Optimum Design of Gear Drive for Non Linear Contact and Dynamic Analysis by Using Finite Element Method

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Abstract - *This study focuses on the reduction of stresses that occur on Spur gear by means of different stress relief features. The Bending stress has to be analyzed by means of the analytical and FEA method then further we will try to reduce them by some material removed from the gear. Some advanced optimization techniques are also employed in order to get the refine data's on the basis of previously generated stress relief features. Then the result of the optimum solution applied on shaft misalign condition means applied to nonlinear contact also we will check and compare dynamic results. In this dissertation work, it is proposed to carry out the contact stress analysis and dynamic analysis of either gear pair used in an industrial system using finite element method (FEM) and analyzes optimum solution for it.*

Key Words: Spur gear, hertz-contact stress, bending stress, finite element analysis, contact stress, Dynamic Load

1. INTRODUCTION

The increasing demand for smooth power transmission in machines, vehicles, elevators, and generators, has created more demand for a more precise analysis of the characteristics of gear systems. In the automobile industry, the use of gears with high reliability and lighter weight is necessary as lighter automobiles continue to be in demand. The success in engine noise reduction promotes the production of smoother gear pairs for further noise reduction. Finally, the effective way of noise reduction is to reduce the vibration associated with them. Designing highly loaded spur gears in power transmission systems that need to be strong and smooth. To achieve these results, analysis methods can easily implemented and also it provide information of contact and bending stresses, along with transmission errors. The FEA Analysis is capable of providing this information.

There is a requirement to design a single-stage speed reduction spur gear drive for the food mixer application with an input power of 1.5 Hp and at 800 rpm. The speed ratio is 1.5:1 and the minimum number of teeth on the pinion is 18. The gear tooth is involute in profile having a 20-degree pressure angle. The starting torque is 150% of the rated torque of the motor. The FOS (factor of safety) can be taken as 1.5. The gears are made of plain carbon steel having Sut =600 N/sq.mm.

2. LITERATURE SURVEY

Deshpande [1] has carried out finite element analysis (FEA) of spur gear for determining the transmission error. FEA results have been obtained by using map-mesh with contacts using quad (2D) elements in ANSYS. The mesh characteristics have been studied in detail in the handover region as it affects the vibration and noise drastically

Yong-Jun Wu et al. [2] has been observed that the method is not only effective in designing and evaluating the tooth profile modification, but also in studied the dynamic meshing characteristics of continuous engaged gear drives with realistic time-varying meshing stiffness and tooth sliding friction.

Jong Boon Ooi et al. [3] studied modal analysis of portal axle has been simulated using finite element method FEM on three different combinations of gear train system commonly designed for portal axle. FEM static stress analysis has been also simulated on the same three different gear trains to study the gear teeth bending stress and contact stress behavior of the gear trains in different angular positions from 0° to 18°.

Yong-Jun Wu et al. [4] in another paper presented the dynamic contact analysis using FEM and its experimental validation for helical gear pair. In this paper, a precise tooth profile modification (TPM) approach of the helical gear pairs has been presented first.

Atanasovska et al. [5] have presented FEM analysis for stress analysis and nonlinear contact analysis of helical gears. In this paper development of the finite element model for simultaneously monitoring the deformation and stress state of teeth flanks, teeth fillets and parts of helical gears during the tooth pair meshing period have been described.

Jain and Parker [6] have presented the effects of mesh stiffness parameters; including stiffness variation amplitudes, mesh frequencies, contact ratios, and mesh phasing; on these instabilities have been analytically identified.

Jeong [7] has investigated the nonlinear behavior of gear pairs considering hydro dynamic lubrication and sliding friction. An analysis of the nonlinear behavior of gear pairs according to the direct contact elastic deformation model over a wide range of speeds, considering the hydrodynamic effects and friction force have been described in this paper.

Kahraman and Blankenship [8] have been investigated experimental study by measuring the dynamic transmission error of several gear pairs using a specially designed gear test rig. A simplified analytical model also has been proposed which accurately describes the effects of involutes contact ratio on dynamic transmission error.

