© 2016 IJSRSET | Volume 2 | Issue 4 | Print ISSN : 2395-1990 | Online ISSN : 2394-4099
Themed Section: Engineering and Technology

Modal Analysis of Asymmetric Helical Gear using Finite Element Modelling for Free and Prestress Condition

Rahul V. Borade*, S. V. Kshirsagar
Department of Mechanical Engineering, AISSMS COE, SPPU, Pune, Maharashtra, India

ABSTRACT

Modal analysis is used to determine the inherent dynamic characteristics of a system in forms of natural frequencies, damping factors and mode shapes. These parameters are important in the design of a system for dynamic loading conditions. Conventionally, modal analysis is performed with specific commercial tools. In this research paper the modal analysis of involute helical gear pair is carried out using finite element analysis tool ANSYS16.0. Gear noise and vibration is a major problem in many power transmission applications this problem becomes more significant in applications with higher operating speeds, where there is vibratory excitation which is related to the gear transmission error. The goal of modal analysis in this research paper is to determine the natural mode shapes and frequencies of the present helical gear pair and to comparatively analyse with frequencies of the geometrically modified asymmetric helical gear pair during free vibration as well as in pre-stress condition.

Keywords: Asymmetric Gear, Helical Gear, Modal Analysis, Mode shape.

I. INTRODUCTION

Now a day's most of the mechanical systems are subjected to dynamic loading which causes & shortens of the usable time, crack, noise and fatigue, in general the total effect of work for the mechanical system is lowered. Reasons for such behavior are type of loading, construction and conditions of work where the mechanical systems operate. The problems created the need of Vibration testing. Modal testing is the form of vibration testing of an object where the natural (modal) frequencies, modal masses, modal damping ratios and mode shapes of the object under test are determined.

A modal test consists of an acquisition phase and an analysis phase. The complete process is often referred to as a Modal Analysis or Experimental Modal Analysis. There are several ways to do modal testing but impact hammer testing and shaker (vibration tester) testing are commonplace. In both cases energy is supplied to the system with known frequency content. Where structural resonances occur there will be an amplification of the response, clearly seen in the response spectra. In the recent era, it is common to use the finite element method (FEM) to perform this analysis because, like other calculations using the FEM, the object being analyzed can have arbitrary shape and the results of the calculations are acceptable. The types of equations which arise from modal analysis are those seen in Eigen systems. The physical interpretation of the eigenvalues and eigenvectors which come from solving the system are that they represent the frequencies and corresponding mode shapes. Sometimes, the only desired modes are the lowest frequencies because they can be the most prominent modes at which the object will vibrate, dominating all the higher frequency modes.

Several studies are carried out on dynamic analysis of gear transmission. Recently Ankush Saxena et al [1] presented a paper, in this work modal analysis is carried out using solid elements in ANSYS Workbench of a geared rotor system supported on ball bearings at the ends. There work had been carried out to study the natural frequencies in different modes, to predict the direction of whirl of various modes as well as to study Campbell diagram. In the end, effect of variation in bearing stiffness has also been found out on natural frequencies in various modes of the geared rotor system. Kapelevich [2] proposed a method for the design of gears with asymmetric teeth. Several equations required for design of asymmetric gears were developed and presented for the synthesis of asymmetric gears.

Moreover, this study included the results of the experimental study conducted on a planet gearbox of an airplane engine. In 2010 Neils Pedersen [3] shows in his work that how bending stresses can be reduced significantly by using asymmetric gear teeth and by shape optimizing the gear through changes made to the tool geometry, he also suggested the use of two new standard cutting tools. In 2008 Fatih Karpat et al [4] studied the dynamic behavior of a spur gear in which he developed a computer program using MATLAB software and find the dynamic transmission error associated with the contact position.

II. METHODS AND MATERIAL

1. Modeling of Helical Gear

Fig. 1 shows the model of the helical gear with asymmetric teeth. In this analysis, the model of input gear and output gear is only considered in FEM analysis to save computing time. This ignores the interaction of the housing fitting and bearing fitting.

Figure 1: Model of asymmetric helical gear.

