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Contact stress analysis of spur gears using the finite element method

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Abstract
Gear is one of the most critical components in mechanical power transmission systems. The analytical methods for calculating gear contact stresses use Hertz's equations, which were originally derived for contact between two cylinders. So, it's necessary to develop and to determine appropriate models of contact elements, and to calculate contact stresses using ANSYS. In this work, the analysis of contact stresses under the application of different modules with the same moment on a spur gear. All spur gear models have been designed in SolidWorks. Analysis was performed in ANSYS by importing spur gear models to the Static structural component of the software. Contact regions were discretized finely in the software in order to replicate the real loading conditions. The contact stress equivalent (Von-mises stress) was obtained on various modules using the FEA solver. The results from the ANSYS software were then compared with Hertz contact stresses.

Keywords: Contact Stress, Involute Spur gear, FEM.

Introduction

Gears are toothed members that transmit power or motion between two shafts by meshing without any slip. Hence, gear drives are also called positive drives. When two gears mesh, if one gear is bigger than the other, a mechanical advantage is produced, with the rotational speeds and the torques of the two gears differing in proportion to their diameters [1-3]. Gears are found in various mechanical devices such as cars, clocks, rotating machinery, etc. Ali Raad Hassan [4] did a research study in which contact stress analysis between two spur gear teeth was studied in different contact positions, representing a pair of mating gears during rotation. The results were presented, and finite element analysis results were compared with theoretical calculations. Bharat Gupta et al. [5] presented a paper to suggest that a thorough study of the contact stress developed between the different matting gears is mostly important for the gear design. They have used Hertz equations, which are an analytical method of calculating gear contact stresses, originally derived for contact between two cylinders. The calculated contact stresses were used using ANSYS and the results were compared with Hertz theory. Ignacio Gonzlez-Peraz [6] developed a finite element model and validated it in terms of contact area, pressure distribution, and maximum contact pressure for those cases where the Hertz theory can be applied, provided partial crowning to the finite element model where the Hertz theory does not work properly, and concluded that the analysis results give good agreement with the Hertz theory for calculating maximum contact pressure, contact area, and deflection with minute errors. Xianzhang FENG [7] uses a precise model in a large-scale CAD software and defines the stress and displacement fields to determine the maximum equivalent stress and maximum displacement. The results agree well with the actual meshing law and not only validate the model but also allow for dynamic gear analysis. Wei Yangang [8] gives theoretical research to find the location of the maximum contact stress for a pair of involute spur gears, whose contact ratio is greater than 1, and validated by the finite element method. Rubin D. Chacon et al. [9]

analysed the contact stress between spur gear teeth using a plane model and validated Hertz stress and AGMA contact stress with finite element contact stress and concluded that FEM is able to simulate contact stress in a pair of mating gear. The contact stress is highest at the higher points on the involutes and lower when a single pair of teeth assumes the full load transmitted and minimal at the pitch point of contact. Seok-Chul Hwanga et al. [10] studied the contact stress analysis for a pair of mating gears during rotation. He investigated the respective variations of contact stress analysis for helical and spur gears with the different contact positions in a pair of mating gears. The variation of contact stresses during rotation at the lowest point of single tooth contact (LPSTC) and the AGMA (American Gear Manufacturers Association) equation for the contact stress Select the design that considers the contact stress is stricter than the AGMA Standard. By using FEA analysis, calculate the contact fatigue strength of the material for the appropriate strength and safety. He carried out FEA analysis using AGMA equations. Prashant Kumar Singh et al. [11] investigated the thermal and wear behaviour of gears made of different thermoplastic materials like Acrylonitrile Butadiene Styrene (ABS), High Density Polyethylene (HDPE), and Poly Oxymethelyne (POM) at different torque levels and different rotational speeds. The authors found that the wear rate of polymer gears increases with torque but decreases with rotational speed.

Each gear tooth is considered as a cantilever beam when it transmits the load, consequently being subjected to bending [12]. In the calculation of Root Bending Stress (RBS) and Surface Contact Stress (SCS) through the Lewis formula and Hertz equation, there are certain drawbacks, such as in the case of bending, the entire load is carried on a single tooth and in the contact condition of gears are considered as two cylinders are in contact [2]. Today, the finite element method is mostly used for predicting any kinds of stress, strains, and deformation in single parts and assemblies. In this study, analysis of contact stress has been done for spur gears made of structural steel under applied moments in the following conditions: Spur gears of different modules, with the same number of teeth on pinions and gears, equal pressure angle and face width, are subjected to a fixed value of moment. The analysis is done on ANSYS 16, and its results are compared with the theoretical results.

