A Review of Finite Element Analysis of Gears in Epi-Cyclic Geartrain

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*Abstract***—** Gearing is one of the most critical components in mechanical power transmission system. Most of the gearing system fails due to the failure of gear tooth. Hence it is important to understand gear tooth failures and carryout analysis for their strength sufficiency. Based on the study of the research papers, it can be concluded that the gear stress analysis and prediction of gear dynamic loads for gear sets are major concerns. But reported research on FE analysis of gears is based on static condition and has limited scopes.

I. INTRODUCTION

Planetary gearbox is used in machineries and machine tools to obtain speed reduction, which in turn increases the torque. These gearboxes are used in many applications such as automatic automobile transmissions and hybrid transmission systems [1]. Planetary gear trains have attracted the attention of designers for the possibility of efficient and quiet transformationof rotation and the transmission of large torques with gears of relatively small dimensions [2]. The power transmission efficiency and compactness is higher. Efficiency loss is usually of order 3% per stage resulting into large proportion of the energy transmitted by the gearbox. As the load being transferred is shared between number of planets, torque capability is greatly increased. As the number of planets in the system increases, there will be higher the torque density and the greater load ability. There is a greater stability due to the even distribution of masses. Some of the disadvantages of the planetary gearbox are: High bearing loads, Constant lubrication requirement, Inaccessibility, and design complexity, the backlash of planetary gears cannot be adjusted.

A pair of teeth 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. In addition there may be surface damage associated seizure of surfaces due to poor lubrication and overloading. Hence it is important to study the potential failure analysis.

II. BASIC COMPONENTS & TYPES:

The epicyclic gear train can be divided into four components viz. Sun, Planet, Annulus and Planet carrier.Figure 2.1 indicates these components.

1. Sun: It is the central gear and rotated by motor.
2. Planet: It is gear in mesh with sun as well as outer ring. The planet rotates about sun.
3. Planet carrier: It supports one or more peripheral planets, which are of the same size and are meshed with the sun.
4. Annulus: It is an outer ring having inward teeth which is in mesh with the planet gears.

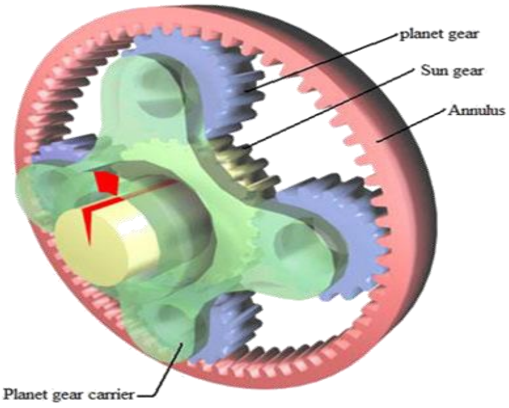

Fig. 1: Basic components of epicyclic gear train

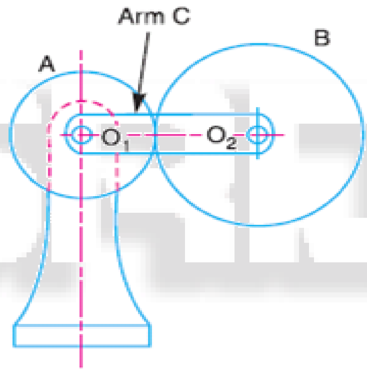

Fig. 2: Epicyclic gear train

Figure 2.2 shows working of a simple epicyclic gear train where arm C and a gear A share a common axis at O_1 around which both rotates. The gear B is in mesh with gear A and it rotates about axis O_2. If the arm is fixed, the gear train is termed as simple and gear A can drive gear B or vice- versa. When gear A is fixed and arm rotates about the axis O_1, then gear B will rotate upon and around gear A. Such a motion is called epicyclic(epi. means upon and cyclic means around) andgear train arranged in such a manner that one or more of their members move upon and around another member is known as epicyclic gear train.

The configuration with fixed sun is termed as *solar* while the one with fixed carried is termed as *star*. The configuration with fixed annulus is called as *planetary*.

III. LITERATURE REVIEW

A. Raptiset al. [5] considered the calculation of maximum stress at gear tooth root when the mating gears are loaded at their most unfavourable contact point (highest point of single-tooth contact). The study used numerical and experimental methods. Finite Element Method (FEM) was used for the numerical stress analysis and photoelasticity is applied for the experimental investigation of the stress field. The experimental results of maximum dimensionless stress derived from photoelasticity experiments are compared to theoretical stress results. . It was found that the deviation between results of applied methods lie between reasonable limits whereas it rises with increasing number of teeth for large gears.

