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Finite Element Analysis of High Contact Ratio Spur Gear and Taguchi Optimization of Gear Parameters

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Abstract: *This paper presents a design method and a vibration analysis of a carbon/epoxy composite drive shaft. The design of the composite drive shaft is divided into two Gears are one of the most critical components of mechanical systems, automobiles and rotating machineries. They vary from tiny size used in watches to very large size used in marine speed reducers, bridge lifting mechanism and railroad turn table drivers. Because of their high degree of reliability, efficiency and compactness gears are widely used as the power transmitting components in most of the mechanical system. Prototype testing a site reports suggest that gear contact ratio has significant effect on the generated noise level, especially in spur gear applications. High contact ratio (HCR) terminology is attributed to gear meshes that have at least two pairs of teeth in contact at all times ie., contact ratio of 2.0 or more. This helps in wider sharing of transmitted load leading to much smoother operation. Since minimum number of teeth pairs in contact are more than one in a HCR gear drive, the load sharing between the pairs is to be estimated accurately for economical design of the gear drive. It also improves structural efficiency, reliability and power to weight ratio. The detailed analysis of HCR gears is important because they are used with high load and large speed applications. This paper contains a finite element analysis of high contact ratio spur gear and taguchi analysis of high contact ratio gears.*
Keywords: *HCR\Spur\Taguchi*

I. Introduction

Gears are one of the most critical components of mechanical systems, automobiles and rotating machineries. They vary from tiny size used in watches to very large size used in marine speed reducers, bridge lifting mechanism and railroad turn table drivers. Because of their high degree of reliability, efficiency and compactness gears are widely used as the power transmitting components in most of the mechanical system.

Spur Gear is the simplest and most common type of gearing used for power transmission in the engineering world. Low-noise behavior in standard industrial gear units is becoming an important selection criterion and a factor indicating gear quality.

Prototype testing a site reports suggest that gear contact ratio has significant effect on the generated noise level, especially in spur gear applications. High contact ratio (HCR) terminology is attributed to gear meshes that have at least two pairs of teeth in contact at all times ie., contact ratio of 2.0 or more. This helps in wider sharing of transmitted load leading to much smoother operation.

Since minimum number of teeth pairs in contact are more than one in a HCR gear drive, the load sharing between the pairs is to be estimated accurately for economical design of the gear drive. It also improves structural efficiency, reliability and power to weight ratio. The detailed analysis of HCR gears is important because they are used with high load and large speed applications.

Gear profiles used in this work are involute profile to ensure conjugate action. Root fillet profile is the original trochoid profile which is generated by the sharp rack cutter during gear manufacturing process. The transmission efficiency of a precision gear system has to be very high, particularly the meshing efficiency of a pair of spur gears, reaching above 99.5%.

High contact ratio can be achieved by increasing the number of teeth, decreasing the pressure angle, decreasing the module and increasing the addendum factor. Previous researchers used the larger addendum to increase the contact ratio. Increase in addendum increases the number of teeth in contact thereby decreases the tooth contact stress. But this makes the tooth depth long and makes the tooth to be deformed easily. Profile modification for the tip relief should be considered to get full advantages of high contact ratio. Therefore, though, the increment in addendum can reduce the tooth contact stress, tooth root bending stress need not get reduced. In the present study higher contact ratio is achieved by varying the module, varying the number of teeth and by varying the pressure angle.

Numerical analysis is carried out using ANSYS finite element software to estimate the load sharing over the entire mesh cycle. Contact and root bending stresses are estimated over a complete mesh cycle. The study is repeated for different input loads.

The aim is to design a gear pair which can transmit maximum load with minimum contact stress and this requires optimization of gear parameters. In the present work, Taguchi optimization technique is used for the optimization of design variable. Here the objective function is to the contact stress by predicting the right combination of module, pressure angle and number of teeth. A confirmation test is also carried out with predicted gear parameters to validate the optimization result.

