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Estimation of Wear Depth on Normal Contact Ratio Spur Gear

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Abstract: Gear teeth mostly fail due to inadequate contact and bending strength. Tooth wear plays a vital role in the surface failure of a gear tooth. The present research work gives an idea to reduce the tooth wear and enhance the bending strength of the gear tooth. The Finite Element Model based contact stress analysis is carried out for estimating the tooth wear through nonstandard gears. Nonstandard gear is one whose tooth thickness at the pitch circle is not equal to 0.5πm. The Unbalanced tooth thickness at the pitch circle in pinion and gear provides a balanced bending strength between pinion and gear when the gear drive is loaded at the highest point of single tooth contact. Finally, the wear depth has been evaluated in both standard and nonstandard gears and compared for one mesh cycle.

Key words: Contact stress · Fillet stress · Finite element model · Tooth thickness · Wear depth

INTRODUCTION

Gears are the most important machine elements in the transmission system, which plays a major role in transmitting power and motion more effectively in various industrial applications. Tooth wear and tooth bending are the major failure modes in gear system. Excessive tooth wear leads to severe form of tooth failures, power loss andless efficiency of the system. Studies on wear characteristics (contact pressure, sliding distance and sliding velocity) become very important to enhance the performance of the gear drives.The first attempt made by the Anderson [1] on study of tooth mild wear for standard spur gear tooth. He has derived the various relations and expressions for estimating the wear depth in the spur gear tooth. Flodin and Andresson [2] have developed a numerical model for wear prediction based on a generalized Archard's wear equations. They reported that the wear depth is higher in the flank region of the pinion tooth. Muthuveerapan and Rama Thirumugan [3] have evaluated the theoretical wear depth and wear volume for normal and high contact ratio standard spur gear drives by using Archard wear equations. A precious attempt was made by Prabhu Sekar and Muthuveerapan [4] on balanced maximum fillet stresses on non-standard gear drives to improve the bending load capacity. Non-standard gear is one whose tooth thickness at the pitch circle is not equal to 0.5πm. In the standard gear drive, the tooth thickness at the pitch circle of the pinion and gear are equal to 0.5πm (Figure 1). Study of the wear characteristics on non-standard gears has not been attempted by any researchers. Hence, the present work provides an idea for evaluating the wear depth in standard and non-standard gear drives using finite element method.

Tooth Engagement Positions: In the NCR spur gear drives ($1 < \varepsilon < 2$), the contact begins, when the tip circle of the gear intersect with the line of action at A (Figure 2). The point A is the highest point of tooth contact (HPTC). The contact ends, when the tip circle of the pinion intersects the line of action at D. The point D is the lowest point of tooth contact. When the trailing pair makes contact at A, at the same time the leading pair makes contact at C (Lowest point of tooth contact). Once the trailing pair reaches to B (Highest point of single tooth contact), the leading pair leaves their contact from D. The regions AB and CD are called as double pair contact regions. The region BC is called as single pair contact regions. In the region BC, the only one pair takes the full load during the motion. The fillet stresses developed at the fillet region is maximum when the gear tooth loaded at the point B. So, the point B is called as critical loaded point by gear researchers.