Modelling of Spur Gear

Spur gears were designed using the SolidWorks 16 version. The modelling of spur gears in various unit systems involves specifying the necessary geometric parameters. The ANSI Metric system was used for designing all spur gears. Table 1 shows the relationship between the shape of a spur gear and its module, pressure angle, and number of teeth, which can be used to figure out the other design parameters. The geometric model of spur gear is shown in Fig. 1.

Table 1 Geometric parameters of spur gear.

Module	4	5	6	6.5
No. of teeth on pinion	20	20	20	20
No. of teeth on gear	30	30	30	20
Pressure angle	14.5˚	14.5˚	14.5˚	20˚
Face width (mm)	25.4	25.4	25.4	25.4

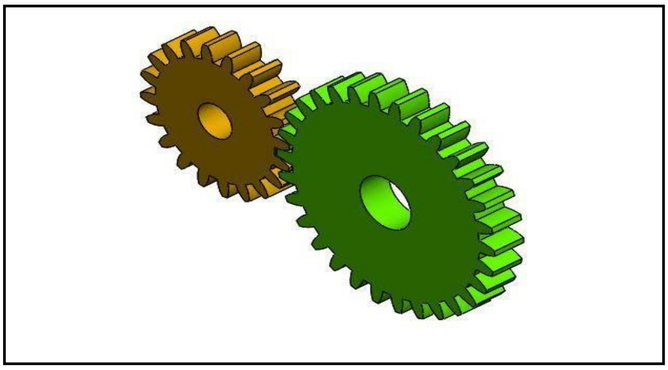

Fig.1 Geometric model of spur gear.

Meshing

The analysis problem of structural steel gears is converted from an infinite degree of freedom, continuum problem to one having a finite number of degrees of freedom, discretized continuum problem.

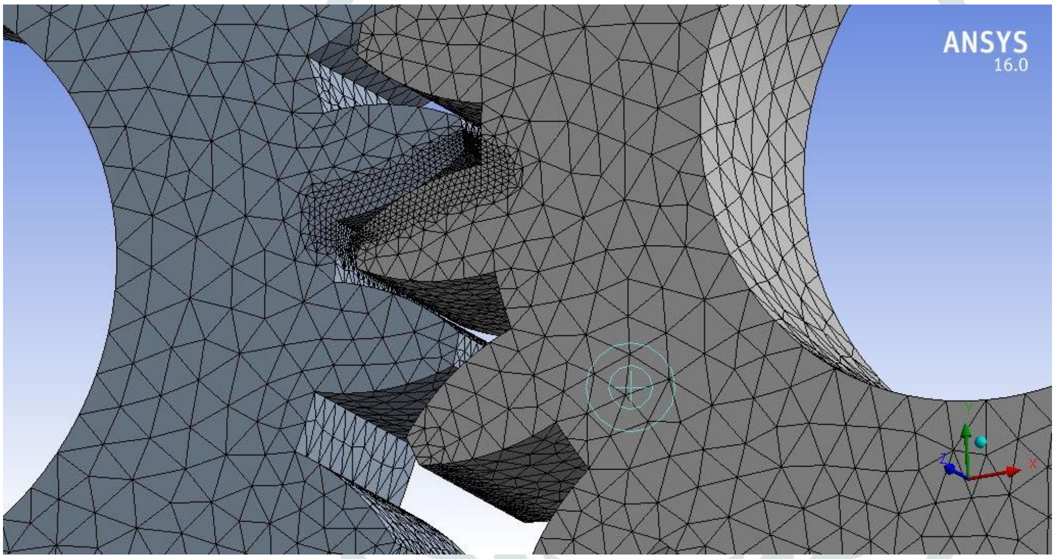

Fig.2. Meshed model of spur gears for module 6.

The continuum is divided into a number of finite elements by imaginary lines or surfaces. The interconnected elements may have different sizes and shapes. The spur gear was meshed after being imported into ANSYS, with the mesh quality set to very fine and the shape chosen being triangular elements and the number of nodes were recorded for all of the spur gears. Meshing was refined near the contact region as depicted by densely spaced elements near the contact surfaces of gear teeth for every spur gear model as shown in Fig.2.

