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# Effect of Adjacent Teeth Load on Bending Strength of High Contact Ratio Asymmetrical Spur Gear Drive

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**ABSTRACT:**

*This paper describes methodology for predicting the bending stress of the spur gear accurately by including the load on the adjacent teeth for high contact ratio asymmetric spur gear drive. Higher contact ratio is obtained by enlarging the addendum from the standard addendum value where as the asymmetric is achieved by keeping various pressure angles (170, 200 and 220) at non drive side while the drive side pressure angle was kept as 200. The bending stress developed for the given load according to the load sharing calculated by using stiffness based method along with the effect of adjacent teeth loads are explored in this work. Computer aided design tool is used for generating the gear tooth profile and ANSYS is used to carry out the finite element analysis. The result shows that the maximum bending stress level in a mesh cycle is increased when the load on adjacent teeth are taken into account. The higher pressure angle at the non-drive side yields lesser stress at the fillet region when compared to the lower pressure angle.*

**KEYWORDS:**

*Asymmetric gear; High contact ratio; Load sharing ratio; Finite element analysis; Transmission system*

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**NOMENCLATURE:**

| | |
|---|---|
| HCR | High contact ratio |
| LSR | Load sharing ratio |
| NCR | Normal contact ratio |
| FLPDTC | First lowest point double tooth contact |
| FHPDTC | First highest point double tooth contact |
| SLPDTC | Second lowest point double tooth contact |
| SHPDTC | Second highest point double tooth contact |
| HPTC | Highest point tooth contact |
| LPTC | Lowest point tooth contact |

## 1. Introduction

Asymmetric gear has different pressure angles at the drive and non-drive sides of a gear tooth and it is suitable for the applications, where the torque is transmitted only one direction. Asymmetric involute gear is manufactured by the same process like symmetric gear [4]. These gears are lower in vibration and noise level then the symmetrical gear and increase the load carrying capacity and durability for the drive [6]. When pressure angle on the drive side is increased noise, contact stress and vibrations are reduced [3]. Asymmetric gears were used in aerospace propulsion drives and proved that it increased the power transmission density, increasing their load capacity and reducing size and weight [11]. It was also used in Tv7-117s turbo prop Russian air plane engine gear box. In which the asymmetric teeth were generated non-traditionally. Asymmetric gears also used in helicopter main drives [1] when the gears transmit power, the tooth subjected compressive stress at the non-drive side and tensile stress at the drive side.

So the tooth fracture always occurs at fillet of the drive side [13]. In high contact ratio gearing number of teeth pairs in contact alternates between three and two in a mesh cycle. It improved load distribution among the teeth in contact [2]. High contact ratios can be achieved by (1). smaller teeth (2). smaller pressure angle (3). Increased addendum [5]. Lower bending strength and increased tooth sliding are the advantages of high contact ratio gears. But gear manufactured should be high degree of precision [12]. Currently gearboxes for military tracked vehicle are having contact ratios ranging from 1.3-1.6; in which number of pair in contact alternate between is one or two [8]. HCR gears are having less gear mesh stiffness and transmission error than NCR gears [10]. In normal contact ratio on (NCR) gears the maximum load will act on the highest point of single tooth contact [15]. It is found that when the addendum increases the contact ratio and power loss increases and also when the pressure angle decreases there is an increase in contact ratio and power loss increases [14].

## 2. Geometry, mesh & boundary conditions

Gear tooth profile is made using mathematical equation in [7]. Both drive side and non-drive side involute profiles are imported into the drafting package to develop the fillets and root circle. Thus developed single tooth sector model of 20-20, 17-20 and 22-20 are shown in Figs. 1, 2(a) and 2(b) respectively. The full rim model is developed using cylindrical array concept in drafting package. Then the IGES model is imported into finite element analysis tool ANSYS. The action tooth area is divided into five area based on triple pair and double pair contact region in order to get the accurate fillet stress, the fillet area is also divided separate from the rest of the area. The typical transitions points of the gears and radius of typical contact region are developed based on the mathematical equations. The area is descritized by using four noded quadrilateral elements having 2 DOF at each node. Plane strain assumption is considered for this study. Mapped meshing is used to descritized the model. The finite element mesh of five teeth full rim model is shown in Fig. 3. The nodes at inner periphery of the gear rim (shaft hole) constrained in all directions. The normal load is resolved into radial and tangential load, and then applied to the model. Deflection of the tooth and stress at root fillet area has been observed by moving the load from LPTC to HPTC.