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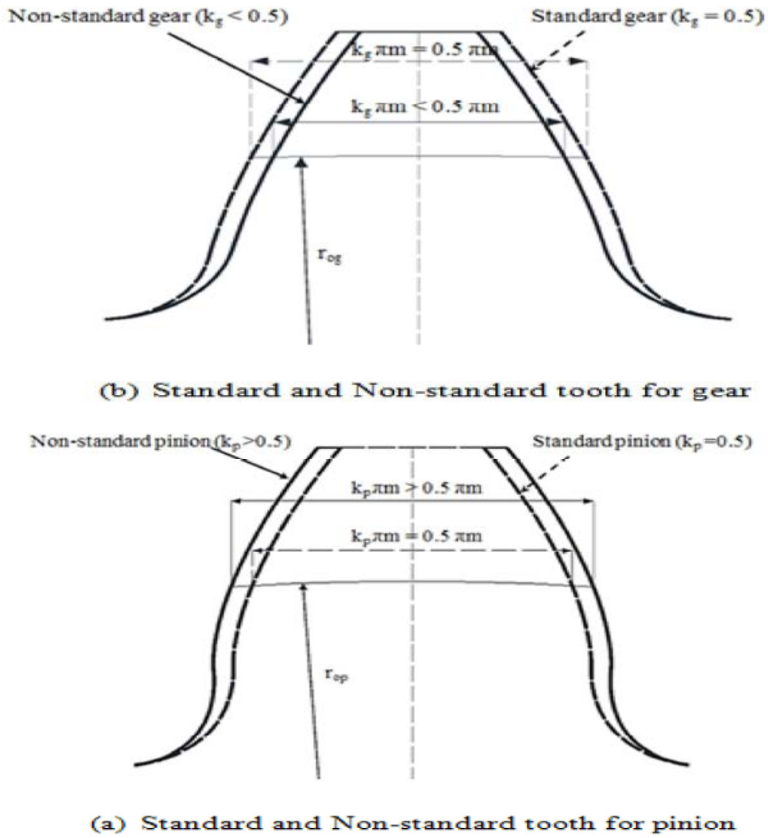


Fig. 1: Tooth thickness coefficients for standard and non-standard pinion and gear.

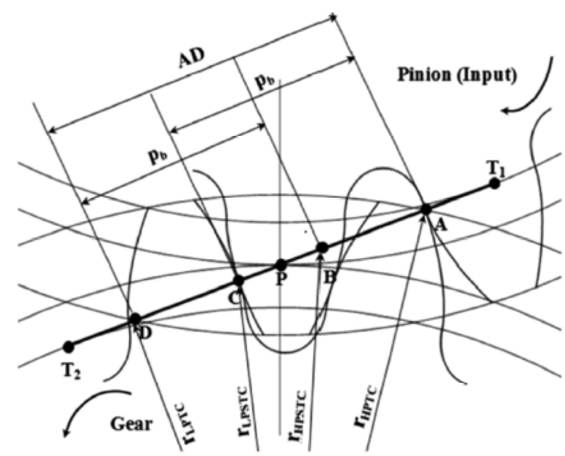


Fig. 2: Contact positions for NCR spur gears

The radius of various critical contact positionsfor gear are given [4] as

$$r_{HPTCg} = r_{ag} \quad (1)$$

$$r_{HPSTCg} = \sqrt{r_{bg}^2 + [(p_b - AD) + \sqrt{(r_{ag}^2 - r_{bg}^2)}]^2} \quad (2)$$

$$r_{LPSTCg} = \sqrt{r_{bg}^2 + [\sqrt{r_{ag}^2 - r_{bg}^2} - p_b]^2} \quad (3)$$

$$r_{LPTCg} = \sqrt{r_{bg}^2 + [\sqrt{r_{ag}^2 - r_{bg}^2} - AD]^2} \quad (4)$$

$$AD = \sqrt{r_{ap}^2 - r_{bp}^2} + \sqrt{r_{ag}^2 - r_{bg}^2} - C_o \sin\alpha_\alpha \quad (5)$$

Simulation of Wear Model: In this work, the single point observation procedure proposed by Flodin and Anderson is used to estimate the wear depth. The wear coefficient (k_w=5X10^{-10} mm^2/ N) is assumed as constant throughout the mesh cycle. The mean contact pressure (p_p) is ¾ of the maximum contact pressure (P_{max}) at the particular contact point. The maximum contact pressure and the actual normal load on the particular contact point are determined through the finite element analysis.The wear depth of the non-standard pinion and gear for one mesh cycle can be estimated by the following relations as such as

