International Journal of Research
Available at https://edupediapublications.org/journals
p-ISSN: 2348-6848
e-ISSN: 2348-795X
Volume 03 Issue 12
August 2016

Bending Fatigue Performanceof Symmetric and Asymmetric Composite Gear

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ABSTRACT: -*Gear is the principle power transmission element finding diverse industrial applications. Polymer gears have found major application in industries where load involved is less and the load transmission is in one direction only. In this work symmetric gear (20/20) asymmetric gears (20/34, 34/20) were chosen to understand the kinematic and load carrying performance. Unfilled and 20 % glass filled polypropylene materials were considered for molding symmetric and asymmetric gears*

In this Paper, work kinematic performance was evaluated by commercial FEA package ANSYS. Gear tooth geometry was modelled. Symmetric and asymmetric gears were injection molded for bending fatigue performance evaluation. Since polymer gears are more prone to bending fatigue failure rather than that of contact fatigue, effort has been put to develop appropriate test facility. The bending fatigue performance of asymmetric gears, a fixture has been designed and fabricated in workshop. Addition of reinforcement found to improve load carrying capacity. Molded gears were subjected to sinusoidal variation of displacement and load required at every stage of cycles was measured and monitored. For both the considered materials, chosen loading frequency and chosen displacement magnitudes, the amount of load required for gear tooth deflection found to decreases. When the loading frequency increases molded gears exhibit improved load carrying performance. Drive side pressure angle and critical gear tooth section found to influence the bending load carrying performance of symmetric and asymmetric gears.

KEYWORDS: -Gear, Transmission error, Friction, Temperature, Bending fatigue.

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1. INTRODUCTION: -Gear is the principle power transmission device. Steel gears are generally used for applications which involve larger load to be transmitted. Now day's polymer gears have found major application in industries where load involved is less. These include printing, automatic Taylor machines, refrigerators etc. Generally in those industries the load is transmitted in one direction only. Traditionally symmetric gears have been in use as they can transmit load with equal efficiency in both the directions.

The teeth of gears are generally composed of two involute profiles. The involute profile is obtained by module and pressure angle. Pressure angle at the pitch point is the angle which the line of action makes with the tangent at the pitch point. Symmetric gears have same pressure angle at both the involute profiles. In the present case, symmetric gear has a pressure angle of 20^0 on both the involute profiles. Asymmetric gears have different pressure angle on both the profiles. The asymmetric gear chosen for the analysis in this work has a pressure angle of 20^0 and 34^0 on the two profilesthe work presented here is aimed at the dynamic analysis of asymmetric gear. In a standard symmetric gear, pressure angle being the same for both the profiles of the tooth, the bending and contact strength would be same at both the sides of a gear tooth. However, in most cases, gears are being used for transmission of power/motion in only one direction. Hence one side of gear tooth (drive side) is significantly loaded for longer periods and other side (coast side) is not loaded/loaded only for shorter period. In spite of this functional difference, standard gears are having symmetric gear tooth profile due to the difficulty in manufacturing.

1.1. Polymer Gear

Polymer gears have several advantages such as light weight, reduced noise, and high degree of freedom in gear geometry, in comparison with metal gears. The main advantage of using polymer gears is their relatively low cost of manufacture when produced on a large scale using a net-shape process such as injection molding. In addition, polymer gears offer the possibility of running without external lubrication (through the use of dispersed additives such as PTFE or MoS_2) and offer a lower noise alternative to metallic gears. The main disadvantage of using polymers is their lower load-carrying capacity as compared to metal gears. In an attempt to use polymeric gears in higher load applications, the size of the gear is sometimes increased.

Among polymer gears made by injection moulding, primarily polyacetal (POM), and also polyamide (PA) and polycarbonate have been used for various parts of copy machines, facsimiles, and printers. Their application fields, however, are rather limited like these, since their strengths are weaker than metals and often their teeth are suddenly worn away with increasing running-time and torque. Investigations on polymer gear performance were extensively conducted.

1.2. Asymmetric Polymer Gears

In spite of extensive work on asymmetric gear design, utilization of asymmetric gears is limited due to its manufacturing difficulties. Manufacturing the asymmetric gear through conventional process such as hobbling, shaping, milling would become time consuming and hence not economical. On the other hand, when these gears are manufactured through mass production techniques such as powder metallurgy and injection molding technique (meant for metal matrix composites and thermoplastic composites), it is found to be economical. Polymeric gears can be manufactured through injection molding process, where there is a need of a single die for multiple gears. These die cavities with complex shape can be manufactured by wire cut electric discharge machining technique, where the tooling cost per single gear becomes insignificant. Injection molded polymeric gears find many applications in motion transmission than power transmission due to its limited load carrying capacity. In this direction, few experimental investigations on polymer gear efficiency were reported in the recent years.