B. Gupta et al. [6] conducted contact stress analysis of spur gear set. The study addresses determination of analytical contact stresses as well as Finite element analysis using ANSYS 13.0. It was shown that the contact stress analysis of spur gear assembly can be successfully conducted using FEA and good agreement in results with analytical method has been observed. The study also developed a methodology to improve hardness of gear tooth profile to improve resistance against pitting failure.

C. Hamandet al. [7] estimated wear stresses and bending stresses in sun gear. The study also estimated various forces acting on gear tooth. The study estimated bending stresses and shear stresses along with deflection using analytical method as well as using FEA software. It was reported that the FEA for various stress and deflections closer to their analytical values. An appreciable reduction in bending and shear stress values was obtained using trochoidal root fillet design in place of circular root fillet design. The study reported higher wear and lesser deflection for trochoidal root fillet design in comparison to circular root fillet design. It is also generalized that the circular root fillet design is optimum for lesser number of teeth in pinion while trochoidal root fillet design is more ideal for higher number of teeth in gear (more than 17 teeth).

D. Yuksel et al. [10] developed computational model of a planetary gear set to study the influence of surface wear on dynamic behaviour of a typical planetary gear set. The overall computational scheme attempted to combine a wear model defining geometric description of contacting gear tooth surfaces having worn and a deformable body, dynamic model of a planetary gear set. The wear model employed a quasi-static gear contact model to compute contact pressures and Archard's wear model to determine the wear depth distributions. The worn surfaces were input into the dynamic model to quantify the impact of gear tooth wear and mesh dynamic forces. The results on a planetary gear set having a fixed planet carrier indicated that the dynamic behaviour is nonlinear due to tooth separations in its resonance regions. The results for worn gear surfaces indicate that the surface wear has great influence in off-resonance speed ranges while it diminishes near resonance peaks due to tooth separations.

E. Chen et al. [11] performed dynamic simulation of spur gear with tooth root crack propagating along tooth width and crack depth. In this study, an analytical model was proposed to investigate the effect of gear tooth crack on the gear mesh stiffness. The model incorporates tooth crack propagation along tooth width and crack depth during early stages. The mesh stiffness of a spur gear pair with different crack length and depth were obtained using analytical formulation. The study also simulated effect of gear tooth root crack size on the gear dynamics along with corresponding changes in statistical indicators.. The results show that RMS (full name) and kurtosis increased with the growth of tooth crack size along tooth width and crack length. Frequency spectrum analysis was also carried out to examine the effects of tooth crack. The results showed that sidebands caused by the tooth crack were more sensitive than the mesh frequency and its harmonics. The developed analytical model predicted the change of gear mesh stiffness with presence of a gear tooth crack. It was concluded that the presented study provides good insight for gear condition monitoring and fault diagnosis, especially for the gear tooth crack at early stage.

F. Huang et al. [15] presented approaches for element construction and dynamic analyses of involute spur and helical gears, including cylindrical and conical categories. Various design parameters such as pressure angle, correction factor, tip relief, crowning modification, and undercutting were incorporated in the models. The equations of involutes, fillets and other curves for teeth were derived after applying coordinate transformation and gear principles to transverse cross section profiles of a rack cutter. The coordinates of intersection point among nonlinear tooth profile were computed using the Newton–Raphson method. This process eliminates need for CAD model of gears and meshed elements were constructed directly using profile equations of gears via a C program

G. Li et al. [16] presented a dynamic method for numerical analysis of gear strength which is quite different from static analysis method. An FE model of gear pair was developed to estimate gear strength including tooth surface contact stress and tooth root stress with change of load factor. The results showed that the tooth root stress and tooth root contact stress increases linearly with the increase of load factor. The FEM and traditional ISO results were analysed and it was concluded that the proposed method was more effective and efficient.

IV. CONCLUSION

There has been a great deal of research on gear analysis, and a large body of literature on gear modeling has been published. The gear stress analysis, the transmission errors, and the prediction of gear dynamic loads, gear noise, and the optimal design for gear sets are always major concerns in gear design.

In spite of the number of investigations devoted to gear research and analysis there still remains to be developed, a general numerical approach capable of predicting the effects of variations in gear geometry, Hertz contact stresses, bending stresses and Von Mises stresses.

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