II. Finite Element Analysis

It is important to study the behavior of high contact ratio spur gear since they are mainly used in high speed and high load applications. Load sharing data, root stress and contact stress are the three important factors considered while designing the gears. Therefore the accurate estimation of these parameters is important to obtain an optimum design. The input torque considered for the present study are 50 Nm, 150 Nm and 350 Nm. The parameters uses in this study are given in the Table 2.1

Sl No	Parameters	Value Used
1	Module	3 mm
2	Number of teeth in pinion	40
3	Number of teeth in gear	60
4	Pressure angle	14.5°
5	Theoretical contact ratio	2.38
6	Centre to center distance	150 mm
7	Addendum	1.15*3 mm
8	Face width	25 mm

Table 2.1 Gear parameters

Coordinates of gear geometry is created using a C++ code. The generated coordinates are fed in to ANSYS finite element software in the form of text file to generate key points. Generated key points are connected with splines to create lines and these lines are used to generate areas. Entire gear hub and five teeth of gear and pinion are modeled. Quadratic 8 noded two dimensional plane 183 elements were used for finite element modeling. Plane strain condition is used under the assumption that the load distribution along the face width is uniform [11].

Coarse mesh is used to model gear hub region and very fine mesh with an element size of 0.05 mm is used to model the gear teeth region. The entire mesh model consists of 127180 elements and 380246 nodes shown in Fig. 2.1. The material properties of steel is used in the analysis are Young's modulus, E = 210GPa and Poisson's ratio μ = 0.3. Gear material is assumed to be homogeneous and effect of case hardening, case depth and other heat treatments are neglected.

Two cylindrical coordinate system are created one at the centre of the gear and other at centre of the pinion. Nodal coordinate systems of gear and pinion inner hubs are changed to these coordinate systems respectively for applying the boundary conditions. All the nodes in the inner hub of gear are completely constrained whereas all nodes in the inner hub of the pinion are constrained in radial direction. Tangential force is applied to the nodes situated on the inner hub of the pinion which contributing the input torque as per the Equation 4.1 [10]. Equation 4.1 gives the tangential force F to be applied to the nodes for contributing the input torque T.

$$\sum_i^n F_i \times R_h = T \quad (4.1)$$

Where T is the input torque, R_h is the radius of inner hub and F_i is the total tangential force applied to the i^{th} node situated on the inner hub radius of pinion.

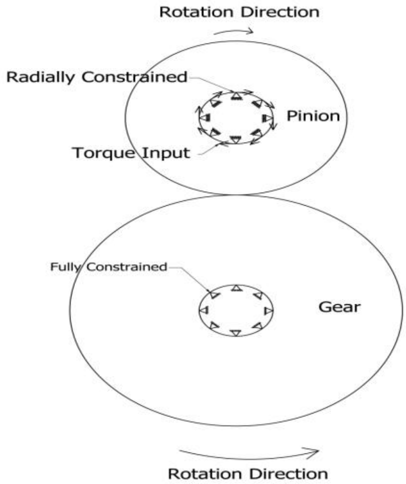


Fig. 2.2 Schematic diagram of gear with applied boundary conditions

Contact manager module was used to define contact between the mating pair of teeth. Both the bodies are considered flexible [11] and line to line contact was defined. For contact modeling, contact 172 and target 169 elements were used and close gap initial adjustment option was selected in order to avoid rigid body motion. Effect of Sliding friction between the mating gear teeth is neglected. The surface aspirates, waviness and all the manufacturing errors are neglected in this study.

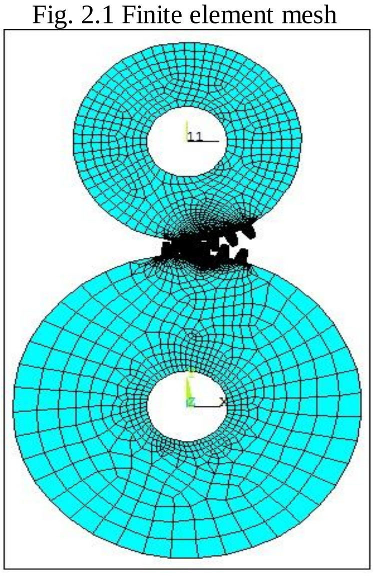
Non linear quasi static analysis was carried out with angular rotation increment $\theta = 0.5°$ given to the pinion and the gear is rotated according to the gear ratio. Analysis is done for complete mesh cycle. The result obtained are post processed to estimate load sharing ratio, contact stress and root stress for each rotation position and plotted against rotation angle.