All the gear pair are modeled in Creo parametric 2.0 following the same gear design parameters and material properties as shown in table 1. Apart from using standard 20 deg. pressure angle here we are using seven different combination of pressure angles. These gears having different pressure angles are defined with the help of coefficient of asymmetry (K) which is a ratio of coast side base circle diameter to drive side base circle diameter.

Coefficient of asymmetry $(\mathrm{k}) = \frac{D_{bc}}{D_{bd}} = \frac{\cos\varphi_c}{\cos\varphi_d}$

TABLE I
GEAR PAIR PARAMETERS AND MATERIAL PROPERTIES

Parameters	Gear pairs			
Material	Steel AISI 1045			
Modulus of Elasticity (E)	210 GPa			
Poisons Ratio (μ)	0.28			
Module (m)	4.5 mm			
Number of teeth (N)	20			
Gear ratio (G)	1			
Top Land thickness coefficient (m_o)	0.4 / N			
Helix angle (β)	25			
Face width (b)	20 mm			
Fillet radius (Rf)	0.4 * m			
Clearance (c)	0.25 * m			
Drive side pressure angle (φd)	20	25	30	35
Coast side pressure angle (φc)	20	25	30	35
Addendum (ha)	1 * m			
Dedendum (hd)	1.25 * m			

2. Modeling of Three Different Type Of Gear Pairs

There are three different gear pair designs considered in the interest of this research, these three type are based upon value of coefficient of asymmetry (K) ,for standard gear it is equal to one, for gear having driven side pressure angle greater than 20 deg it is greater than one and conversely for gear having coast side pressure angle greater than 20 deg it is less than one. These three pairs are assembled as shown in table 2 so that FEM simulation can be carried out separately.

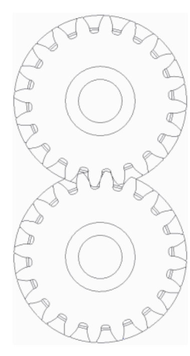
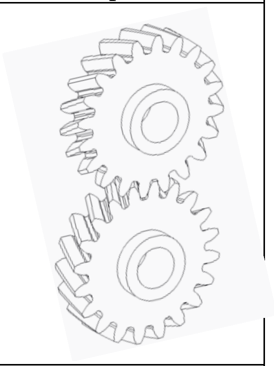
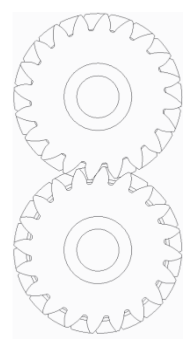
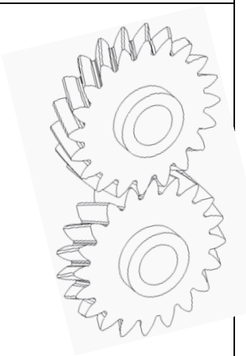
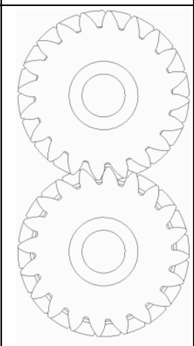
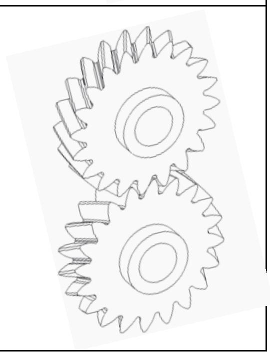
3. Modal Analysis

Modal analysis, which means the study of the structure mode shape under excitation to its natural frequency, is important in design stage. the modal analysis of the gear pair with different combination (k =0, k < 1, k > 1) was analysed under free stress condition and pre- stress condition.

A. Mode Shapes

The mode shapes of the gear in FEM were calculated independently of the excitation, which means that the structure is only mass and stiffness distribution dependent. According to Berlioz and Trompette, low resonance modes (first few modes) correspond to higher global modal response (higher amplitude of excitation) compared to higher resonance modes. The authors also verified that higher modes are more sensitive to structural modifications. Local modifications of the geometry generally have impact on the strain energy distribution. Hence, structural changes may be used as efficient tool if they correspond to a significant modification of the masses and stiffness, as the resonance frequencies depend on ratio of K/M.

TABLE III
ASSEMBLY MODELS OF THREE TYPES OF GEAR PAIRS.

