IJSRD - International Journal for Scientific Research & Development| Vol. 4, Issue 04, 2016 | ISSN (online): 2321-0613

Effect on Strength of Involute Spur Gear Tooth by Considering Uncertainties in Geometry and Material Properties by Using Finite Element Methods

C.C. Pawar[1] S.G.Kulkarni[2] P.P.Kulkarni[3]
[1]Student [2]Professor
[1,2]Department of Mechanical Engineering
[1,2]SKN COE, Korti, Pandharpur

Abstract— There are several failure mechanisms for spur gears. Bending failures is one of the main failure modes in spur gears. The Bending stresses in a spur gear can be successfully calculated by finite element analysis. The analysis of spur gear is made in present work for different geometric conditions and material properties. The uncertainties in various geometric parameters like fillet radius, face width, tooth thickness etc. and the uncertainties in material properties are considered to analyze its effects on beam strength of spur gear. A parametric model of spur gear is modeled with the help ANSYS Workbench and then finite element analysis is carried out on the model with the help of ANSYS software. A finite element model with a segment of one tooth is considered for analysis. The gear tooth is meshed and a load is applied on the gear. By applying boundary conditions, the Bending stresses in a spur gear tooth are calculated by Finite Element Analysis which is then compared with the bending stresses calculated by using the LEWIS Equation.

Key words: Reliability, Uncertainties

I. INTRODUCTION

A pair of teeth of gears in action is generally subjected to two types of cyclic stresses: bending stresses inducing bending fatigue and contact stress causing contact fatigue. Both these types of stresses may not attain their maximum values at the same point of contact. However, combined action of both of them is the reason of failure of gear tooth leading to fracture at the root of a tooth under bending fatigue and surface failure, like pitting or flaking due to contact fatigue. These types of failures can be minimized by careful analysis of the problem during the design stage and creating proper tooth surface profile with proper manufacturing methods. The finite element method is capable of providing this information, but the time needed to create such a model is large. In order to reduce the modeling time, a preprocessor method that creates the geometry needed for a finite element analysis may be used. Finite element method is very often used to analyze the stresses in an elastic body with complicated geometry, such as a gear.

To get the gear of more durability we can use improved material, hardening the gear surfaces with heat treatment and carburization, shot penning can be done to improve the surface finish, to change the pressure angle by using asymmetric teeth, introducing the stress relieving features of different shape, changing the addendum of the spur gear and altering the design of root fillet are the other methods.

Traditional deterministic design approaches compensate for uncertainties through the use of empirical safety factors, which do not provide sufficient information to achieve optimal use of available resources in terms of material, manufacturing and operational costs. To design a product that will perform a function reliably, the reliability must be considered as an important functional requirement all the way through the design process, from the customer’s need to the final product. Addressing these issues comprehensively at an early design stage is necessary to produce competitive product that functions consistently during its intended service life. Indeed, consistent levels of safety and reliability can be achieved based on the probabilistic design methods.

Probabilistic design, such as reliability-based design and robust design, offers tools for making reliable decisions with the consideration of uncertainty associated with design variables or parameters and simulation models. It allows the designer to assess the reliability of the mechanical system. This is impossible with the factor of safety approach. One important task of a probabilistic design is uncertainty analysis, through which we understand how much the impact of the uncertainty associated with the system input is on the system output by identifying the probabilistic characteristics of system output. The uncertainty in a design performance is described probabilistically by its mean variance, the probability density function or the cumulative distribution function

By understanding the probability distributions of the design parameters, the designer can design for a specific reliability or quality level by producing designs that are robust to variations [10].