Result and Discussion

The structural steel spur gears of modules 4, 5 and 6; pressure angle of 14.5 and face width of 25.4 mm with 20 teeth on pinions and 30 teeth on gears were loaded by a moment of 70 Nm in static structural analysis, ANSYS. The power transmission value for each spur gear pair is 10kW at a rotational speed of 1365 rpm in the anti-clockwise sense. The contact stress value for spur gear pairs as obtained by the finite element method is shown in figures 3 and 5. The numerical results are compared with the theoretical results and the percentage difference is estimated by the relation:

$$\%\ \text{Difference} = \frac{(Theortical\ result - numerical\ result)}{Theoretical\ result} 100 \quad (1)$$

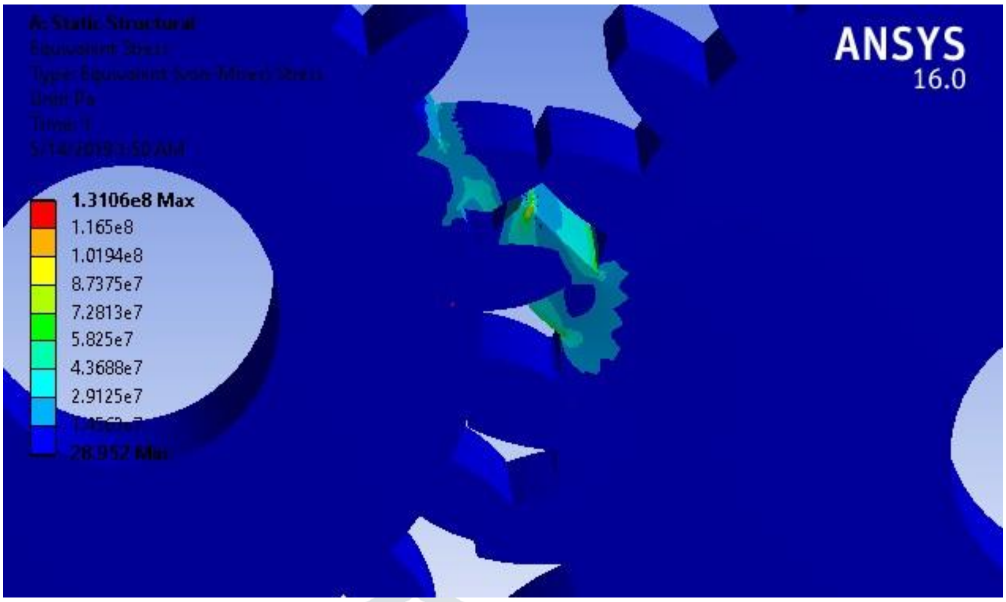

Fig.3 Von-mises stress on structural steel spur gear of module 4.

In Fig. 3 pinion is on the left side and is rotating in anti-clockwise sense. The result of analysis shows the maximum contact stress to be 131.06 MPa, occurring along the line of contact in a very localized spatial region. Theoretical contact stress, using Hertz equation for the given condition and by using safety allowance was calculated to be 142.45 MPa, and the percentage difference is given by equation 1 is of 8 % which is acceptable. From the report of the analysis, critical region i.e. surface along the fillet radius of teeth develops moderate intensity of contact stress shown by green-yellow region on the stress contour ploton the teeth. The inner region gear material is in least amount of contact stress. Comparison of front and rear parts of meshed pair shows on an average a higher intensity of stress developing at the contacting surfaces then the farther face or flanks.

From Fig. 4, the maximum contact stress value is 84.85 MPa. It is occurring along the line of contact in a very localized spatial region. At 1365 rpm, the pinion turns the gear counterclockwise, so the front face and flank of the gear have more contact stress than the back face and flank. Theoretical contact stress calculated using the Hertz equation and using safety allowances for the given condition was found to be equal to 87.47 MPa. Compared with the numerical contact stress with the theoretical contact stress, the percentage difference as given by equation 1 is 3%, which is acceptable. With an increase in module, the size of spur gear teeth also increases. The stress plot shows relatively lower values of contact stresses over teeth surfaces and interior regions compared to Fig 3. Contact stress decreases with an increase in module 4 to module 5.

