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Analysis of Influence of Tooth Depth in Spur Gear Vibration

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Abstract— Gears are one among the most vibrating structure in the machine elements. Gear vibration control is one of the challenging tasks for the maintenance engineers. Modern Gear Design concept introduces several methods for controlling the vibration in gears in the design desk. Gear vibration can be reduced by profile modification is one of the way, specifically in the root of the gear. In this work, the tooth depth is increased to identify the reduction of vibration in a spur gear. A two tooth model has been generated, discretized and analysed by FEA software. The amplitude of vibration of the tooth model has been done under harmonic analysis to identify the displacement and acceleration of the gear tooth due to transmitted (tangential) and radial load. Analysis have been done for the existing tooth model (TOOTH A) and the modified one (TOOTH B). The result shows that the modified tooth has a better control in the displacement and acceleration. Hence an increase in the tooth depth within certain limit ensures the control of magnitude of the vibration.

Index Terms— Spur Gear, Tooth Depth, Vibration Analysis, FEA Software, Harmonic Response, Discretization, Three Dimensional Analysis.

1 INTRODUCTION

VIBRATION is one of the pre-dominant factors for the failures in rotating machineries. Especially, in the spur gear power transmission system, for a heavy load transmission, vibration is one of the important parameter to be maintained within the limit of control. A vibration in spur gear happens due to several reasons such as errors (i.e., composite errors which includes Static and Dynamic Transmission Error), tooth stiffness variation, frictional force on the tooth and the impulse on the pitch point. Apart from those, cinematic error is also being considered for vibrations which can be reduced by means of base pitch and gear profile modification. Modifying the circular pitch is also one of the ways of reducing the vibrations. The vibration in spur gear can also be reduced by increasing the contact ratio, decreasing the pressure angle, decreasing the dynamic load, increasing the tooth height. Many research works has been carried out in controlling the gear vibrations by tip relief and root relief through crowning. Crowning has been done axially or radially along the tooth to reduce the contact stresses and also to reduce the vibration.

Vijaya Kumar Ambarisha,& Robert G. Parker [01] examines the complex, nonlinear dynamic behavior of spur planetary gears using two models: (i) a lumped-parameter model, and (ii) a finite element model. The two-dimensional (2D) lumped-parameter model represents the gears as lumped inertias, the gear meshes as nonlinear springs with tooth contact loss and periodically varying stiffness due to changing tooth contact conditions, and the supports as linear springs. These comparisons validate the effectiveness of the lumped-parameter model to simulate the dynamics of planetary gears.

Shunting Li [02] has done a fundamental study on resonance frequency behavior of three-dimensional, thin-walled spur gears from experimental tests and finite element analyses. Strain phase method is presented in his work to identify the resonance mode shapes of the thin-walled gears when they are running in a complete resonance state. He found that when wall thickness of the gear is increased, the2nd, the 3rd and the 4th order bending vibrations of the web disappears and only bending vibrations of the rim remains. He also found that rim and web thickness have greater effects on structural frequencies of the thin-walled gears than other parameters. Also Tooth module can also exerts an effect on the natural frequencies of the thin-walled gears. He concludes that, with the increments of web and rim thickness, all the natural frequencies of the thin-walled gear become higher and higher. Gear module, web position and gear face width also affect structural frequencies of the thin walled gear, but the effects of these parameters are not so greater than the rim and web thickness.

Hong-Sen Yan, Ta-Shi Lai,[03] presents a concept of elementary gear trains such that the tooth-profiles of the pinion are cylindrical. He has worked out the surface equation of proposed novel elementary planetary gear trains with cylindrical tooth-profiles by using the triple vector product of differential geometry to deal with the equation of meshing. When the cross-section of the tooth is round, for convenience of geometric analysis, the pitch radius is replaced by gear radius that is the distance between the gear center and the tooth center. The mathematical expressions of the envelope equations are applied to design the internal elementary planetary gear trains in which the pinion is with cylindrical tooth-profiles.

All these above studies shows that vibration analysis have been done in various aspects and in parameters, but only a little attention has been given in identifying the influence of the tooth depth of a gear for vibration analysis. However major works in the research of gears are carried out in fault diagnosis of the gear box and methods to detect the faults. Only few researches have been made in the vibration analysis by profile modification. Hence an effort has been taken to study the behaviour of a gear having an increased tooth depth.