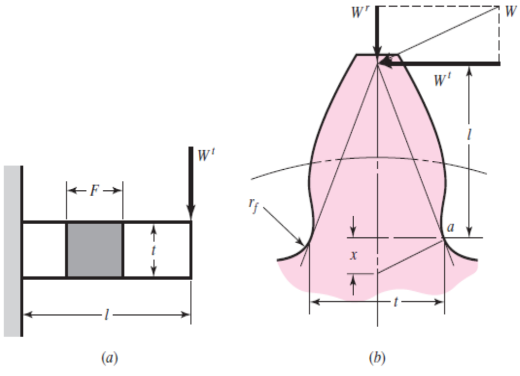

Fig. 1: Gear tooth

II. REVIEW OF LITERATURE

Wilfred Lewis developed the basic model for bending stress in gear teeth in 1892. In his analysis, Lewis considered a gear tooth to be a loaded cantilever beam with a force applied to the tip of the gear. He made the assumptions like the load is applied to the tip of a gear tooth; only the tangential component of the force will be a factor (the radial component is neglected); the load is distributed uniformly across the entire face width of the gear; forces due to tooth sliding friction are negligible; and no stress concentration is present in the tooth fillet. [7] Christos A. Spitas and Vasilis A. Spitas [1] did his work on a new spur gear 200 design that works interchangeably with standard 200 system and achieves increased tooth bending strength and hence the load carrying capacity. In this design, circular fillet replaces the normal trochoidal fillet, yielding large cross sectional at the tooth root and lower stress con-centration. V. Spitas, Th. Costopoulos and C. Spitas [2] did his work on spur gear teeth with circular instead of the standard trochoidal root fillet is introduced and investigated numerically using Finite Element Analysis. M. Savage and Rubadeux [3] has propose a bending strength model for internal spur gear teeth, this mod-el assist design efforts for unequal addendum gears and gears of mixed materials. M Koilraj, Dr. G Muthuveerappan and Dr. J. Pattabiraman [4] on the basis of their work give the conclusion that, the stress correction factor and the form factor in-creases with the increase in positive profile correction. The modified pitch cone method is first presented and verified in the gear research center of Dong Feng vehicle-bridge Co. Ltd. [6] 3D static and dynamic contract/impact analysis of gear drives were developed by Rufang Li. The tooth load allocation and result are derived under the static load. The analysis of the stress distribution of gear system under dynamic loading conditions and simulates the stress of gears under conditions of initial speed and a sudden load being applied. [6]

Kawalec et al developed a comparative analysis between ISO and AGMA tooth root strength evaluation methods and verification by FEM. Li studied contact strength and bending strength of spur gears with machining errors, assembly errors and tooth modifications by 3-D FEM models. Pedersen proposed asymmetric profiles to improve the bending stress in spur gears. But neither of them considers analytical, non-uniform models of load distribution. [9]

Zeping Wei [6] used three dimensional finite element methods to conduct surface contact stress and root bending stress calculations of a pair of spur gears with machining errors, assembly errors and tooth modifications. Deng used tooth contract analysis, loaded tooth contact analysis and finite element method to analyze the meshing behavior, tooth surface contract stress, maximum tensile, bending stress and maximum compressive bending stress.

Praveen Kumar, Harsh Raghuvanshi, studied designing of the spur gear on different geometric conditions and finding the effect of these on tooth load like by changing the concentration of SIC in SIC based aluminium gear. Addition of SIC increases the strength of Spur Gear. Effect is also analyzed by changing the modules of the gear and by changing the tooth width. Tooth load is calculated with help of Lewis equation & dynamic tooth load is calculated with help of Buckingham equation. Static analysis of the gear is done to find the Von-mises stress on the tooth of the gear in while

III. SIMULATION

The parametric modeling of spur gear tooth is carried out in ANSYS workbench by taking various parameters like module, pitch circle diameter, base circle diameter, pressure angle, addendum circle diameter, circular pitch, thickness of tooth, face width, fillet radius etc. Meshing is carried on the gear tooth model and fine meshing is done at the fillet radius region.

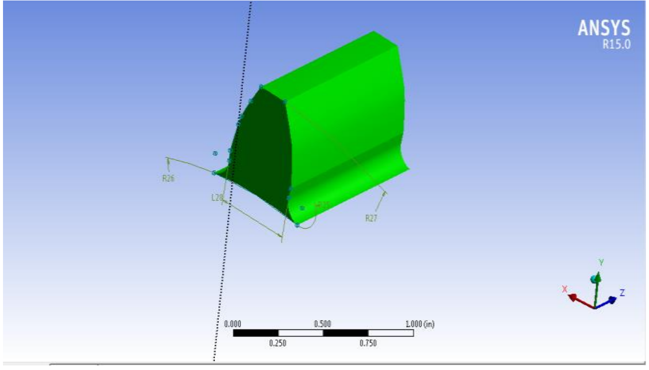

Fig. 2: Parametric Model of gear tooth

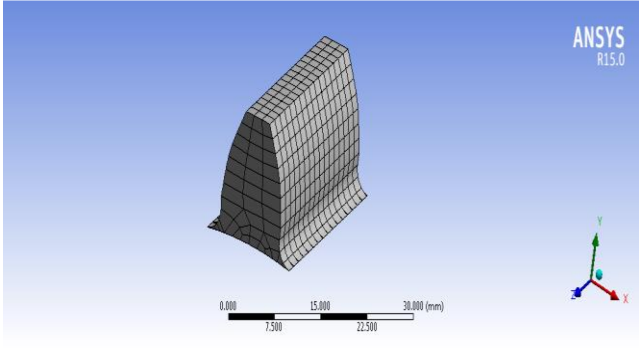

Fig. 3: Finite element meshed tooth.

The gear tooth is meshed with 3D elements with number of nodes 3747 and number of elements as 672.