Gear pair	2-d view	Isotropic view
Symmetric Helical Gear Pair K = 1		
Asymmetric Helical Gear Pair K = 1.1471		
Asymmetric Helical Gear Pair K = 0.8717		

B. Mesh Setting

In ANSYS workbench, the three gear train designs are imported. In modal analysis using FEM, each gear pair model was meshed with purely tetrahedron elements with average element size of 2mm. fig. 2 shows the mesh model of the gear pair having gear ratio equal to one. The model consists of 268430 nodes and 92453 elements.

C. Boundary Conditions

ANSYS workbench program was used to apply the motion constraints and contact conditions. There are two different conditions being analysed in conducting modal analysis of the gear pairs. The first six natural frequencies and their mode shapes were analysed on the gear pair without pre-stress state involving constraints only and pre-stress state which is inclusive of loads and constraints. Fig. 4 shows the constraints applied for the gear pair under free stress state.

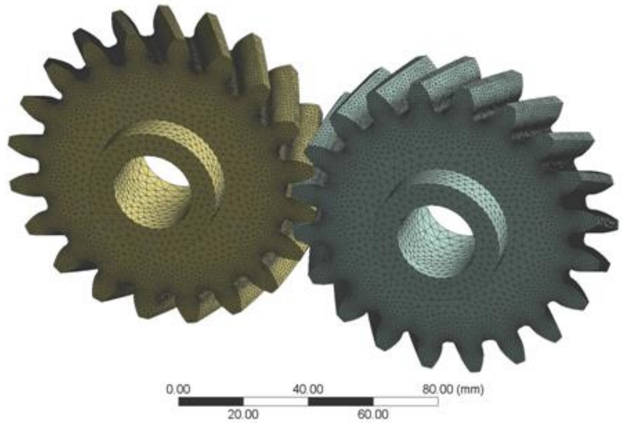


Figure 2: Meshing model of helical gear with K=1

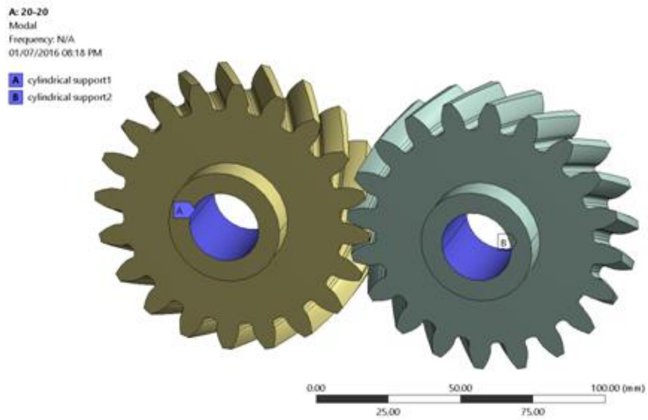


Figure 3: Free stress state of gear pair having K=1

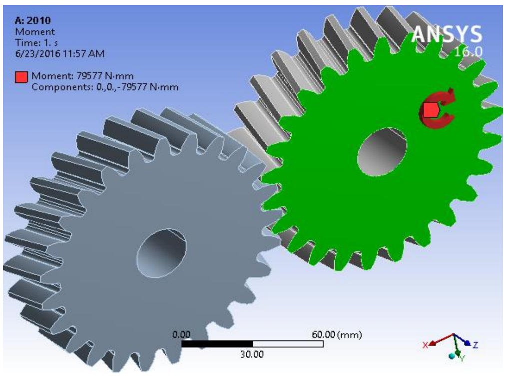


Figure 4: pre-stress state of gear pair with K=1

In the free stress state, cylindrical supports were assumed as the bearing supports for the gear components, which allow rotational motion along the shaft axis but restricts axial motion and radial motion. Fig. 4 shows the gear pair under pre-stress state. In the pre-stressed state of the gear pair, the cylindrical support is set the same as the one in the free stress state. A torque of 79.577 Nm was applied on input gear. The cylindrical support of the output gear was fully constrained to create static loading condition. In pre-stress state, the stress due to moment and gravity was calculated first before modal analysis was applied. To set the single pair tooth contact in FEM, the contact surface and target surface were defined and contact type was set to “No Separation” contact.