Ognjanovic and Agemi [9] have studied gear vibration measurements and frequency analysis (FFT-Analysis) has been performed in very high speeds of gear rotations as high as 40,000 rpm. A mathematical model for experimental results synthesis has been established.

Podzharov et al. [10] have presented a simple method to design spur gears with a contact ratio nearly 2.0 has been used. It consists of increasing the number of teeth on mating gears and simultaneously introducing negative profile shift in order to provide the same center distance.

Abhishek Tiwari. [11] have studied a design and optimization of Spur gear bending stress and related fatigue life. By the means of stress relief features we can reduce the bending stress and increase the fatigue life of spur gear, but along with this they discussed about the optimization of the Bending stress and fatigue life with respect to the size and location parameters of the stress relief holes.

3. EXPERIMENTAL DETAILS

Experimental modal analysis will be carried out to find the natural frequencies and the mode shapes of gear pair using Fast Fourier Transform Analyzer (FFT). The experimental results will be co-related with the results obtained by FEM analysis.

Experimental set up will be designed and developed as shown in figure 3.1 for the dynamic analysis of gears.

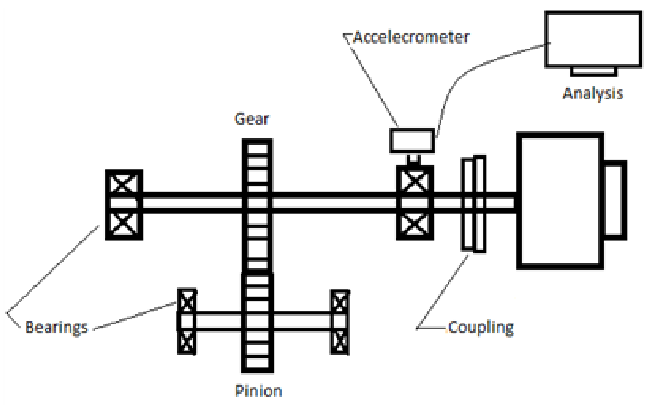


Figure 3.1: Experimental Set Up

Test set up is proposed to induce a defect on a particular tooth of a chosen gear in the gear box & generate the vibrations. To sense the vibration signal generated by the gearbox an accelerometer has been used. In order to process the vibration signal sensed by the accelerometer on FFT Analyzer has been selected. With the use of experimental set up we can identify the natural frequencies for the component using the principle of resonance.

4. STATIC STRESS ANALYSIS OF GEAR PAIR USING FINITE ELEMENT METHOD

In the present analysis, meshing was carried out in ANSYS Workbench 15.0 software. There are two methods available for meshing in this software viz. free meshing and mapped meshing. For mapped meshing, the element must be a brick element (a solid element with straight edges) rather than a solid187 element. Again area of cross-section should be uniform for mapped meshing. For the case of gears, both types of meshing are possible in this analysis. Brick meshing can be performed by using the SOLID95 element. But as stated earlier, this element has straight edges so SOLID187 approximates the curved edges of the gears more accurately and precisely than SOLID95, and hence free meshing with SOLID187 element has been considered as shown in fig 4.1 for the convergence criteria.

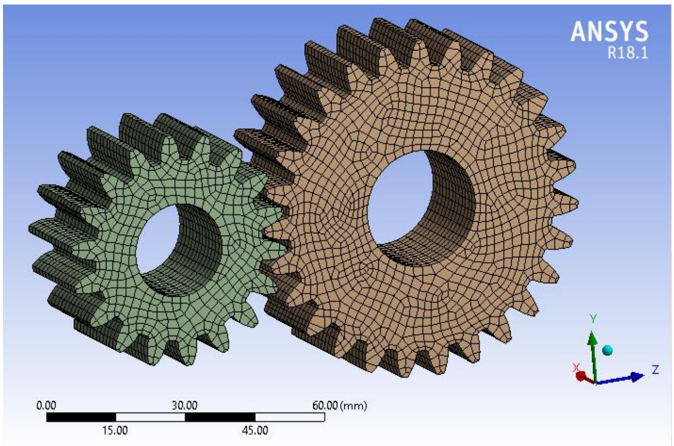


Figure 4.1: Meshed Model of Gear Pair with SOLID 187Element

Table-1: Material Property

Property	**Value**
Modulus of Elasticity	2 X E5 *N/mm*
Poisson's Ratio	0.30
Density	0.7850 X E-9 *Tonns/mm*