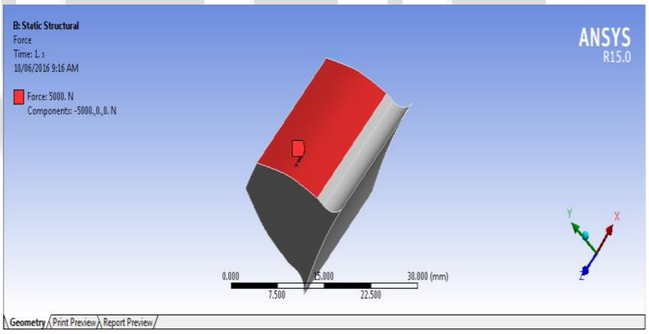
B: Static Structural
Force
Time: 1. s
10/06/2016 9:16 AM
Force: 5000. N
Components: -5000.,0.,0. N

Fig. 4: Loading

The finite element analysis is carried out on the model with the help of ANSYS software. A finite element model with a segment of one tooth is considered for analysis. The gear tooth is considered as a cantilever beam during analysis. A point load is applied at tip of the gear and the boundary conditions are applied.

A structural parametric analysis is carried out for 1000 samples Fig. shows project schematic with parametric correlations. A parametric correlation method is employed to find out the variation in bending stress by considering uncertainties in various geometric parameters like fillet radius, face width, tooth thickness, radius of addendum circle, radius of dedendum circle etc. and Material properties like Young’s modulus, Poisson’s ratio, Tensile strength etc. The Probabilistic design is used to determine the effect of one or more variables on the outcome of the spur gear tooth. Present work considers:

– Geometric parameters: Fillet radius, Face width, Tooth thickness etc.

- Material parameters: Young modulus, Poission ratio, Tensile strength etc.
- Load parameters: Force

Using uncertainties as stated above, probabilistic design is performed using ANSYS to know sensitivity of each parameter on Bending Stress. PDS within ANSYS uses Monte Carlo Simulation approach and analysis was looped through 1000 sample points considering the variations defined in the input variables and the corresponding statistical analysis of the output parameters.

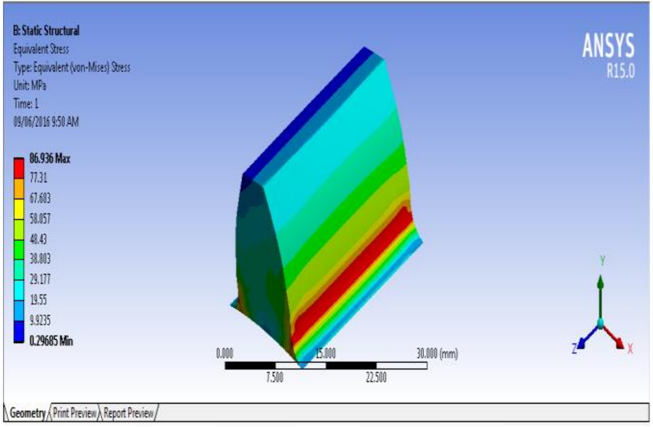

Fig. 5: FE Representation of the Model

Parameter type	Parameter	Value
Geometric	Fillet radius	0.1 in
	Face width	1 in
	Tooth thickness	0.43206 in
Load	Force	5000 N
Material	Young’s modulus	2E+10 pa
	Poisson’s Ratio	0.3
	Yield tensile strength	2.5E+8

Table 1: Parameters used in probabilistic design of spur gear tooth

After creating parametric model, analysis file for spur gear tooth has been created. The analysis file has been created for use during the probabilistic analysis. It is a parametric model of the problem geometry, materials, and loads. Within the analysis file, input variables are initialized and output variables are retrieved. An overview of the data is provided by several plots which shows Scatter plots of each input and output parameters. Scattered plot shows uncertainty in Bending Stress. Polynomial distribution of powers is indicated by colored lines. As degree of polynomial distribution is small, there is more uncertainty in Bending Stress. If linear correlation coefficient of scatter plot is small then there is less uncertainty in Bending Stress and if larger then there is more uncertainty in Bending Stress. The same is true for rank order correlation coefficient.

The Lewis bending equation is shown below is used for validation of results.

$\sigma_b = (W_t *P)/ (F*Y)$

IV. RESULTS AND DISCUSSION

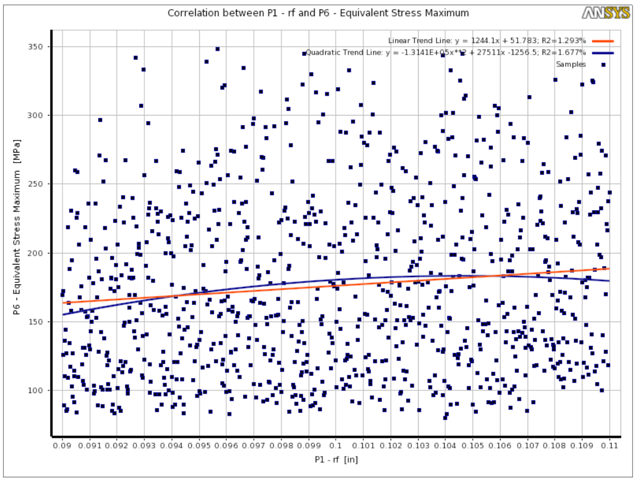

Fig. 6: Scatter Plot of fillet radius vs bending stress

Figure 6 shows bending stress in MPa vs. root fillet radius in inches. It is obtained after 1000 samples (tests) .Output parameter with combination of input parameters is plotted. Higher order Polynomial is used to plot scattering. It is observed that there is more scatter of bending stress from polynomial line within the fillet radius range 0.091 to 0.11 inch. The value of maximum bending stress at within fillet radius range 0.095-0.096 inch is around 350MPa.