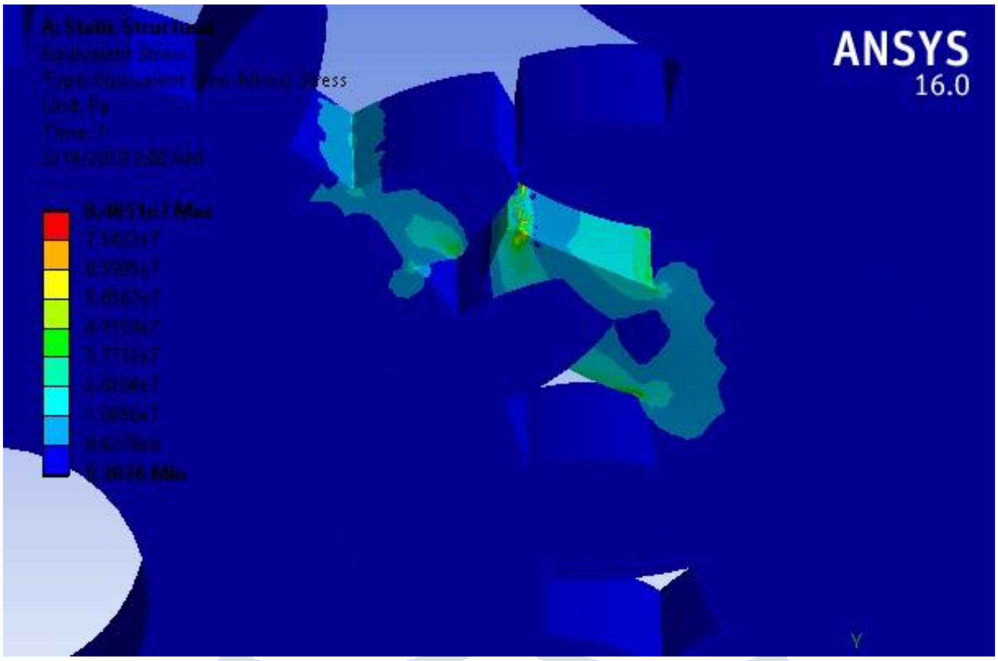

Fig.4 Von-mises stress on structural steel spur gear of module 5.

The contact stress values of the structural steel spur gear of module 6, pressure angle 14.5, and face width of 25.4 mm obtained from static structural analysis in ANSYS workbench are depicted in Fig. 5. The operating condition was 1365 rpm at a power transmission of 10 kw in an anti-clockwise sense. The result of the analysis shows the maximum contact stress to be 59.6 MPa, occurring along the line of contact in a very localised spatial region. Theoretical contact stress, using Hertz equation 7.10 for the given condition and by using safety allowance, was calculated to be 56.05 MPa, and the percentage difference given by equation 1 is 6.0%, which is acceptable. The comparison of the spatial distribution of contact stresses of module 6 spur gear with that of stresses developed in module 5 spur gear brings to notice an overall reduction in the intensity experienced by structural steel with an increase in modules.

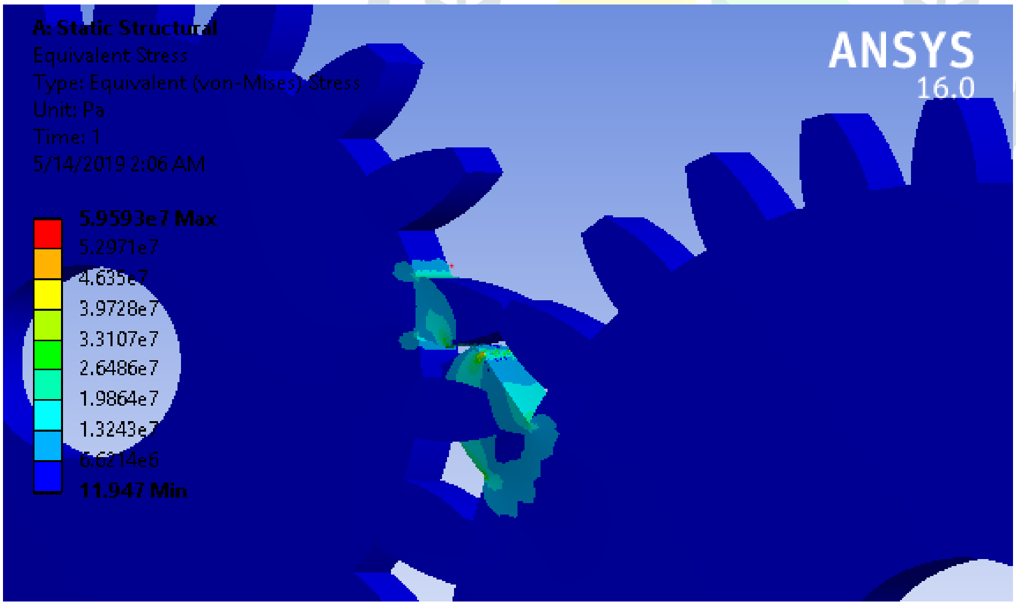

Fig.5 Von-mises stress on structural steel spur gear of module 6.