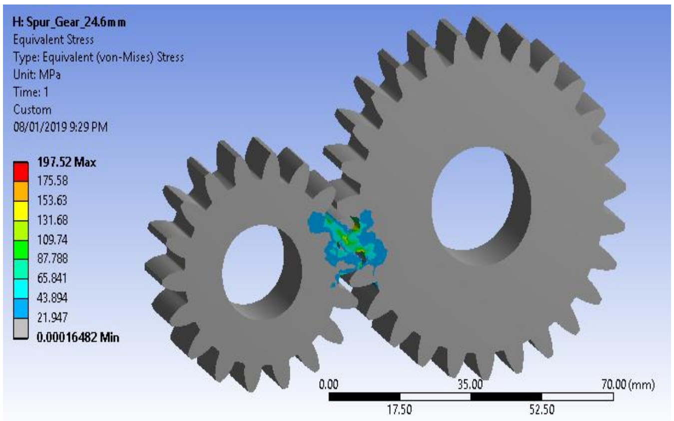


Figure 4.2: Bending Stress Counter

The maximum stress value of equivalent stress obtained is 197.5 Mpa which is at the root of the fillet as shown in fig 4.2.

5. NON LINEAR CONTACT ANALYSIS OF GEAR PAIR USING FINITE ELEMENT METHOD (FEM)

For contact stress analysis between two interacting gear teeth, frictional contact is assigned. The coefficient of friction of the contacting gear tooth surface was set to 0.1. In the ANSYS contact settings, Augmented Lagrange was selected as the solver for the contact non-linearity problem. To create a contact surface the input gear contact surface is defined as the 'Contact' and the contact surface of the output gear was the 'Target' of the contact.

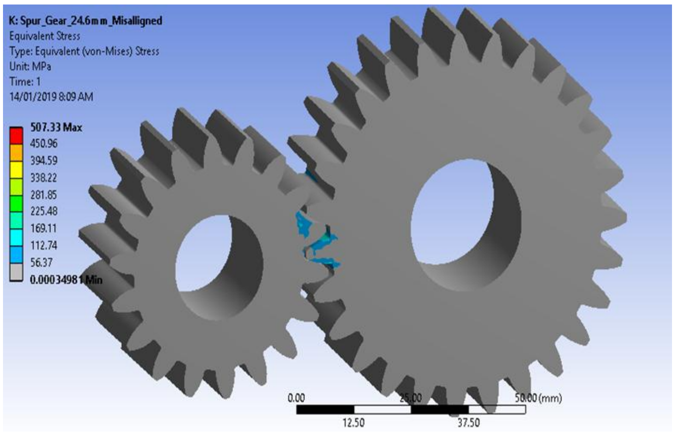


Figure 5.1: Bending Stress Contour of Nonlinear contact analysis

Fig 5.1 shows Equivalent Von Misses stress contour of nonlinear contact analysis, the maximum stress is at contact surface between gears and pinion its magnitude is 507.33 Mpa.

6. STRESS OPTIMIZATION

6.1 LINEAR CONTACT ANALYSIS

We can further move on to achieve the objective of reduction of stress at spur gear tooth. We are further taking some cases where we are going to insert some stress relief feature in order to reduce the stresses.

6.1.1 CASE 1: CIRCULAR GROOVE AT ROOT

The first case we are taking is the, one circular groove at the root location. For this case we have selected one random hole with some specific size and location to observe the result of it; whether it reduces the stress and increases the life or not.