III. Optimization Of Parameter

Taguchi optimization technique is used to minimize contact stress. The gear parameters selected for the study are Addendum ratio, Pressure angle and Number of teeth. For each parameter three levels are selected. Parameters and their selected levels for the present analysis are given in Table 3.1

Based on the number of factors and levels, a suitable Taguchi orthogonal array for the experiments is selected by using MINITAB statistical software. Since there are three parameters with three levels each, L_9 orthogonal array is chosen and the experiment table corresponding to L_9 array is given in Table 3.2

Fig. 2.1 Finite element mesh



Parameters	Level

	1	**2**	**3**
Addendum Ratio	1	1.1	1.15
Pressure angle	14.5°	20.0°	22.5°
Number of teeth	32	40	48

Table 3.1 Parameters and their levels

Exp.No	Level		
	Addendum Ratio	Pressure angle	Number of teeth
1	1	1	1
2	1	2	2
3	1	3	3
4	2	1	2
5	2	2	3
6	2	3	1
7	3	1	3
8	3	2	1
9	3	3	2

Table 3.2 L_9 orthogonal array experiment table.

Quality response	**S/N Ratio Formula**
Nominal the best	$S/N = 10\log\frac{Y^2}{S^2}$
Larger is better(maximize)	$S/N = -10\log\frac{1}{N}\sum\frac{1}{Y^2}$
Smaller the better (minimize)	$S/N = -10\log\frac{1}{N}\sum Y_i^2$

Table 3.3 Recommended Performance Statistics

Nine finite element analyses are performed as per the above experiment table. The maximum contact stress developed in experiment is estimated and Tabulated in Table 3.4. For each experiment in the orthogonal array, signal to noise (S/N) ratio is calculated. The quality response is of three main types, namely the larger the better (LTB), the smaller the better (STB) and the nominal the best (NTB). Equations used to calculate S/N ratio for each quality response is given in Table 3.3.

Where $\bar{Y}$ is the calculated average of measured characteristic value and S is the corresponding standard deviation. Y_i is the measured characteristic value in the i^{th} measurement. N denotes the number of measurements.

The S/N ratios are calculated based on STB criterion for each experiment and tabulated in the Table 3.4. In the present study contact stress is the only one measured characteristic contact stress and N = 1.

Exp. No	**Contact tress (MPa)**	**S/N Ratio**
1	398.6813	-52.0125
2	418.96	-52.4435
3	253.4312	-48.0772
4	313.7933	-49.9329
5	252.605	-48.0488
6	377.3925	-51.5359
7	271.59	-48.6783
8	355.8133	-51.0244
9	293.3734	-49.3484

Table 3.4 Table of orthogonal array with response

IV. Result And Discussion- Fea

This study analyzes the response of high contact ratio spur gears under different input loads. The input loads used are 50 Nm, 150 Nm and 350 Nm. Quasi static analysis over a complete mesh cycle is carried out and the results obtained are discussed below. Contour plot of von Mises stress (refer Fig.4.1) illustrates the stress distribution occurring in the contact zone. Higher stress observed at the root fillet regions of the mating teeth are due to bending action as well as adjacent areas. Maximum Stress is observed at the contact zone and the stress decreases as distance from contact zone increases. For all input loads, maximum stress is observed at the same location for all angular positions during the entire mesh cycle

Load Sharing Ratio

The nodal resultant force at each node in the contact region is obtained using ANSYS post processing module for each individual gear tooth [11]. Maximum value of load sharing ratio is found to be 0.55, only 55% of the total load transferred is taken by a teeth. Load sharing ratio curve follows almost same behavior for different input loads (Fig. 4.2) because there is not much change is observed in working contact ratio as the load sharing behavior purely depends on the working contact ratio. Input loading conditions used in this study do not affect the contact ratio but at higher loads due to the increase in deformation, working contact ratio may slightly increase. At triple teeth contact zone load sharing ratio decreases to a value around 30% and at the tip region load sharing decreases to around 20% which is an advantage because the contact stress and root bending stress are observed to be minimum during this zone.