PARAMETER SELECTION:

An EN24 Steel material has been selected for a spur gear having 36 teeth with the pitch circle diameter of 114.3mm. The face width of the gear is 50.8mm, which is more satisfactory to transfer the power of 11.2kW with a speed of 1440rpm. The

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velocity ratio of the gear is 1. The gear is designed based on the Lewis and Buckingham Equation model to find out the suitability of the gear for the wear load capacity and beam strength of the gear. The proportions of addendum, dedendum and the working depth have been finalized based on the AGMA Standards. Based on radial and tangential loading, cyclic in nature, the analysis has been started with a frequency domain.

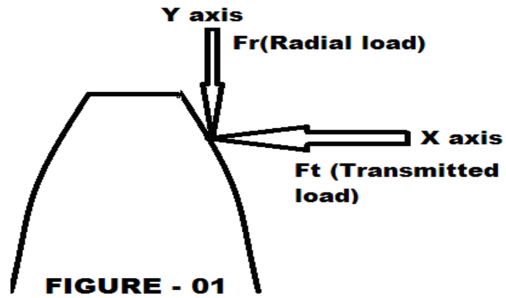


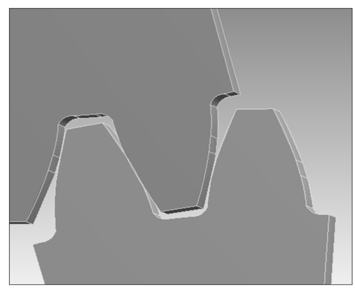
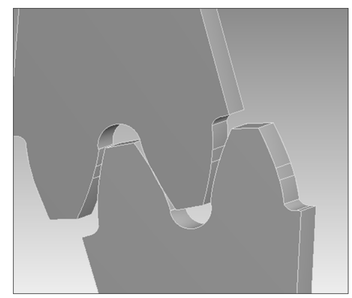
FIGURE - 01

MODELLING AND ANALYSIS OF A GEAR TOOTH:

Gear tooth is modeled as a pair of three dimensional two tooth model by using modeling software and it has been discretized and analyzed by using a FEA Software. Since the gear is operated at a speed of 24rps, the fundamental meshing frequency or gear mesh frequency (GMF) has been finalized as the product of number of teeth and the speed i.e. 864Hz. The transmitted force of 1298N and the radial load of 473N (as shown in figure – 01) are considered as a load for a single tooth, which has been given as force components on the face width and flank of the gear with a cylindrical support on the centre of the gear.

Initially, the gear tooth is analyzed under modal analysis to find out the natural frequency and then the tooth is subjected to frequency domain harmonic analysis with the frequency range of 20Hz to 2500Hz with the solution interval of 20. Natural frequency of the system can be analyzed by considering the gear tooth has a cantilever beam, where $\omega n = \sqrt{\frac{k}{m}}$, where k= $\frac{3EI}{l^3}$, is the stiffness and m is the mass of the gear tooth. The radial load implies the load on the axial rod (i.e. from the tip to the root of the tooth), which is having more stiffness and can be derived as k = $\frac{AE}{l}\begin{bmatrix} 1 & -1 \\ -1 & 1 \end{bmatrix}$. However the stiffness is more in the case of radial load direction, therefore the influence of load for vibration in this direction will be less. Since the stiffness is more, the natural frequency will also be more. Hence the major amplitude of vibration (i.e. displacement, velocity and acceleration) are to be considered only for the transmitted load or tangential load. For the low frequency of vibration, the amplitude of vibration may be considered majorly for displacement. For a very high frequency of vibration, amplitude of acceleration will be certainly high. For a medium frequency, the amplitude of vibration is to be analyzed in both the aspects i.e. displacement and acceleration. Generally the level of vibration is being considered as in control if the displacement is within 2 mills (i.e. 50.79 microns).

Both the gear tooth pairs are analyzed with same loading conditions and with the same parameters. The difference in the modified gear tooth (TOOTH B as shown in figure – 02) is a small recess has been taken for a depth of 1.3mm throughout the face width at the root of the gear. A convex groove made in the root, protrudes the mass to be the influencing parameter since it tends to increase the length. Whereas the stiffness is still safe since the natural frequency is 14338rad/sec which is very high when comparing with the forcing frequency of 5429rad/sec.

