical gear model in pro-E

To make the analysis four teeth’s in the first gear and three teeth in second gear are taken. Then both the gears are meshed which as shown in fig 1.5 which exported to the finite element analysis package ANSYS through IGES format.

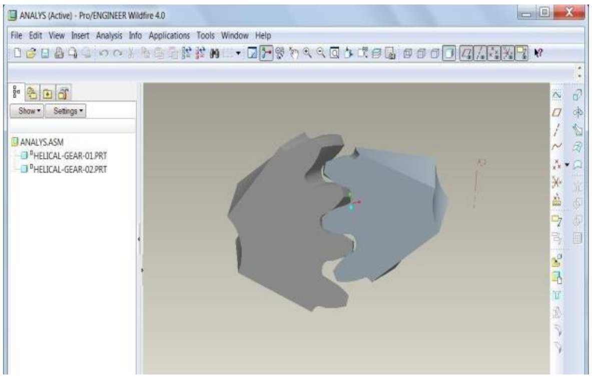

Fig 2.2 Model of helical gear

III. FINITE ELEMENT ANALYSIS

The geometric model of the gear pair is imported in to the ANSYS software. The imported model in to the ANSYS environment is shown in figure below

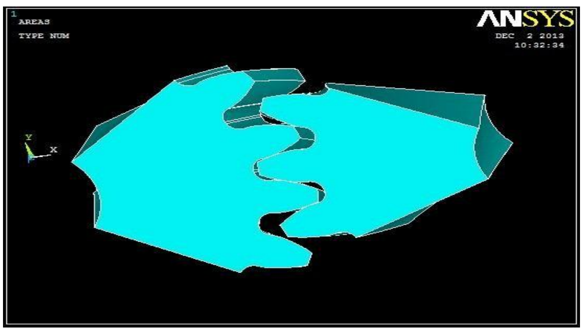

Fig 3.1 Gear model imported in ANSYS

The geometric model is meshed with Solid 187 element. The geometry, node locations, and the coordinate system for this element are shown in figure. Solid187 element is a higher order 3-D, 10-node element having three degrees of freedom at each node: translations in the nodal x, y, and z directions. Solid187 has quadratic displacement behaviour and is well suited to modeling irregular meshes. The element has plasticity, hyper elasticity, creep, stress stiffening, large deflection, and large strain capabilities.

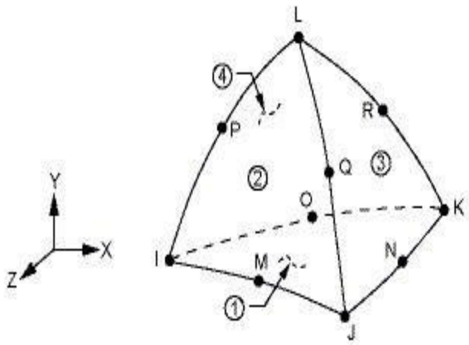

Fig 3.2 Solid187 Element Geometry

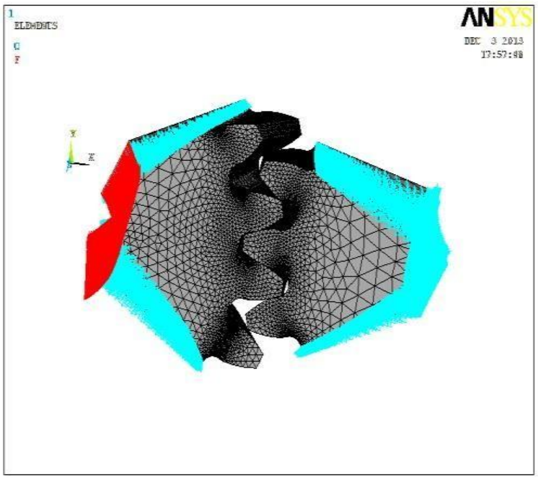

Fig 3.3 Finite Element Model of the Gear Drive

IV. RESULTS AND DISCUSSIONS

In this work Pro Engineer is used to create the Gear geometry and then it is imported in Ansys APDL 14.0 for analysis. Gear analyses in the past were performed using analytical methods, which required a number of assumptions and simplifications.