Contact Stress Variation

Contact stress is estimated as the ratio of load transferred through contact and the area of contact and variation of contact stress developed with rotation angle is presented in Fig. 4.3 [11]. It is observed that at initial point of contact and at the point of contact of the tooth tip, contact stress rises to maximum values owing to less contact areas available. There was an upward shift of contact stress as the input load increases. At double teeth contact zone the contact stress is observed to increase in value. At 50 Nm load, maximum value of contact stress estimated is 105MPa. Maximum value of contact stress obtained at 350 Nm is 675 MPa. Maximum contact stress observed at the tip zone as the contact area available at the tip zone is small.

Maximum Root Bending Stress

Maximum root stress is estimated from the maximum value of first principle stress and the validation of root stress

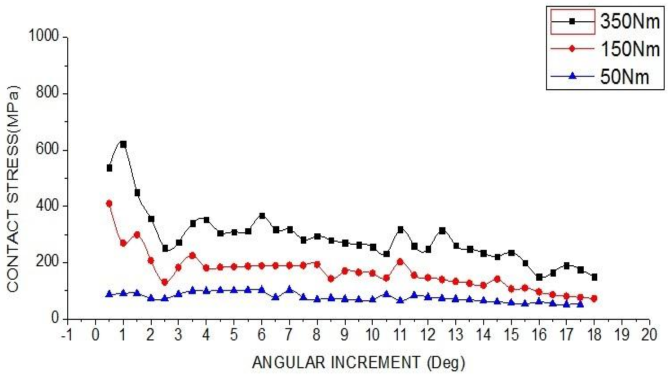


Fig 4.3 Contact stress variation over a complete mesh cycle with different input loads

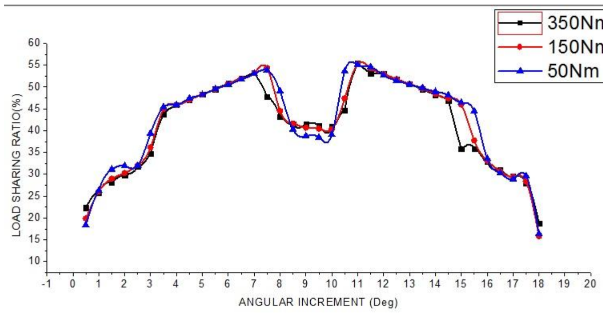


Fig. 4.2 Load sharing ratio over a complete mesh cycle with different input loads