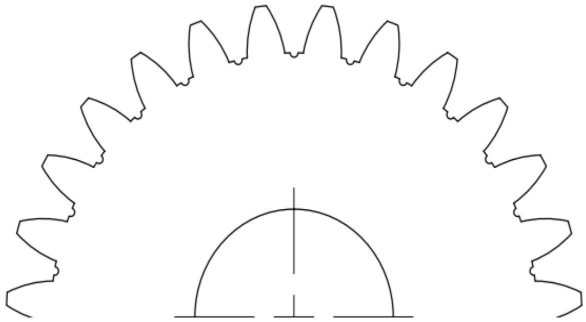


Figure 6.1: Location of the Groove at the Root

The groove created at the root side, 1 the diameter (dia.) of the hole is 1 mm is shown in the fig 6.1.

The result of the static analysis of case one is shown in fig 6.2. The Bending stress generated due to the tangential force is 95 Mpa. Which is less than the initial (without any hole as 198 Mpa) one.

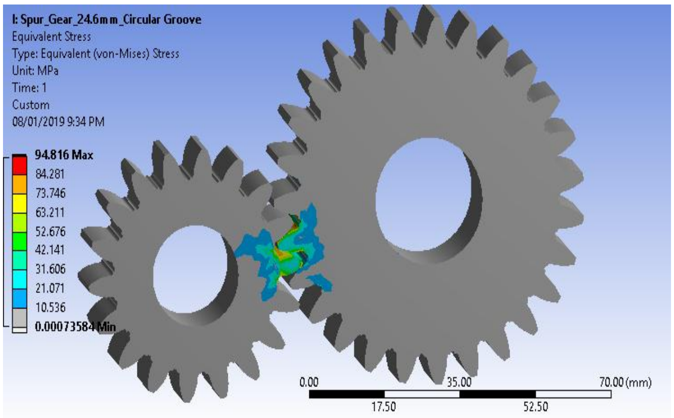


Figure 6.2: Bending Stress of Case 1

It has been clearly observed that there is a fair amount of stress reduction with the help of grooves created. We can say that the reduction in the stress with respect to the initial (without groove) and with the groove of case 1 the stress has been reduced up to great extent.

6.1.2 CASE 2: PATTERN OF CIRCULAR HOLES

If we are creating a single hole it doesn't make any sense because there are 20 other teeth that are going to be in contact with other gears' teeth. So checking a single hole is not a good concept, for this reason, we are going to create the pattern of the hole for the spur gear.

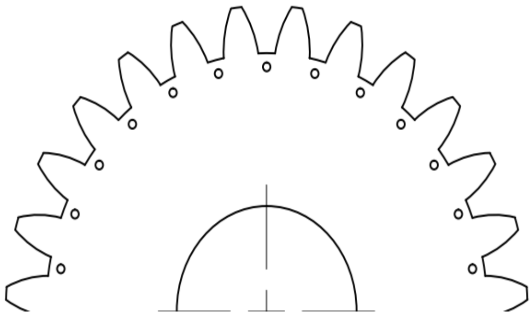


Figure 6.3: Location of the Holes at the Root

For this 18 and 27 holes are placed on Gear and Pinion in equi distance at 360º on the spur gear as shown in figure 6.3.