1.3. Bending Fatigue

In certain space applications of gears, the level of loading applied to the gear mesh members can be large enough to cause a great deal of the gear's life to be used in a short number of cycles. The use of this standard at bending stress levels above those permissible for short number of cycles requires careful analysis. To estimate the fatigue live of gear teeth, the standard's analysis techniques calculate a life at a given stress for 99 present reliability at the component level. This means that a large population of components designed using the allowable stress values should experience crack initiation at a rate not greater than 1 per 100 for the given cycle count. To provide a credible estimate of fatigue life at 99 percent reliability, extensive experimental data is required to establish the load-life relationship in the low-cycle fatigue (<10,000) regime.

2. LITERATURE SURVEY

2.1. Kinematic Analysis

Jiande Wang [1] investigated spur gears in mesh numerically as well as experimentally. By restraining the gear body at the hub and finding out the deflection at the tip of the tooth transmission error was estimated. The aim was to propose a design for a particular load at which the transmission error for symmetric gear would be constant. The analysis was carried out in ANSYS software.

Houser et al. [2] proposed a procedure for computing static transmission error for low and high ratio contact spur gears. The range of contact of two symmetric spur gears was divided into small angles and at each of these angles the transmission error was computed. The total compliance between a pair of teeth was assumed to be contributed by: cantilever beam deflection due to bending and shear, rigid body tooth rotation and Hertzian deflection.

Karimpouret al. [3] described an investigation into the contact behaviour of polymer gear transmission error using numerical finite element method and analytical techniques. A polymer gear was modelled and the analytical results were calculated using ABACUS software. The simulations showed that the kinematic behaviour of the polymer gear is substantially different from that predicted by the classical metal theory. The extreme tooth bending and the difference between the analytical and the numerical stresses observed in the simulations suggest that any future work must account for the load sharing ratio and friction particularly in dry running conditions.

2.2. Bending Fatigue

Akataet al. [5] proposed the three point bending loading. The three point bending test fixture consists of upper and lower parts of the test fixture were machined from a H13hot work tool steel, then hardened and tempered to a hardness level of 54-RC Needle element shaving 2mm diameter were attached to V-shaped grooves that were machined on the upper part of the fixture. The gear to be tested was placed on a lower test fixture and bending loads were applied via its upper part by means of needle element shitted on the fixture. The distance between the loading centres of the needle elements was supposed to be the diameter of HPSTC.

Daniewiczet al. [6] proposed that fatigue life of machine elements may be increased if beneficial compressive residuals tresses are introduced. One mechanical method of introducing compressive residual stresses is pressurising or presenting. Ming-Haung Tsai *et* *al*. [10] Young's modulus of the polymer material is lower than that of steel by two orders, the effects of large tooth deflection on the static transmission errors of plastic gears become significant. A multi-tooth contact analysis using finite element method for calculating the static transmission errors of plastic spur gears is established to compare with the existing method for steel spur gears. According to the comparison results, a modification of the existing method is proposed for plastic spur gears and verified by the same finite element contact analysis.

2.3. Friction between Mating Gears

Vijayaranganet al. [7] attempted to study the contact stress of a pair of mating scar teeth, under static conditions, by using a two-dimemional finite element method and the Lagrangian multiplier technique. The contact condition at the neighbouring node pair, namely, the contact existing over a number of node pairs in the contact zone, has been considered. To study the effect of friction between the mating gear teeth a range of average static friction coefficients, from 0.0 to 0.3, has been considered. The actual length of contact against the calculated length of contact is discussed. The variation of contact stress along the contact surface in a direction normal to the mating surfaces, which will give an idea about the depth of hardening required, is also presented.

2.4. Effect of Temperature on Polymer Gears

Santhosh Kumar [8] evaluated the effect of temperature on symmetric and asymmetric polymer on gear tooth profile. Gears were subjected to various temperatures and change in tooth profile was examined. As the temperature increases, the amount of expansion in the profile area increases. The displacement is clearly visible at highest temperature where as for other temperatures it is almost seems to be coinciding due to very less displacement.

3. GEAR DESIGN

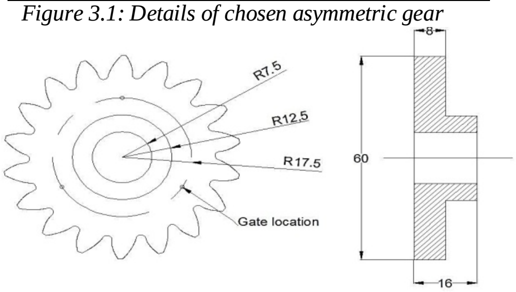
In the present investigation, commonly used 3 module spur gear is chosen. In order to avoid interference for the standard 20° pressure angle, 18 numbers of teeth is chosen. It was decided to use maximum possible pressure angle at other side to get maximum advance of asymmetric gear. Major limitations of increasing the pressure angle is the reduction of gear tooth thickness at addendum circle. Tooth shape becomes more and more pointed as the pressure angle increases. Consequently the top land becomes correspondingly smaller and ultimately results in pointed tip.