In general, gear analyses are multidisciplinary, including calculations related to the contact stresses and the failures. In this work, load sharing based fillet and contact stress induced for the given load in helical gear drive analyses are performed, with the main aim of designing helical gears to resist contact stress and bending stress failure.

Nowadays computers are becoming more and more powerful, and that is the reason why people tend to use numerical approach to develop theoretical models to predict the effects. Numerical methods can potentially provide more accurate solutions since they normally require much less restrictive assumptions. However the important thing is to choose the correct model and the solution methods to get the accurate results and also reasonable computational time.

In this design first the solid model of the helical gear is made with relations and parameters modeling option in Pro Engineer. After the modeling of helical gear the assembly is created of two helical gears in contact. The contact is defined at the pitch circle radius with the appropriate centre distance between the two gears.

Then the whole assembly is imported in ANSYS APDL 14.0 for bending stress and contact stress analysis. The results of ANSYS 14.0 are then compared with the AGMA standards for the specified gear set in contact. The load sharing based bending stress for the helical gear drive with 20^0 helix angle is shown in Figure 5.1.

The maximum stress developed at the fillet area due to bending effect is 1.6 MPa. Also it has been observed that while one pair is leaving the contact from a mesh cycle the other pair already in contact this will reduce the impact load on the gear drive.

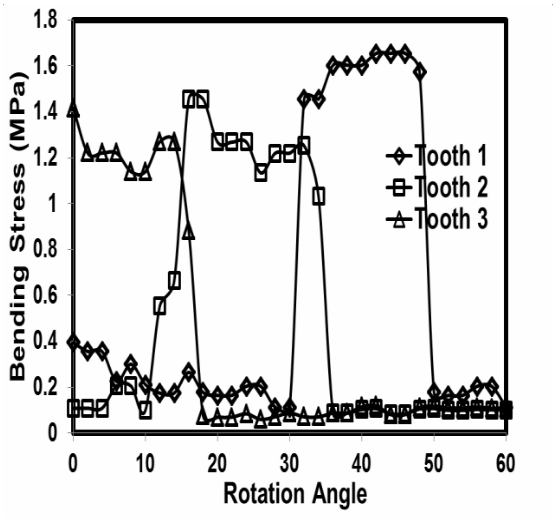

Fig 4.1 Bending stress for the helical gear drive with 20^0 helix angle

The stress at the fillet area of the Tooth 2 of pinion for the rotation angles of 16^0 and 32^0 is shown in figure below

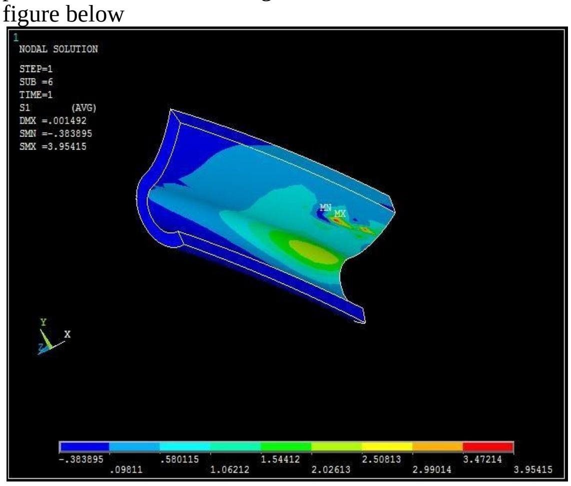

Fig 4.2 Stress at the fillet area for the rotation angles of 16^0

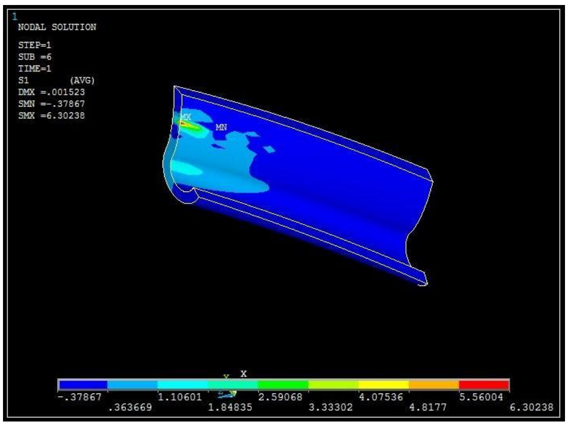

Fig 4.3 Stress at the fillet area for the rotation angles of 32^0

It is observed from the figures that the maximum bending stress occurs at the back side of the face width for the rotation angle 16^0 and it occurs at the front side of the face width for the rotational angle of 32^0.