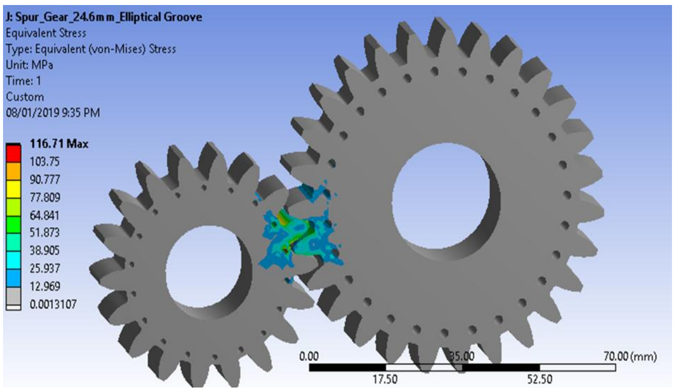


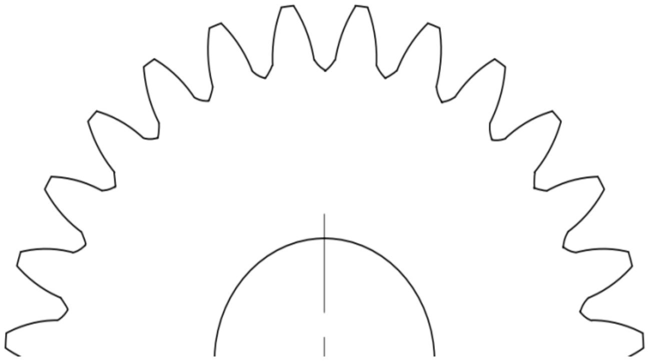
Figure 6.4: Bending Stress of Case 2

After creating 18 and 27 holes on the gear and pinion, let us check the variation in the stress, which is shown in fig 6.4. The bending stress is 117 Mpa for case 2.

It is observed that there is a stress reduction with the help of holes created. We can say that the reduction in the stress with respect to the initial model (without hole) but as compare to case 1 the stresses are quite more.

6.1.3 CASE 3A: ELLIPTICAL GROOVE AT ROOTS

In this case, we are taking two types of Elliptical grooves (3A: elliptical groove with a full cut at root and 3B: elliptical notch groove at the root). For this case, we've chosen one random elliptical groove with some random size and location to observe the result of it.



.**Figure 6.5:** Location of the Elliptical Groove at the Root

We are analyzing whether it reduces stress and increases life or not. The elliptical groove with a full cut at root created at the root side is shown in fig 6.5.

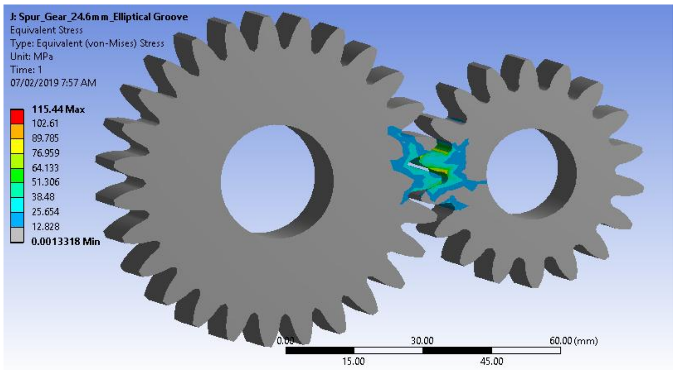


Figure 6.6: Bending Stress of Case 3A

The result of the bending stress of the case 3Aelliptical hole is shown fig 6.6. The Bending stress generated due to tangential force is 115.4Mpais less than the initial model (without any hole as 198 Mpa).

6.1.4 CASE 3B: ELLIPTICAL GROOVE NOTCH AT ROOT

In this case, we are taking, one elliptical groove notch at the root location. For this case, we've selected a random elliptical notch with a random size as shown in fig 6.7.

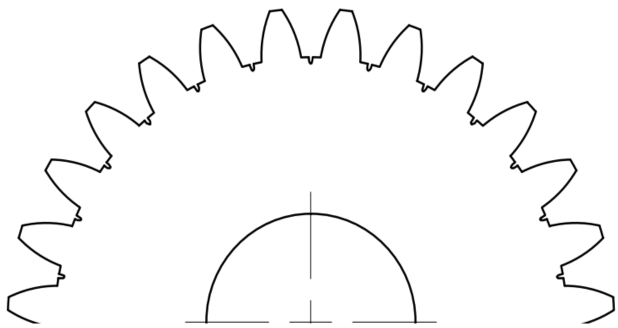


Figure 6.7: Location of the Elliptical Groove Notch at the Root

Similar to the previous case we are going to study this model for stress reduction and life increase. The input parameter as the maximum diameter (Dmax.) of the hole is 1mm.