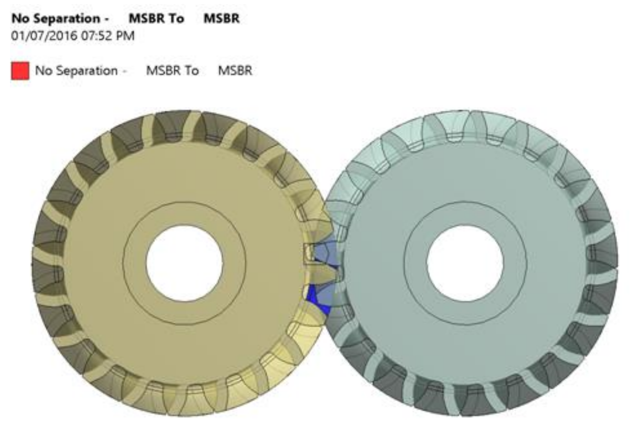


Figure 5: Detailed view of gear tooth contact between two gears.

4. Modal Analysis of Gear Pair In Freestress And Pre-Stress State

Fig 6 shows the first two mode shapes of the gear pair under free stress state and subjected to single pair tooth contact.

TABLE IIIII
EFFECT OF COEFFICIENT OF ASYMMETRY ON FIRST SIX MODE SHAPE

	Coefficient of asymmetry (K)						
ω	1	1.036	1.085	1.1471	0.964	0.921	0.8717
1st	1824	1835	1840	1843.1	1846	1840.9	1869
2nd	4301	4435.2	4256	4465.1	4386	4280.1	4484
3rd	6642	6634.5	6636	6648.1	6665	6632.6	6656
4th	7667	7614.9	7604	7652.8	7667	7613.3	7705
5th	11443	11417	11408	11512	11419	11398	11499
6th	11863	11811	11846	11828	11808	11842	11848

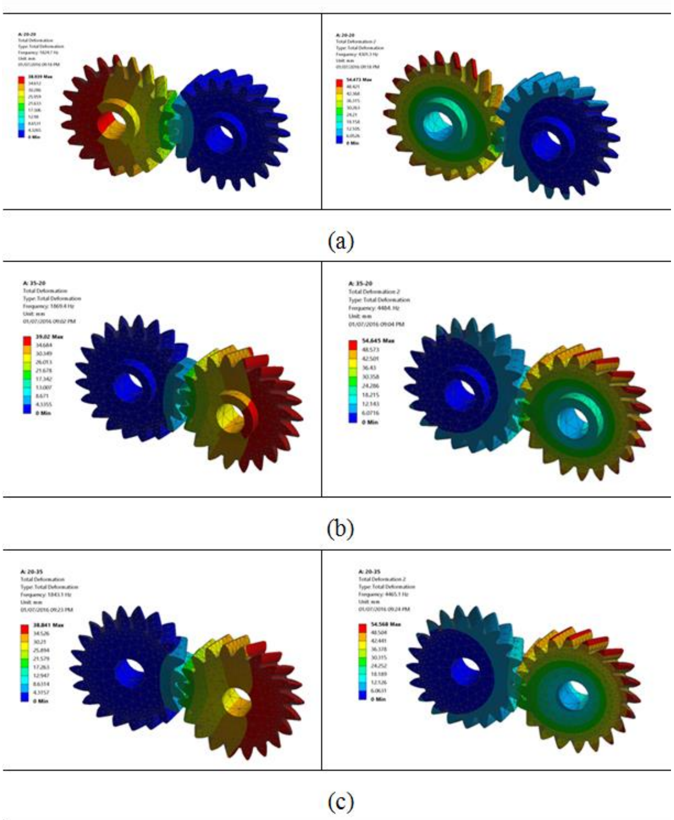
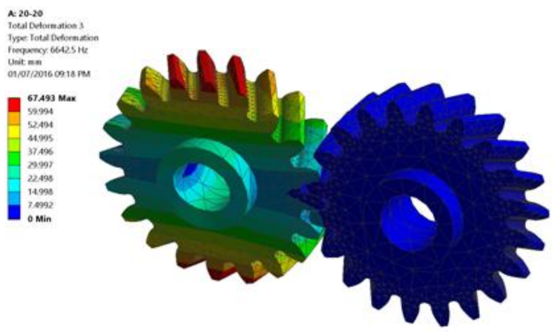
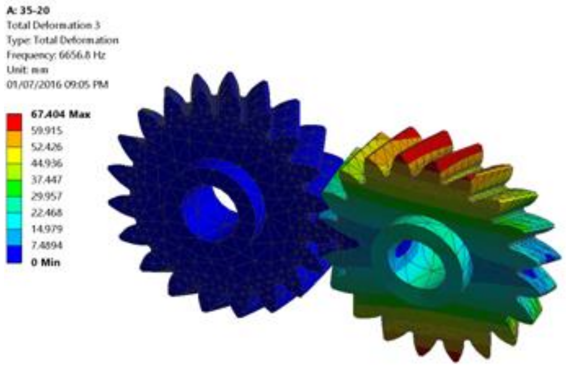