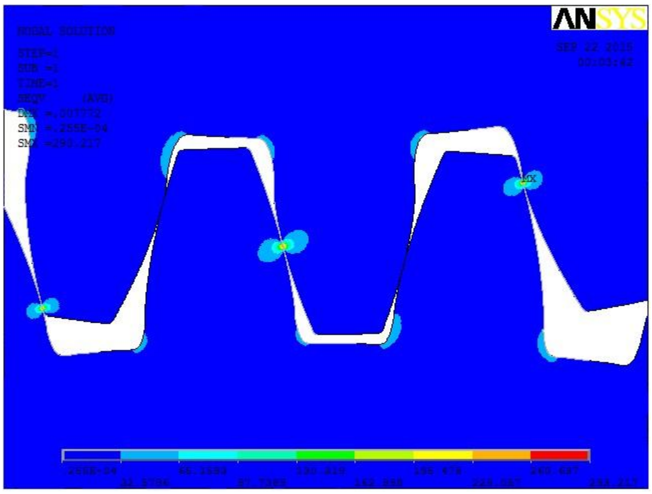


Fig. 4.1 Contour plot of von Mises stress developed at an intermediate position

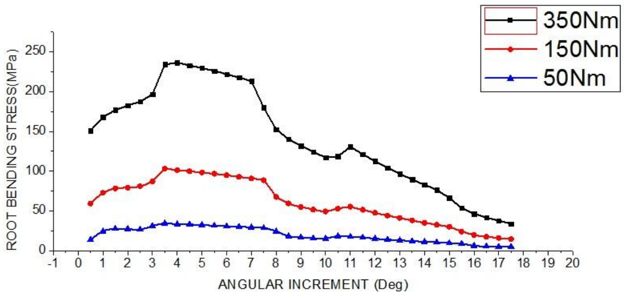


Fig 4.4 Root bending stress over a complete mesh cycle for different input loads

With rotation angle is presented in Fig. 4.4 for different input loads.

Root stress is observed to be minimum at contact closer to the root and gradually increases as the position of contact moves to the tooth tip. The graph shows a decrease of root bending stress value in triple tooth contact zone and increases as contact changes to double tooth contact zone. Bending stress is found to increase proportionally with increase in input load.

Numerical Analysis And Optimization

Effect of gear parameters on contact stress was studied. The FEM results of contact stress were transferred to S/N ratio. The 9 S/N ratios were tabulated in the Table 4.1. Statistical software MINITAB is used for calculating mean and S/N ratio by using the STB criterion. Then the graph is plotted as shown in the Fig 4.5 & 4.6. According to Taguchi's idea, maximizing signal to noise ratio will get maximum robustness. The main effects of all control variables are obtained from the graph. From the graph the level of each parameter with maximum value of signal to noise ratio is selected [13]. The combination obtained is as follows, Addendum Ratio = 1.15, pressure angle = 22.5° and number of teeth = 48

Exp. Run	Contact Stress (MPa)	S/N Ratio
1	398.6813	-52.0125
2	418.96	-52.4435
3	253.4312	-48.0772
4	313.7933	-49.9329
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9	293.3734	-49.3484

Table 4.1 Table of orthogonal array with response and their respective S/N ratio

FACTORS	LEVEL (L)			D = Max(L)-Min(L)	Rank
	1	2	3		
ADD RATIO	-50.84	-49.83	-49.68	1.1607	2
PR ANGLE	-50.20	-50.50	-49.65	0.8518	3
NO. OF TEETH	-51.52	-50.57	-48.26	3.2562	1