Figure 7 provides information about the linear correlation between root fillet radius and equivalent bending stress. The degree of correlation is indicated by color in the matrix. The correlation value for the two parameters i.e. root fillet radius and equivalent stress associated with the square are 1 and 0.1137 respectively.

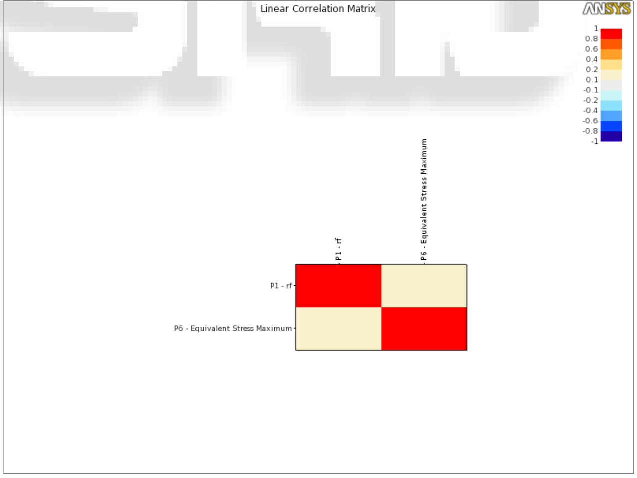

Fig. 7: Linear Correlation Matrix of fillet radius and bending stress

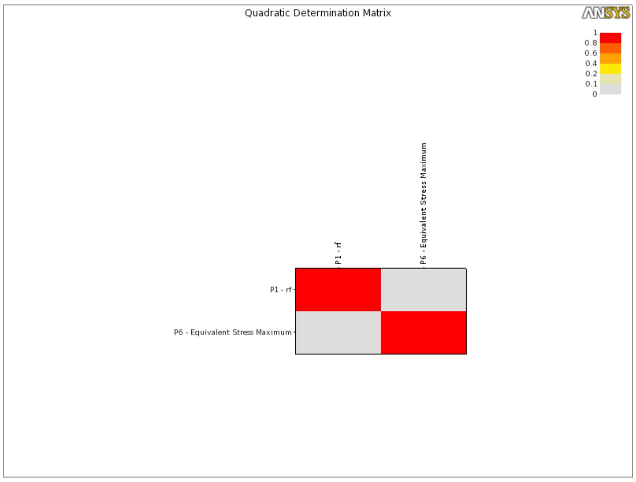

Fig. 8: Non-Linear correlation matrix of root fillet radius and equivalent bending stress

Figure 8 provides information about the nonlinear (quadratic) correlation between root fillet radius and equivalent bending stress. The degree of correlation is indicated by color in the matrix. The correlation value for the two parameters i.e. root fillet radius and equivalent stress associated with the square are 1 and 0.01131 respectively.

Figure 9 shows the sensitivities Chart**.** Sensitivities Chart***s* a**llows you to graphically view the global sensitivities of each output parameter with respect to the input parameters. Here the input parameter is root fillet radius and the output parameter is equivalent bending stress. Using uncertainties as stated above, probabilistic design is performed using ANSYS to know sensitivity of each parameter on Bending Stress.

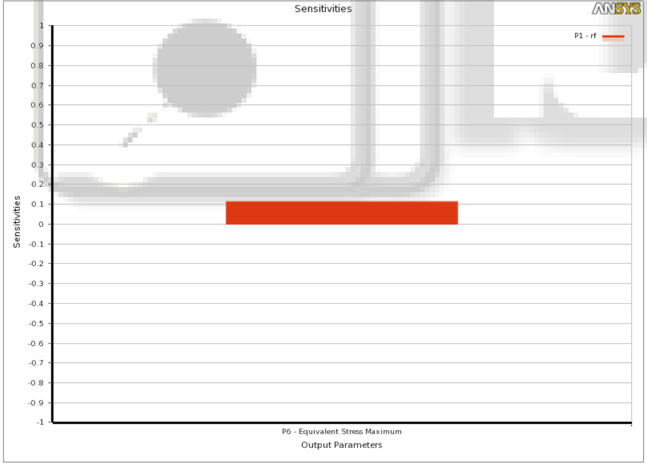

Fig. 9: Sensitivity chart of fillet radius and bending stress

Figure 10 shows bending stress in MPa vs. face width in inches. It is observed that there is more scatter of bending stress from polynomial line within the face width range 0.98 to 1.07 inch. It can be said from the graph, the value of maximum bending stress is observed within face width range 0.92-0.93 inch which is around 350MPa.