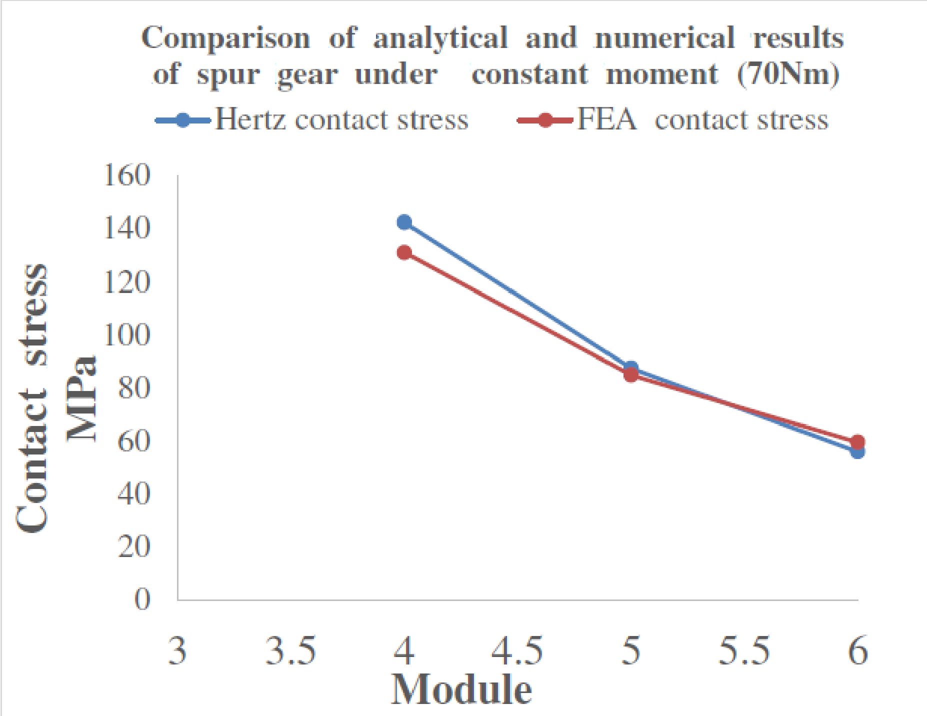

Fig. 6 Variation of contact stress with module for 70 Nm applied moment.

The variation of contact stress with different modules is shown in Fig. 6. The contact stress derived by FEM for modules 4, 5, and 6 under a moment of 70 Nm is compared with analytical results in Table 2.

Table 2 Comparison of analytical and numerical results

Gear module	Hertz contact stress MPa	Contact stress (FEA) MPa	Percentage difference %
4	142.45	131.06	8
5	87.47	84.85	3
6	56.05	59.60	6

Structural steel spur gear is used for high torque, low speed transmission purposes. This implies a higher stress environment on the surfaces of gear teeth in contact. For the above loading conditions, the maximum value of contact stress occurs for the smallest spur gear pair, which is 142.45 MPa for module 4 structural steel spur gear. In all of the cases, the maximum contact stress obtained by both methods is less than the yield stress of the material. Theoretical calculation of contact stress presents the maximum contact stress under any specific loading condition. The comparison of maximum contact stress obtained by ANSYS software with maximum Hertzian contact stress shows a similar trend of a decrease in contact stress with an increase in module, and the percentage error is less than 10%.

Conclusion

The following conclusions have been drawn from the contact stress analysis of spur gears with various modules:

- The contact stress on the gear develops non-uniformly during power transmission at a fixed operational speed.

- The maximum contact stress obtained by FEA and for spur gears pairs of different modules (m) m=4, m=5 and m=6 under 70 Nm moment, was 131.06 MPa, 84.85 MPa & 59.06 MPa respectively.
- When the module of the spur gear pair is increased, then the contact stress decreases.
- The FEA and theoretical contact stress results are comparable with a 10% under difference.

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