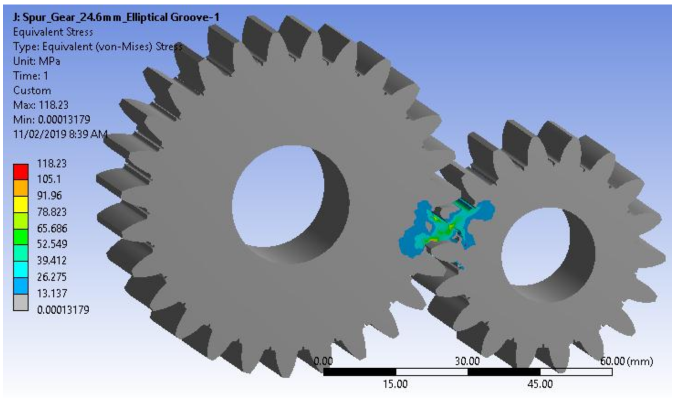


Figure 6.8: Bending Stress of Case 3B

The result of the case 3B is shown in fig 6.8. The Bending stress generated due to tangential force for elliptical groove notch is 118 Mpa. That is less than the initial (without any hole as 198 Mpa).

Table 2: Result of Bending stresses for Linear contact Analysis of different cases

Cases	FEA Analysis Results	Experimental Results	% Variation in Result
Linear Contact (Normal Profile)	198 Mpa	180 Mpa	9.01%
Case 1 – Circular Groove at Root	95 Mpa	91 Mpa	4.21%
Case 2 – Circular Hole Pattern near Root	117 Mpa		
Case 3a – Elliptical Groove at Root	115.4 Mpa		
Case 3b – Elliptical Groove with V Notch	118 Mpa		

From the above table 5.1, it has been shown results of bending stresses for different profile gear mesh model. The result is calculated by the FEA analysis method.

Firstly comparison of Bending stress results carried by FEA analysis and experimental method for the Normal profile gear mesh model is shown with the 9% variation in results that is within the acceptable range. For circular groove at root model the variation of result is 4.21%. The second comparison is done between the FEA results of all models. From these comparisons it is clearly shown that the Case1-Circular groove at the root gear mesh model is having the lowest bending stress.

6.2 NON LINEAR CONTACT ANALYSIS

Contact problems are more often nonlinear and require significant computer resources to solve. It is important that you know about the physics of the problem and takes the time to set up your model to run as efficiently as possible.

There are number of settings that are required to be carried out so that after analysis solution converges as expected. Some of the contact settings are described below.

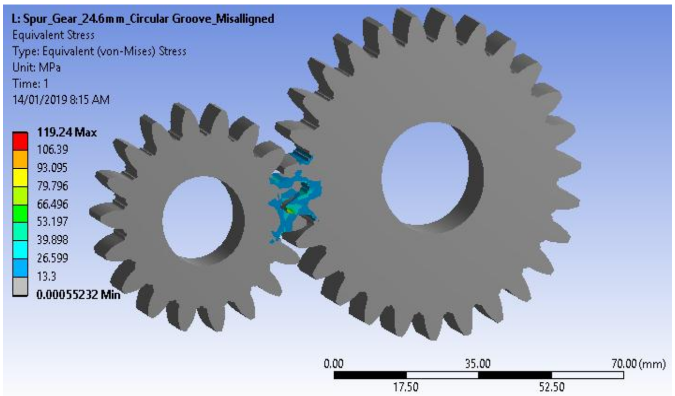


Figure 6.9: Bending stress at Misalign Condition

After giving inputs to preprocessor of ANSYS, result are evaluated to get nonlinear contact results. Stresses are Maximum 119 Mpa shown in Fig 6.9 at the curve end portion of gear, near the first contacting region, which is very less than the stress in initial condition of nonlinear miss aligned case that is 507 Mpa.

Table 5.2: Result of Bending stresses for Non Linear contact Analysis of different cases

Cases	FEA Analysis Results	Experimental Results	% Variation in Result
Non Linear Contact (Normal Profile)	507 Mpa	495 Mpa	2.36%
Case 1 – Circular Groove at Root	119 Mpa	114 Mpa	4.20%

The above table 5.2 shows the results of two models of gear mesh recorded by FEA analysis. The results of bending stress for Non-Linear contact analysis for case 1-circular groove at the root model is much lower than normal profile model so it is within the acceptable range.