TOOTH A	TOOTH B

FIGURE - 02

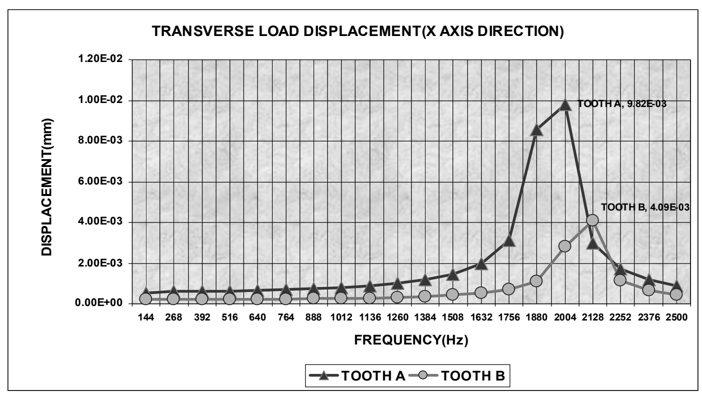


FIGURE - 03

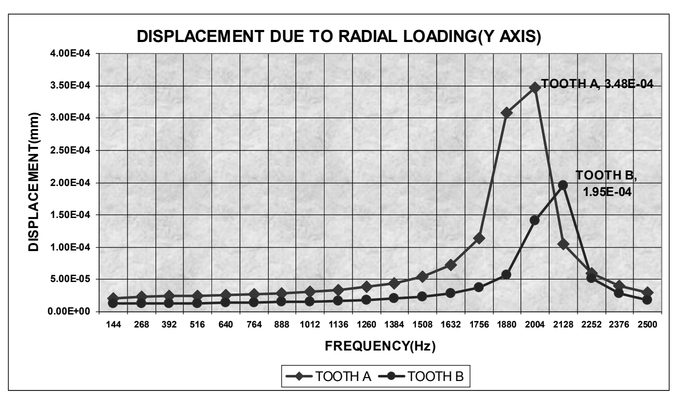


FIGURE – 04

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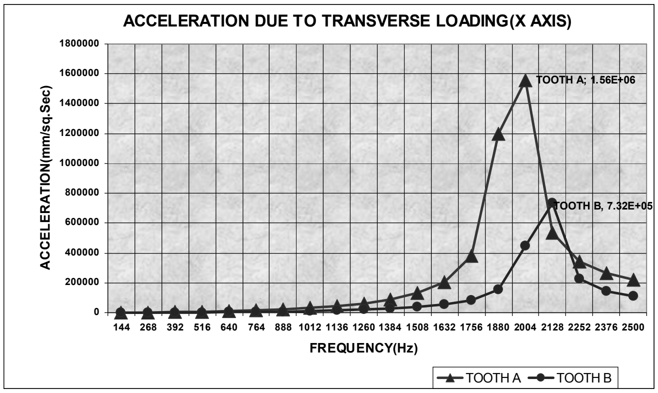
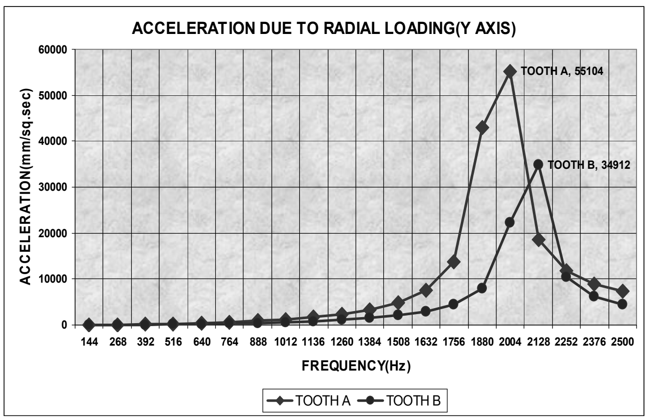