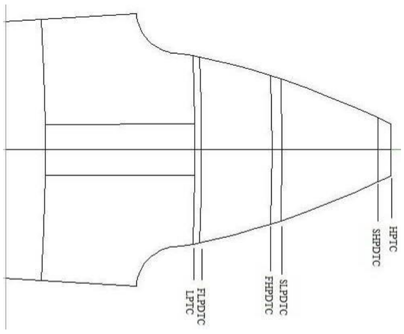
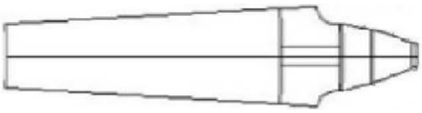



**Fig. 1: Single tooth profile 20°-20°**



**(a) 17°-20°**

**(b) 22°-20°**

**Fig. 2: Single tooth profiles**

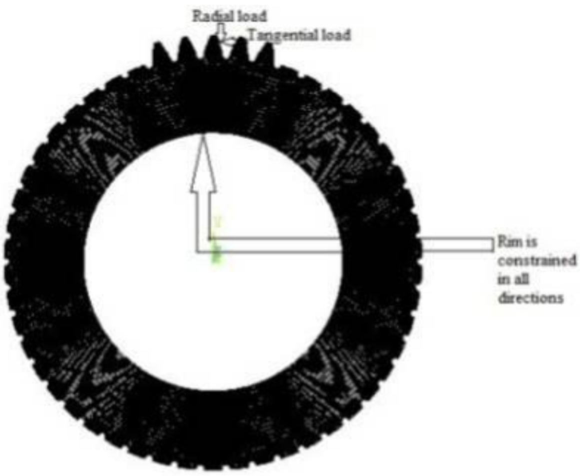



**Fig. 3: Five tooth full rim model**

## 3. Load sharing calculation

The procedure given in [9] is applied for load sharing ratio (LSR) calculations. Fig. 4 shows the deflection at the tip of the tooth due to the normal load. Since the tooth deflection is varying when the load is moving along the active profile. The stiffness also varies accordingly. The variation in stiffness will affect two load shared by a pair. The stiffness of the tooth is calculated as follows:

$$k = Fn/\delta \tag{1}$$

The single tooth stiffness has been mentioned as $k_p$ for pinion and $k_g$ gear. The individual tooth stiffness ($k_p$ and $k_g$) is used to found equivalent mesh stiffness ($k_{equ}$) of each contact pair as follows,

$$k_{equ} = \frac{k_{p1}k_{g1}}{k_{p1}+k_{g1}} \tag{2}$$

Similarly the equivalent stiffness of the other simultaneously contacting pair has been calculated. LSR is defined as the amount of normal load ($F_n$) shared by the one of the contact tooth pairs and calculated as,

$$LSR_1 = \frac{k_{equ1}}{k_{equ1}+k_{equ2}} \tag{3}$$

$$LSR_2 = \frac{k_{equ2}}{k_{equ1}+k_{equ2}} \tag{4}$$

Adjacent tooth loaded model is shown in Fig. 5.

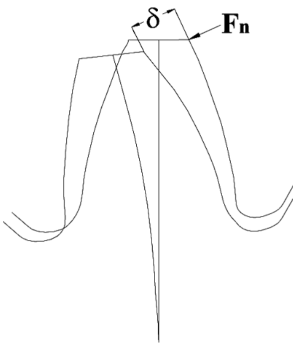



**Fig. 4: Deflection of the tooth due to normal load**

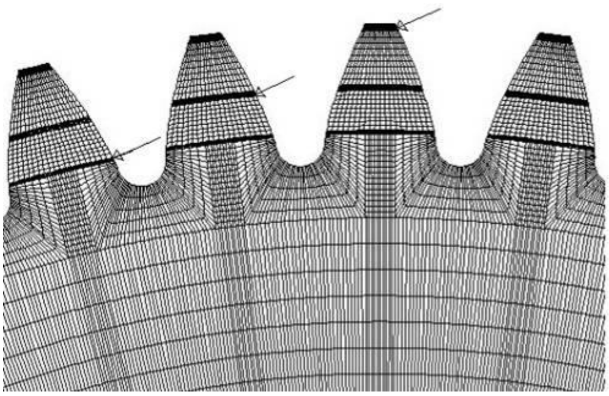


**Fig. 5: Adjacent tooth load at triple tooth contact**

## 4. Effect of adjacent teeth load

The tooth load sharing ratio calculated for 20°-20°gear pair is shown in Fig. 6. The deflection along the path of contact is shown in Fig. 7. It can be observed that the maximum load is acting at the FLPDTC and SHPDTC. When the full load is moving towards the HPTC, the deflection and stress are increasing due to increase in moment arm distance as observed in Figs. 8 & 9. In order to explore the effect of adjacent teeth load on the bending stress, a gear tooth is applied with the load based on load sharing ratio and adjacent teeth also loaded based on the load sharing ratio. The stress levels are observed in the fillet region. That the load in the loading tooth decreases the stress at the fillet region where as the load on the trailing tooth increase the fillet stress. There is a definite increase of about 2.5% in the maximum bending stress is noted when adjacent tooth also loaded. It can be seen from Fig. 9. This indicates that there is an influence of adjacent teeth load on the bending strength of the gear tooth. Hence, it has to be properly taken care while designing the gear drive.