Figure 6: First two mode shape of gear pairs having different coefficient of asymmetry in free stress state (a) k = 1 (b) k = 1.1471 (c) k = 0.8717

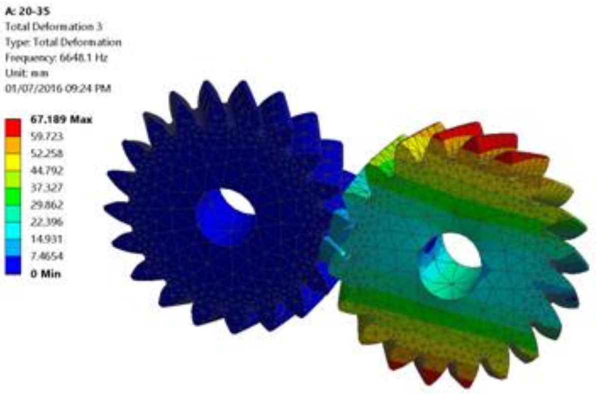
The first and second mode shape of the gear pair corresponded to radial expansion and twisting around the gear teeth. We know gear is a continuous mass system having infinite number of natural frequencies. But we takes only first few natural frequencies in to consideration. Here we use first six natural mode shape to understand the effect of asymmetry over symmetric profile. Results for all gear pairs in free stress state are shown in table 3.



(a)



(b)



(c)

Figure 7: The first natural frequency for gear pairs having different coefficient of asymmetry in pre-stress state
(a) k = 1 (b) k = 1.1471 (c) k = 0.8717

III. RESULTS AND DISCUSSION

The result in table 3 shows the variation of mode shapes with coefficient of asymmetry. In this study we used six mode shapes to compare the effect of asymmetry on natural frequency. The first two mode result for all gear pairs are plotted in graphical formate and are shown in fig 8 and 9. As we know resonance needs to be avoided when system is running at high frequency and to do the same we consistently work to find the methods to avoid this situation. To get the above stated goal here we find the effect of symmetry over first two resonance frequencies. From fig 8 it is seen that first natural frequency for standard gear is 1824.9 Hz in free stress state and it increases with increased in coefficient of asymmetry ,for gear pair with k=1.1471 it is found to be 1843.1 Hz. While in case of gears having k<1 it first increases then decreased by some amount and again increases.

The result observed for third type of gears having k<1 is also found in 2nd mode shape result but here these variation is common both for gears having k>1 and gears having k<1, fig. 9 shows this variation. These result shows that gears with asymmetric gear can be used as a tool to avoid interference in high speed transmission.

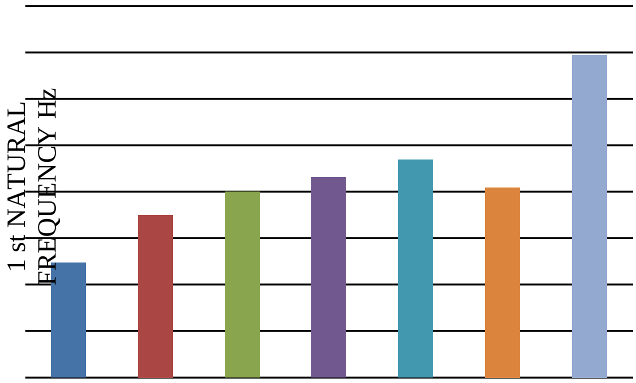


Figure 8: Variation of 1st natural frequency with coefficient of asymmetry (K)