FIGURE – 05



FREQ (Hz)	DISPLACEMENT(mm)			
	X AXIS		Y AXIS	
	TOOTH A	TOOTH B	TOOTH A	TOOTH B
144	5.36E-04	2.01E-04	2.11E-05	1.30E-05
268	6.04E-04	2.04E-04	2.37E-05	1.31E-05
392	6.17E-04	2.08E-04	2.42E-05	1.33E-05
516	6.36E-04	2.13E-04	2.48E-05	1.36E-05
640	6.63E-04	2.21E-04	2.58E-05	1.40E-05
764	6.98E-04	2.31E-04	2.71E-05	1.45E-05
888	7.45E-04	2.45E-04	2.87E-05	1.51E-05
1012	8.08E-04	2.62E-04	3.10E-05	1.60E-05
1136	8.93E-04	2.85E-04	3.40E-05	1.71E-05
1260	1.01E-03	3.16E-04	3.82E-05	1.86E-05
1384	1.19E-03	3.59E-04	4.45E-05	2.07E-05
1508	1.47E-03	4.22E-04	5.44E-05	2.37E-05

FREQ (Hz)	ACCELERATION(mm/s^2)			
	X AXIS		Y AXIS	
	TOOTH A	TOOTH B	TOOT H A	TOOTH B
144	438.64	164.81	17.269	10.64
268	1711.7	577.68	67.143	37.188
392	3743.2	1260.1	146.54	80.735
516	6689.7	2243.6	261.18	142.82
640	10721	3577.4	417.09	225.83
764	16094	5334.3	623.48	333.3
888	23204	7622.4	894.39	470.49
1012	32668	10604	1251.8	645.28
1136	45494	14532	1731.7	869.9
1260	63428	19814	2396.2	1164.2
1384	89767	27154	3362.8	1562.3
1508	1.32E^{05}	37871	4881.9	2128.1

INFERENCE OF THE ANALYSIS:

Based on the harmonic analysis, the tangential load applied on the gear tooth shows that the magnitude of displacement of the tooth A is 9.82 microns(in the X Direction) at the frequency of 2004Hz, which is more when comparing with the modified tooth B, which is having a displacement of 4.09microns. Hence the displacement is being reduced to more than 50%. But still both the displacement levels are within the controlled limit of vibration. The figure-03 shows that the tooth B is a better selection for vibration damping even at lower speeds, which is more effective in damping for the entire frequency range. In the case of radial load direction (i.e. Y axis direction), the results are more encouraging that the modified tooth B is conveniently settles at the maximum displacement of 0.195 microns, which is very less when comparing with the tooth A(figure-04). the displacement of the tooth A is 0.348microns, which is one the important parameter for the failure of shaft due to bending. The displacement in the radial direction may not affect the gear, but the cyclic stress produced in this direction leads to gear shaft failure. Hence the Tooth B has a major role in reducing the failure due to radial load.

Since only two teeth have been generated for our analysis, the influence of the remaining portion of the gear is not being concluded. Hence the amplitude of acceleration resembles to be more at this level for a gear analysis. Due to the loading condition specified in the contact region of the analysis, the acceleration is to be done for a more rough condition. Hence the contacting surfaces of the two gears are mentioned as frictional in the FEA software with the co-efficient of friction as 0.03 and also the stiffness is updated for every sub step of the analysis. This updating of stiffness for every sub step can make the analysis for a betterment of prediction. Based on this idea, we found that the gear tooth A has an amplitude of acceleration has 132m/s^2 (table – 01). In the case of tooth B (as shown in figure-05), the acceleration is maximum of 38m/s^2 at a frequency of 1500Hz. Hence the acceleration due to the tangential load is controlled up to 75%. In the case of radial load

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direction i.e., Y axis direction, the acceleration of the tooth B is 2.12m/s^2 which is very less comparing to the tooth A which is 4.8m/s^2. Acceleration parameter analysis intends that the gear tooth B is more compromising than tooth A for the reduction of vibration. The output of the analysis shows as in table-01, that the Existing model of tooth A attains its maximum Amplitude of vibrations at 2004Hz. whereas the modified tooth model B attains the highest value at 2128Hz.(both in displacement and acceleration). But since the meshing frequency is only 864Hz, the acceleration is need to be considered at the maximum of 1500Hz.

CONCLUSION:

Gear tooth profile modification can be done by increasing the tooth depth which influences much in reducing the gear vibration at medium frequency range. Hence it is concluded that the influence of the tooth depth is a considerable parameter in vibration analysis of a gear. Further this analysis can be extended to identify the gear rattling effect and also the residual stress reduction by varying the tooth depth. Suitable progression ratio of the tooth depth can also be identified for a better working condition of the gear for various applications.

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