The contact stress developed at the Tooth 1, Tooth 2 and Tooth 3 is shown in fig 1.11. It is observed from the figure that the maximum contact stress developed in a pair of gears is 24 MPa.

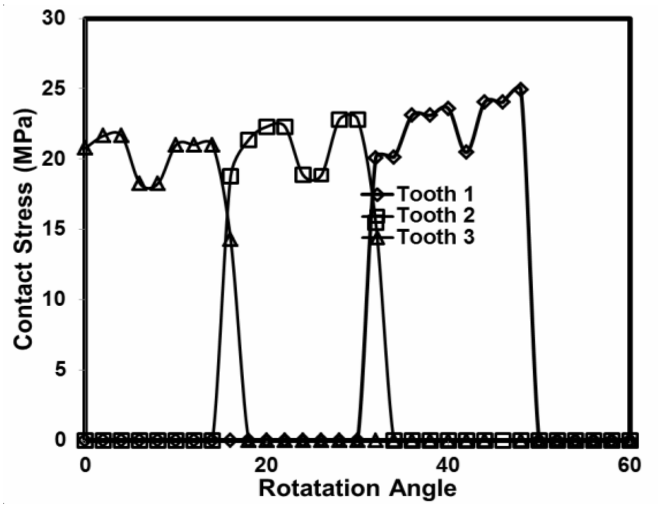

Fig 4.4 Contact stress for the helical gear drive with 20^0 helix angle

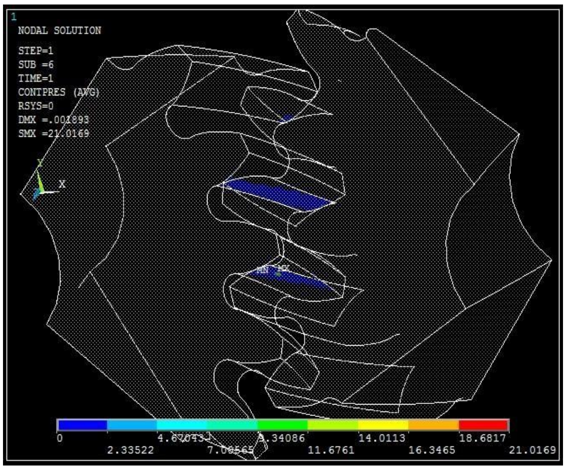

Fig 4.5 Contact stress for the rotation angle of 14^0

The contact stress developed for the simultaneously contacting pair for the rotation angle of 14^0 and 32^0 is shown in figure.

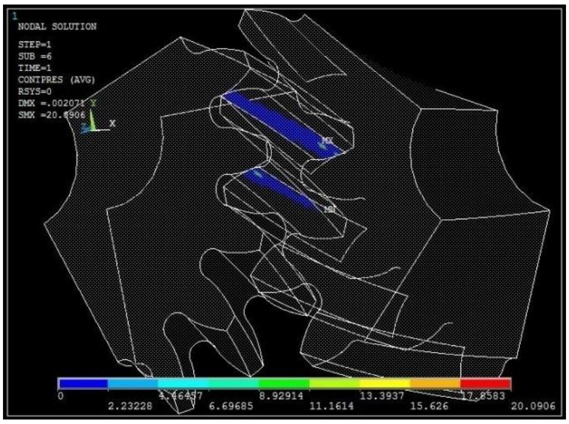

Fig 4.6 Contact stress for the rotation angle of 32^0

From the above Fig 4.5 and Fig 4.6 it can be observed that there is double pair contacts exist in the gear system.FEA results of the contact and bending stresses are compared with the AGMA stress calculations and it is given in table.

FEA Result Compared With the AGMA Calculations

Description	AGMA	FEA
Contact stress	295.69MPa	24MPa
Bending stress	5.52 MPa	1.6MPa

It is evident that there is difference between the stress level obtained using AGMA standard and the FEA results due to the reason that the load sharing effect and actual stress concentrations are included in the FEA model.

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