This phenomenon is termed as “peaking”. The peaking limit sets a boundary to the maximum magnitude of pressure angle. Gear standard procedures such as IS, recommended that the tip thickness should be greater than equal to 0.2 times the module for the hardened gears. To avoid peaking, maximum possible pressure angle 34° was decided for the chosen module and number of teeth. Addendum of 1 mm module was selected similar to the standard full depth tooth system. Since fillet radius was most important design parameter in the polymer where bending failure is most common failure. As per the design recommendations 0.38 times the module was selected (1.14 mm) for the fillet radius for the chosen asymmetric gear. Table shows the all the gear dimensions and figure shows the considered symmetric and asymmetric gear.

Parameter	**Value**	
Pressure angle (°)	20	34
Module (mm)	3	
Number of teeth	18	
Pitch circle diameter (mm)	54	
Tip circle diameter (mm)	60	
Root circle diameter	47.04	

(mm)		
Base circle diameter (mm)	50.74	44.77
Face width (mm)	8	
Root fillet radius (mm)	1.14	

Figure 3.1: Details of chosen asymmetric gear



injection molding machine (Texair 40 Ton) is used for molding symmetric and asymmetric gears. Unfilled and 20% glass fibre reinforced polypropylene materials are used for molding gears. Gears are looks like below.

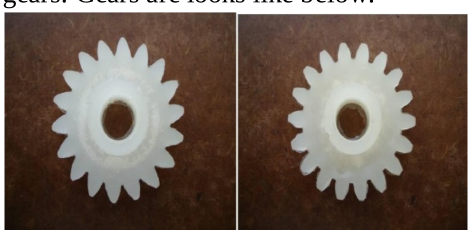


Fig 3.1a: Asymmetric gear with unfilled PP
Fig 3.1.b: *Symmetric gear with unfilled PP*

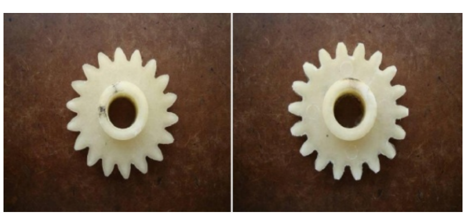


Fig 3.2.a: *Asymmetric gear withunfilled pp with 20 % of glass fibre.*
Fig 3.2b: Symmetric gear withunfilled pp with 20 % of glass fibre

4 EXPERIMENTAL EVALUATIONS

4.1. Bending Fatigue Fixture

The model was fabricated it was put up for bending fatigue test.. For fixing the bending fatigue on instron machine grippers are removed and instead of that an extension rod was fitted. Both hydraulic ram and extension rod consist of internal thread (M30). By means of two studs bending fatigue fixture was fixed to instron. Ram was raised to a 69.795 mm for proper alignment. The composite gear was restricted to all the direction of motion by washers, nut and small pipe. Composite gear was pressed against the hollow pipe by nut . Vertical shaft was shaft was fitted to connecting rod by the means of pin joint.

4.1.1. Load variation of Unfilled Polypropylene Symmetric Gear (20/20)

Polymer gear was subjected to sinusoidal fatigue load under constant displacement mode. In the beginning stage of testing the load caring capacity of polymer gear was high and as the cycles progresses the load caring capacity was gradually decrease. Fig 4.1.1 -4.1.4 exhibits the variation of load at various stage of service of unfilled PP symmetric gear. Load is plotted on the ordinate and number of cycle is plotted on abscissas. The rate of decrease is found to depend upon frequency and deflection amplitude. Figure 4.11 shows the load variation of unfilled polypropylene gear at 1Hz and 3 mm deflection. Initially the load was 1.7 KN, as the cycles increased the load drops down by 35 %.

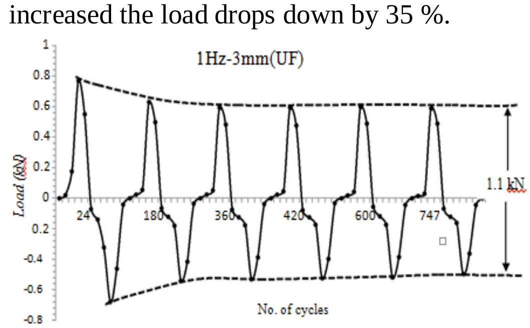


Figure 4.1.1 Unfilled polypropylene at 1Hz and 3mm deflection

Similarly figure 4.1.2 shows unfilled polypropylene loaded at 1.5 Hz and 2.5mm deflection. Gear was loaded up to failure. In the beginning load carried by single teeth was 1.24