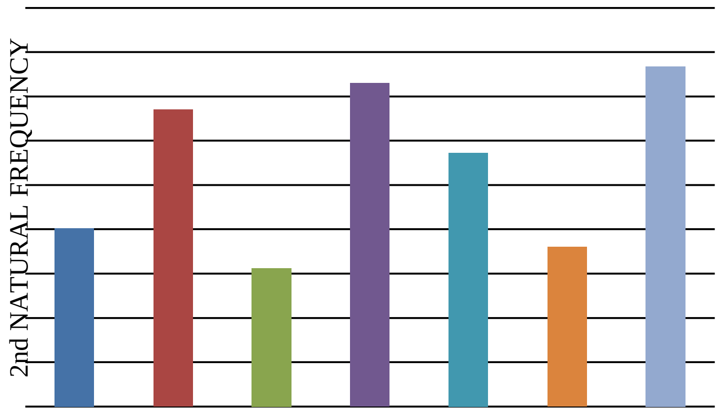


Figure 9 : Variation of 2nd natural frequency with coefficient of asymmetry (K)

IV. CONCLUSION

A finite-element model is developed to study the dynamic behavior of helical gear pair using FEA software. The results of natural frequencies and their corresponding mode shapes are reported. It has been found that first two modes are radial expansion and twisting modes due to transverse vibrations whereas fifth mode is torsion mode which is due to torsional vibration. Effect of coefficient of asymmetry on natural frequencies of the system is also studied using graphs and figure which shows that for first natural frequency as coefficient of asymmetry increases natural frequency also increases while for decreased in coefficient of asymmetry it cannot gives linear variation . Finally, it has been shown that the asymmetric gear profile can be used to avoid resonance condition for system running at high speed application. This type of study may be helpful in understanding modal behavior of geared rotor systems. In future, modal analysis using experimental approach will be tried for similar type of gear system.

V. REFERENCES

[1] Ankur Saxena, Anand Pareya and Manoj Chouksey, "Study of Modal Characteristics of a geared rotor system", Procedia Technology 23 (2016) 225 – 231

[2] Alexander Kapelevich, "Geometry and design of involute spur gears with asymmetric teeth", Mechanism and Machine Theory, vol. 35, pp. 117-130, 2000.

[3] N. L. Pedersen, "Improving bending stress in spur gears using asymmetric gears and shape optimization", Mechanism and Machine Theory, vol. 45, pp. 1707–1720, 2010.

[4] Fatih Karpat, Stephen Ekwaro-Osire, Kadir Cavdar and Fatih C. Babalik, "Dynamic analysis of involute spur gears with asymmetric teeth", International Journal of Mechanical Sciences, vol. 50, pp. 1598–1610, oct 2008.

[5] S. Shanmugasundaram, M. Kumaresan and N. Muthusamy, "Effects of pressure angle and tip relief on the life of speed increasing gearbox: a case study", Springer Plus, vol. 3, pp. 746, 2014.

[6] Santosh S. Patil, Saravanan Karuppanan, Ivana Atanasovska and Azmi Abdul Wahab, "Contact stress analysis of helical gear pairs, including frictional coefficients", International Journal of Mechanical Sciences, vol. 85, pp. 205–211, may 2014.

[7] P. Marimuthu and G. Muthuveerappan, " Design of asymmetric normal contact ratio spur gear drive through direct design to enhance the load carrying capacity", Mechanism and Machine Theory, vol. 96, pp. 22–34, 2016.

[8] YumeiHua, Yimin Shao, Zaigang Chen and Ming J. Zuo, "Transient meshing performance of gears with different modification coefficients and helical angles using explicit dynamic FEA", Mechanical Systems and Signal Processing, vol. 25, pp. 1786–1802, 2011.

[9] Kahraman, A., H. N. Ozguven, D. R. Houser and J. J. Zakrajsek (1992). Dynamic analysis of geared rotors by finite elements. Journal of Mechanical Design. 114(3): 507-514.