Table 4.2 Average S/N Ratio for Different Factor Level

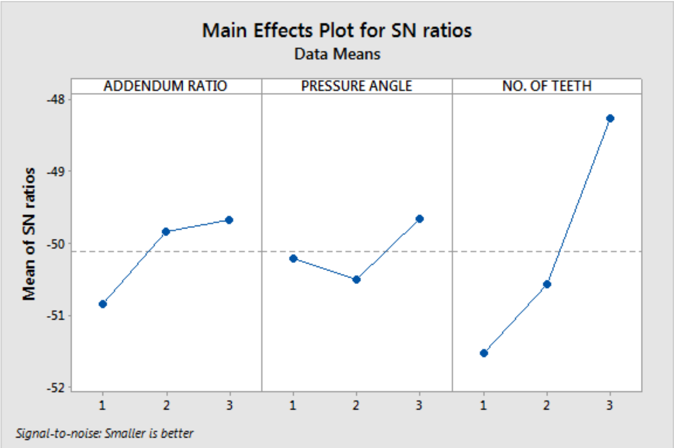

Fig. 4.5 Main effects plot for S/N ratio

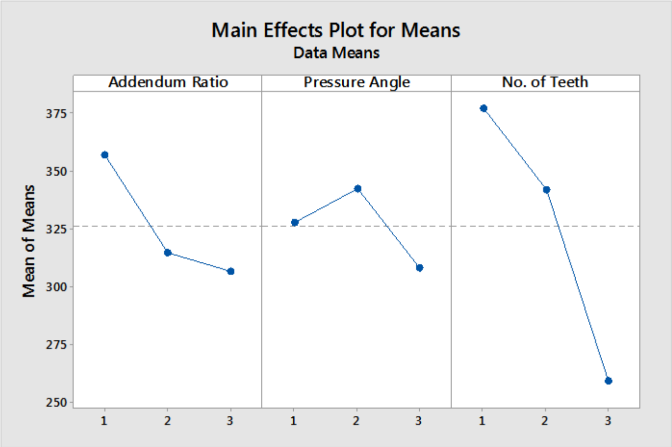

Fig. 4.6 Main effects plot for means

Confirmation Test

After conducting Taguchi analysis the optimum conditions are obtained. By using these optimum process parameters the result should be verified. Numerical analysis are carried out by using Addendum ratio = 1.15, Pressure angle = 22.5° and number of teeth = 48. The contact stress distribution during the confirmation test is shown in Fig. 4.9 The maximum contact stress obtained for this combination was 247.28MPa which is smaller than the 9 combinations used in the Taguchi orthogonal array

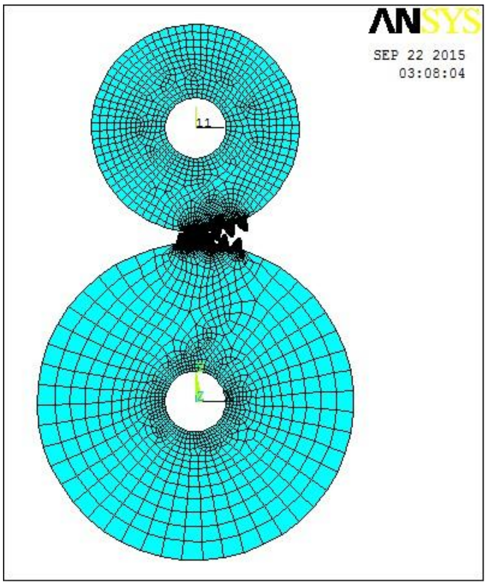

Fig. 4.7 Finite element meshed model for confirmation test

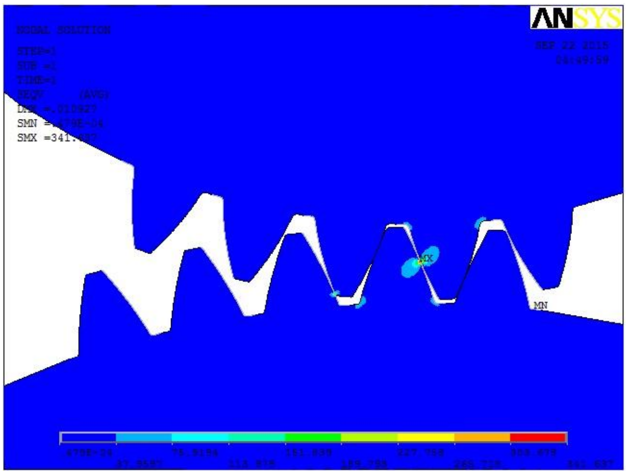

Fig. 4.8 von Mises stress contour of the gear teeth for confirmation test

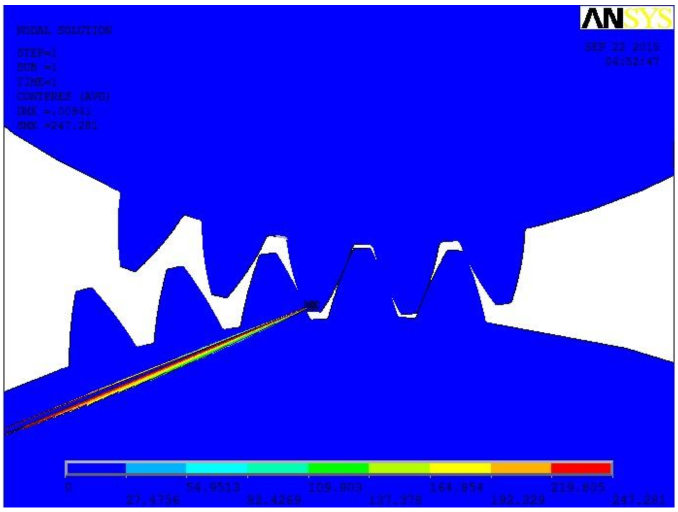

Fig. 4.9 Contact Stress of the gear teeth

ANOVA Analysis

Response variable	Degrees Of Freedom	Sum Of Squares	Percentage Effect of Each Variable On Response
Addendum Ratio	2	2.381780512	10.37%
Pressure Angle	2	1.121102386	4.88%
Number Of Teeth	2	16.82523044	73.23%
Error	2	2.646216203	
Total	8	22.97432954	