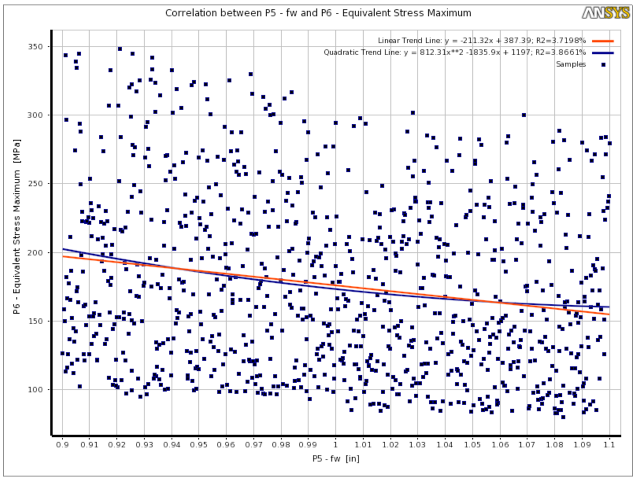

Fig. 10: Scatter plot of face width vs. bending stress

Figure 11 provides information about the linear correlation between face width and equivalent bending stress. The degree of correlation is indicated by color in the matrix. The correlation value for the two parameters i.e. face width and equivalent stress associated with the square are 1 and -0.1928 respectively.

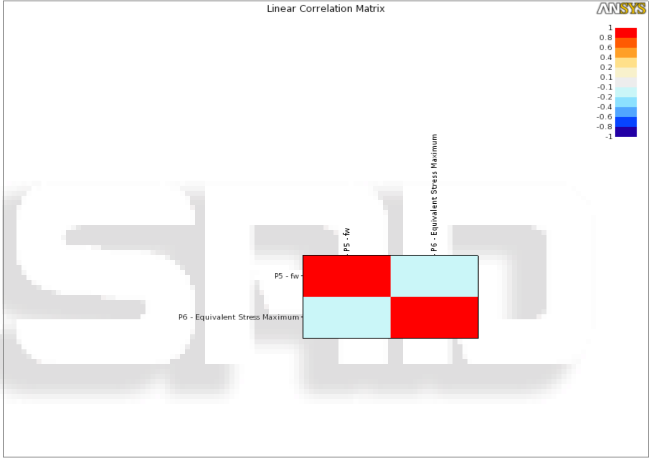

Fig. 11: Linear correlation matrix of face width and equivalent bending stress

Figure 12 provides information about the nonlinear (quadratic) correlation between face width and equivalent bending stress. The degree of correlation is indicated by color in the matrix. The correlation value for the two parameters i.e. face width and equivalent stress associated with the square are 1 and 0.037759 respectively.

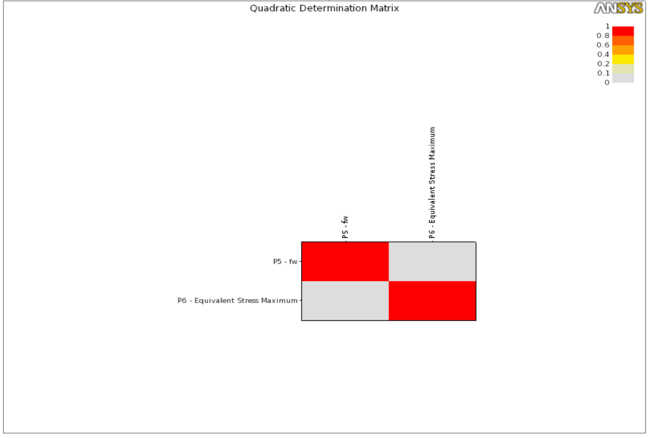

Fig. 12: Non- Linear correlation matrix of face width and equivalent bending stress

Figure 13 shows the sensitivities Chart**.** Sensitivities Chart*s* **a**llows you to graphically view the global sensitivities of each output parameter with respect to the input parameters. Here the input parameter is face width and the output parameter is equivalent bending stress. Using uncertainties as stated above, probabilistic design is performed using ANSYS to know sensitivity of each parameter on Bending Stress.