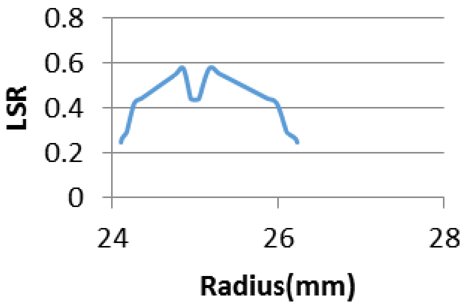



**Fig. 6: LSR vs. Radius (20°-20°)**

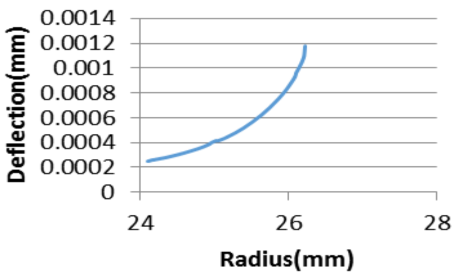



**Fig. 7: Deflection along the path of contact (20°-20°)**

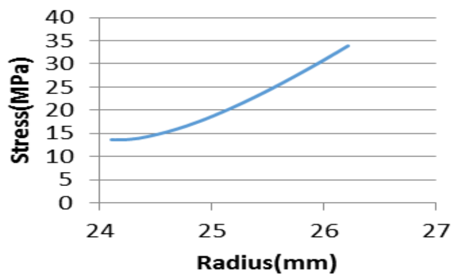



**Fig. 8: Fillet stress along the path of contact (20°-20°)**

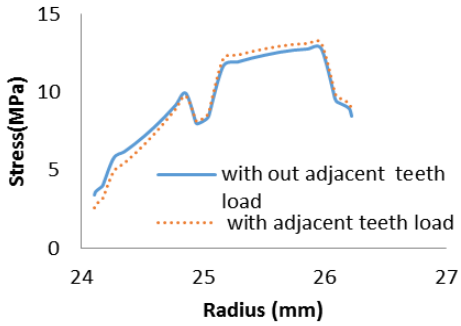



**Fig. 9: Effect of adjacent teeth load (20°-20°)**

## 5. Effect of asymmetry on the LSR and fillet stress

The load sharing ratio calculated for symmetric and asymmetric gears are compared in Fig. 10. There is not much change in the load shared by and pair along the mesh cycle. The fillet stress induced due to the load sharing based load and load on the adjacent teeth in symmetric and asymmetric gear is shown in Figs. 10 & 11. It can be observed from the Fig. that when the non-drive side pressure angle is less (17°). There is an increase in the stress level about 1.8% when compared to symmetric spur gear (20°-20°). When the non-drive side pressure angle is high (22°-20°) there is a reduction in the stress level about 1% which compared to symmetrical spur gear. This may be due to the increase in the sectional modules at the critical section for higher pressure angle at the non drive side.

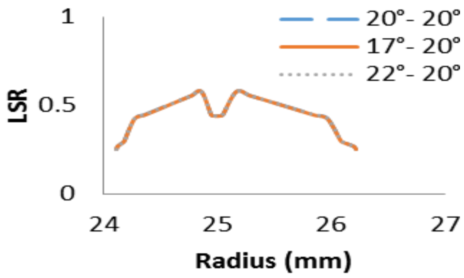



**Fig. 10: LSR for symmetric and asymmetric gear**

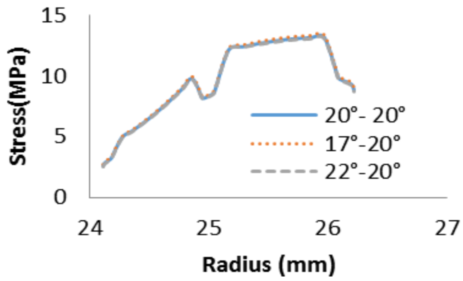



**Fig. 11: Stress at the tooth with load acting at the adjacent tooth**

## 6. Conclusion

The effect and important of the LSR and load on the adjacent teeth on the high contact ratio asymmetric spur gear drive has been explored in this work. There is an increase in the stress level at the fillet of about 2.5% when the adjacent teeth loads are considered for stress calculation. The higher pressure angle at the non-drive side yields lesser stress at the fillet region when compared to the lower pressure angle. Hence based on the static analysis results, it is recommended to higher pressure angle at the non-drive side for asymmetric spur gear drive.

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