Table 4.3 Percentage effect of response variables on the response

ANOVA estimates three sample variances, a total variance based on all the observation deviations from the grand mean, an error variance based on all the observation deviations from their appropriate treatment means and a treatment variance. The treatment variance is based on the deviations of treatment means from the grand mean, the result being multiplied by the number of observations in each treatment to account for the difference between the variance of observations and the variance of means The fundamental technique is a partitioning of the total sum of squares *SS* into components related control variables to the effects used in the model.

ANOVA analysis was carried out to identify the affect of each variable on the contact stress. Result of the ANOVA analysis given in Table 4.3 says that Number of teeth has a dominant effect with a contribution of 73.3%. Addendum ratio has an effect of 10.37% Pressure angle has an effect of 4.88%.

Mesh Convergence Study

Mesh convergence study is conducted by varying number of elements in contact zone. The result of mesh convergence study is given is Fig. 4.10 Mesh convergence study started with element size 0.2 mm at the area of contact and the variation in Von-Mises stress is found to be negligibly small from 125000 elements onwards that correspond to an element size of 0.05mm at the area of contact.

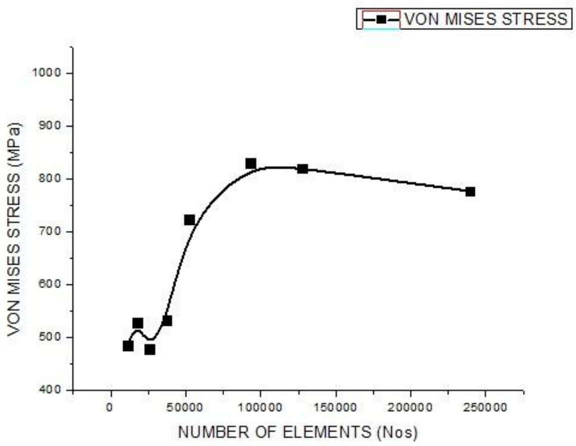

Fig. 4.10 Mesh convergence study of maximum von Mises stress at contact region

V. Conclusions

High contact ratio spur gear with involute tooth profile and trochoid root fillet profile is analyzed numerically. In the present study high contact ratio is achieved by varying addendum ratio, number of teeth and pressure angle.

Quasi-static analysis is carried out on high contact ratio gears for the complete mesh cycle under different input loads. Variations of load sharing ratio, contact stress and root bending stress with rotation angle are studied for various input loads. Load sharing ratio did not vary much with input load, confirming the fact that behavior is depends more on the working contact ratio of the mating gears. For the selected input loads the deformations observed is quite small leading to not much changes in the contact ratio.

Contact stress and root bending stress developed exhibit an upward shift as the load increases as expected. Increase in contact stress is smaller at higher load level compared to that at lower load levels. This is because of the increase in contact area developed at higher load levels, which is evident from increased number of interacting nodes at higher loads.

Taguchi analysis is carried out to optimize the control variables for minimum contact stress. The optimum values of control variables are Addendum Ratio = 1.15 pressure angle = 22.5^0 and number of teeth = 48. The confirmation test estimated a contact stress 247.28 MPa which is lower than values obtained in all the nine experiments conducted, there by validate the optimization results. ANOVA analysis predicted that No. of teeth has 73.3 % effect on the contact stress developed. This is evident from the fact that as the no. of teeth increases, the gear pitch circle diameter also increases, leading to smaller contact and bending stress.

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