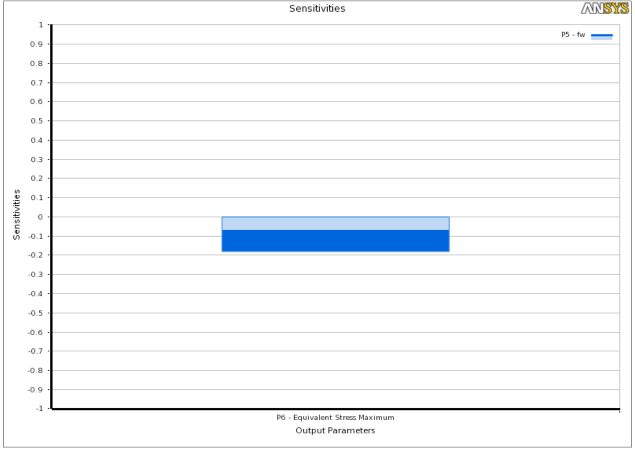

Fig. 13: Sensitivity between face width and equivalent bending stress

Figure 14 shows bending stress in MPa vs. tooth thickness in inches. It is observed that there is more scatter of bending stress from polynomial line within the tooth thickness range 0.3 to 0.34 inch. The value of maximum bending stress is observed within tooth thickness range 0.3-0.31 inch which is around 350MPa.

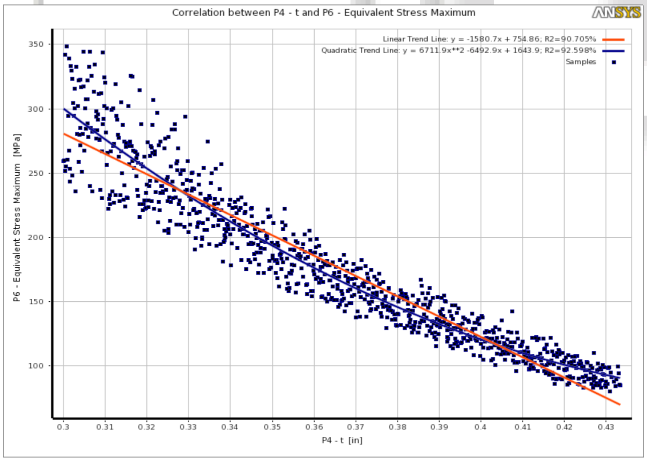

Fig. 14: Scattered plot of tooth thickness vs. bending stress

Figure 15 provides information about the linear correlation between tooth thickness and equivalent bending stress. The degree of correlation is indicated by color in the matrix. The correlation value for the two parameters i.e. tooth thickness and equivalent stress associated with the square are 1 and -0.9523 respectively.

Figure 16 provides information about the nonlinear (quadratic) correlation between tooth thickness and equivalent bending stress. The degree of correlation is indicated by color in the matrix. The correlation value for the two parameters i.e. tooth thickness and equivalent stress associated with the square are 1 and 0.95106 respectively.

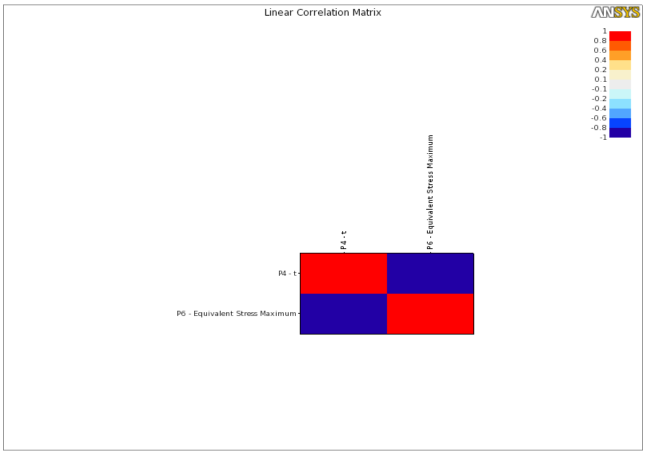

Fig. 15: Linear correlation matrix of tooth thickness and equivalent bending stress

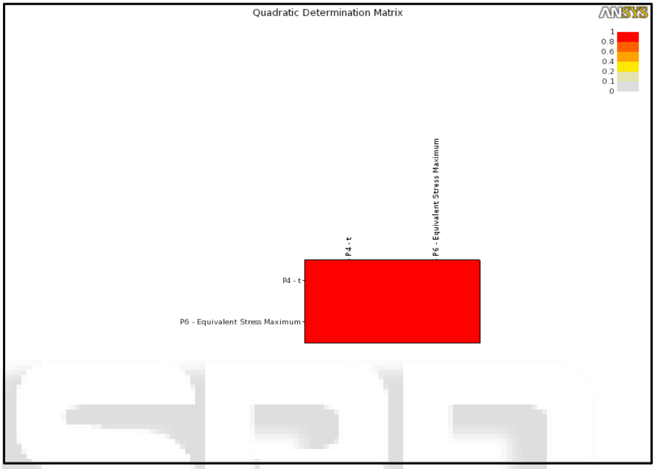

Fig. 16 Quadratic determination matrix of tooth thickness and equivalent bending stress

Figure 17 shows the sensitivities Chart**.** Sensitivities Chart*s* allows you to graphically view the global sensitivities of each output parameter with respect to the input parameters. Here the input parameter is tooth thickness and the output parameter is equivalent bending stress.