6.3. DYNAMIC ANALYSIS

Using ANSYS workbench for the dynamic analysis of the gear system, modal analysis of the gear system must be done in the first place. The modal analysis belongs to the dynamic analysis and its main purpose is to find out the natural frequency of the gear system. According to the natural frequencies, actual working speed can be adjusted to avoid resonance.

Table 3: Natural Frequency by FFT and FEA

Mode No	Natural Frequency in Hz Normal Profile FEA Analysis	Natural Frequency in Hz Normal Profile FFT	% Variation in Result [Analysis Vs. Experiment]
1	1570.3	1615	2.76%
2	1520	1580	3.79%
3	1744.7	1798	2.96%
4	2132.4	2175	1.95%
5	2319.5	2369	2.08%

This desertion work contains the application of ANSYS software and FFT analyzer to determine the natural frequency and mode shapes of the gear drive. In order to prevent the resonance of the gear drive, it is necessary to find the natural frequency and mode shapes. Table 3. shows a comparison between reading recorded by FFT during experimentation of Normal profile gear mesh model and reading of FEA analysis of same Normal profile gear mesh model. The variation of both results is within the acceptable range which is shown in the above table.

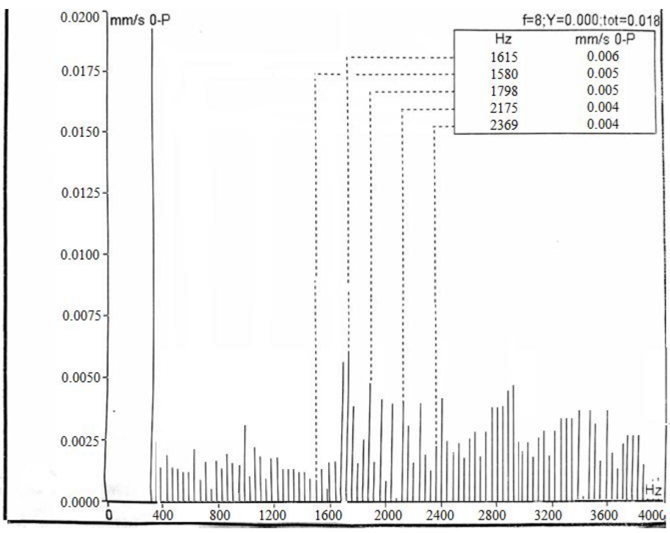


Figure 6.10: Plot of Natural frequency by Experiment

Above Fig 6.10 is a plot of Natural frequency taken during experiment of normal profile gear mesh model by using FFT analyzer. The plot generated for 5 modes as shown in plot.

Table 4: Natural Frequency of two models by FEA Analysis

Mode No	Natural Frequency in Hz Normal Profile	Natural Frequency in Hz Case 1 – Circular Groove at Root
1	1570.3	1598.6
2	1520	1610.3
3	1744.7	1626.9
4	2132.4	1941
5	2319.5	2135.4

The following solution is recommended, after the detailed dynamic analysis conducted on the gear mesh model. The gear mesh model was operating dangerously close to resonance, so the dynamic analysis was carried to shift the natural frequencies of the gear mesh.

The different modification was tried and the best feasible modification (Case1-Circular Groove at Root) is suggested to shift the natural frequencies of the gear drive. The cost of Case1-Circular Groove at Root is the lowest among all the alternative models.

7. CONCLUSIONS

In this dissertation work, we carried out the linear contact stress analysis as well as nonlinear contact stress analysis apart from that we complete dynamic analysis of gears system using finite element method (FEM) and validated results with Experimental results taken by using FFT analyzer.

From linear contact analysis, it is concluded that Case1-Circular groove at the root gear mesh model is having 95 Mpa that is the lowest bending stress among all remaining models. Similarly from Nonlinear contact analysis results, it concludes the bending stress 119 Mpa for case 1-circular groove at root model is much lower than the normal profile model.

From the dynamic analysis, we conclude the best feasible modification (Case1-Circular Groove at Root) is suggested to shift the natural frequencies of the gear drive. Also, the cost for Case1-Circular Groove at Root is the lowest among all the alternative models.

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