Figure 18 shows bending stress in MPa vs. Young's Modulus in Pa. Output parameter with combination of input parameters is plotted. From the graph below, it is observed that there is more scatter of bending stress from polynomial line within the range 1.8422 x $10^{11}$to 2.1922 x 10^{11} ·It can be observed from the graph, the value of maximum bending stress lies within range 1.9422 x $10^{11}$to 1.9922 x 10^{11} which is around 360MPa.

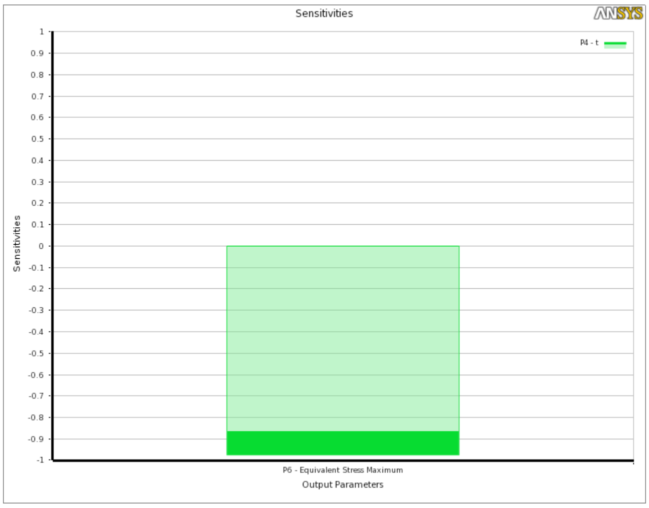

Fig. 16: Sensitivity between tooth thickness and equivalent bending stress

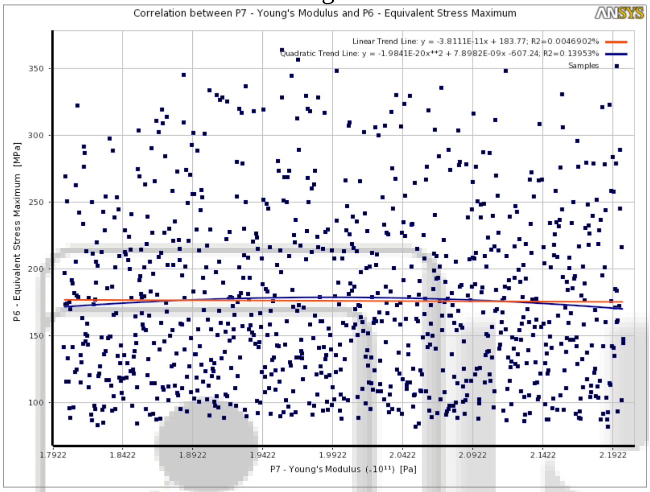

Fig. 17: Scattered plot of Young's Modulus vs. bending stress

Figure 18 shows bending stress in MPa vs. Poisons Ratio. It is obtained after analysis of 1000 samples (tests) by considering uncertainty in both geometrical parameters and material parameters .Output parameter with combination of input parameters is plotted. From the graph below, it is observed that there is more scatter of bending stress from polynomial line within the range 0.27 to 0.33. It can be observed from the graph, the value of maximum bending stress lies within range 0.32 to 0.325 which is around 360MPa.

Figure 19 shows bending stress in MPa vs. Tensile Strength in Pa. It is obtained after analysis of 1000 samples (tests) by considering uncertainty in both geometrical parameters and material parameters .Output parameter with combination of input parameters is plotted. From the graph below, it is observed that there is more scatter of bending stress from polynomial line within the range 2.2909 to 2.7409. It can be observed from the graph, the value of maximum bending stress lies within range 2.6909 to 2.7409 which is around 360MPa.

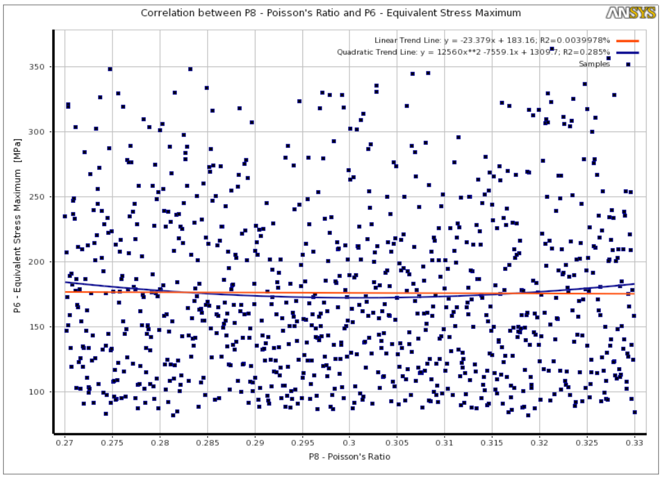

Fig. 18: Scattered plot of Poisson’s Ratio vs. bending stress

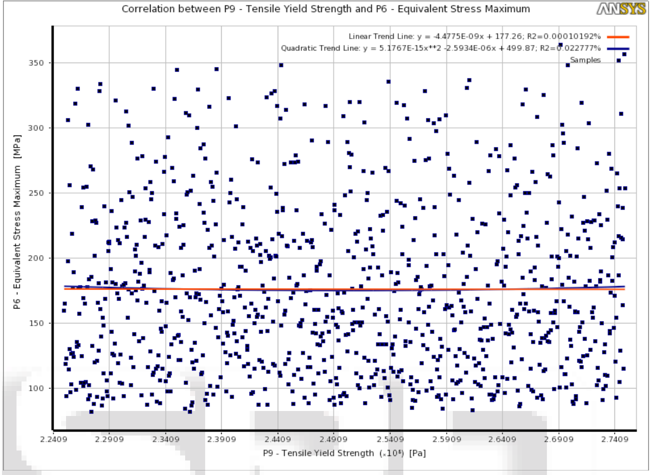

Fig. 19: Scattered plot of Tensile strength vs. bending stress

Validation

The results obtained from Finite Element analysis are then compared with the results obtained by using Lewis bending stress equation for given geometry of spur gear tooth. The table of validation is given below:

Sr.No.	Load N	Bending stress σ_b by FEM, MPa	Bending stress σ_b by Analytical methods, MPa	% Error
1	5000	86.94	96.28	9.7
2	5500	95.63	105.9	9.6
3	6000	104.32	115.54	9.7
4	6500	113.02	125.17	9.7
5	7000	121.71	134.79	9.7

Table 2: Validation

V. CONCLUSIONS

Major conclusions for present study are listed as below:

- From the study, it is concluded that the geometric parameters like fillet radius, tooth thickness, face width, and addendum circle radius plays an important role in deciding the beam strength of the tooth of spur gear and have significant influence on the output parameter for bending stress.
- On the other hand, material parameters like Young’s modulus, Poisson’s Ratio have a significant influence on for equivalent bending stress.

- Analytically calculated bending stress results have been validated using Finite Element Method with errors within 10%.

REFERENCES

[1] Christos A. Spitas and Vasilis A. Spitas, "Generating Interchange-able 200 Spur Gear Sets with Circular Fillets to Increase Load Carrying Capacity", International Conference on Gears, Vol. I, VDI-Verlag, Dusseldorf, Germany, July/August 2006, pp. 927-941.
[2] Spitas, Th. Costopoulos and C. Spitas, "Increasing the Strength of Standard Involute Gear Teeth with Novel Circular Root Fillet De-sign", American Journal of Applied Sciences, Science Publication 2005, pp. 1058-1064.
[3] M.Savage, K.L. Rubadeux and H.H.Coe, "Bending Strength Mod-el for Internal Spur Gear Teeth", Army Research Laboratory, Technical Memorandum San Diego, California, July 10-12, 1995.
[4] M. Koilraj, Dr. Muthuveerappan, Dr. J. Pattabiraman, "An Improvement in Gear Teeth Methodology Using Finite Element Method", IE Journal MC, Vol.88, October 2007.
[5] Zeping Wei: Stresses and Deflection in Involute Spur Gears by Finite Element Method, M.Sc. Thesis, University of Saskatchewan, Saskatchewan, 2004
[6] Y. Sandeep Kumar, R.K. Suresh, B.Jayachandraiah,Optimization of design based on Fillet radius and tooth width to minimize the stresses on the Spur Gear with FE Analysis, ISSN: 2277-3878, Volume-1, Issue-3, August 2012
[7] V.B. Bhandari, "Design of Machine Elements".
[8] Ashwini Joshi, Vijay Kumar Karma, Effect on Strength of Involute Spur Gear by Changing the Fillet Radius Using FEA, International Journal Of Scientific & Engineering Research Volume 2, Issue 9, September-2011 ISSN 2229-5518
[9] Miryam B. Sánchez, José I. Pedrero Miguel Pleguezuelos, Critical stress and load conditions for bending calculations of involute spur and helical gears, UNED, Department de Mechanical, Juan del Rosal 12, 28040 Madrid, Spain.
[10] El-Sayed S. Aziz1, Constantin Chassapis, Probabilistic Simulation Approach to Evaluate the Tooth-Root Strength of Spur Gears with FEM-Based Verification Production Engineering and Mechanical Design Department, Faculty of Engineering, Mansoura University, Mansoura, Egypt, Department of Mechanical Engineering, Stevens Institute of Technology, Hoboken, USA, November 15, 2011.