$$h_p = k_w p_p * 2a_w \; 1 - \frac{u_g}{u_p} \quad (6)$$

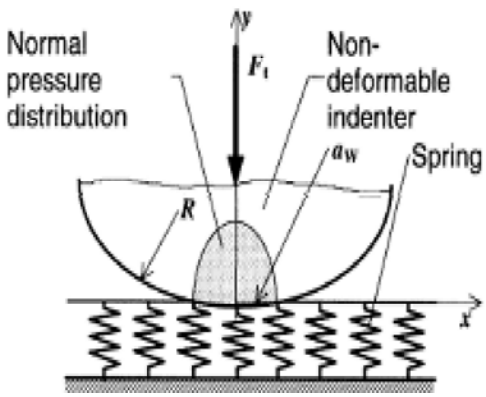


Fig. 3: Half contact width (a_w) in the contact region.

$$h_g = k_w p_p * 2a_w \left(1 - \frac{u_p}{u_g}\right) \tag{7}$$

where:
h - Wear depthfor one mesh cycle in mm,
a_w–Half contact width in mm
p_p-Mean contact pressure N/mm^2

The half contact width (a_w) is given by (Figure 3)

$$a_w = \sqrt{\frac{\frac{8F_n}{\pi b} * \frac{1-\gamma^2}{E}}{\frac{1}{R_p} + \frac{1}{R_g}}} \tag{8}$$

where, F_n normal load at particular contact point in N.
b – unit face width in mm
γ – poisson's ratio =0.3
E – Youngs modulus = 210 Gpa

If the tooth engagement is modeled as two cylinders in contact with the same peripheral velocity and sliding speed as the gear, the radius of the cylinders becomes:

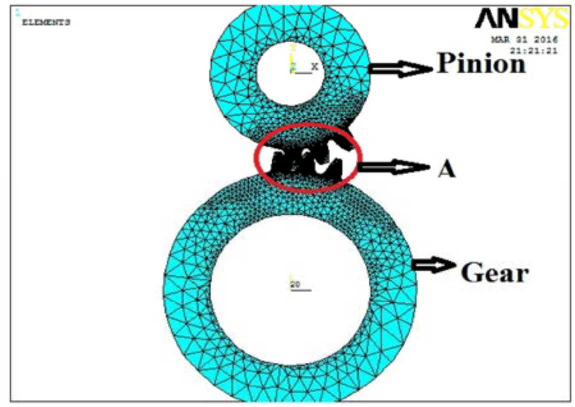
$$R_p = \frac{r_{op}}{2} \sin \alpha_o + y \tag{9}$$

$$R_g = \frac{r_{og}}{2} \sin \alpha_o - y \tag{10}$$

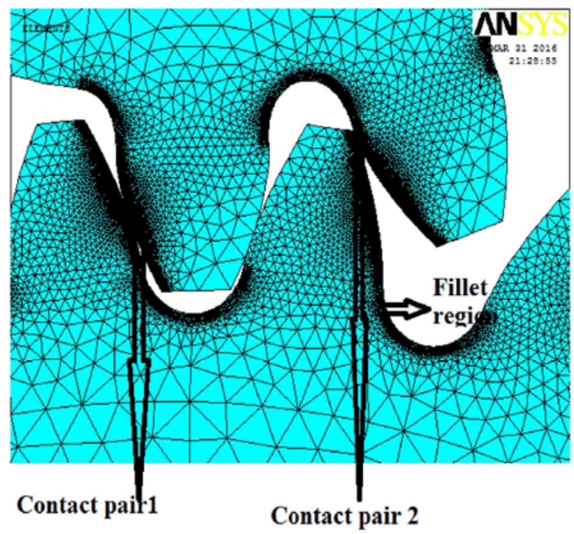
where, α_o is the pressure angle at pitch surface is distance from pitch point in mm.

The peripheral velocities of the pinion and gear flanks are:

$$u_p = \omega_p R_p \tag{11}$$



(a) Multi pair contact model



(b) Magnification of position A
Fig. 4: Finite Element Model of NCR Non-standard spur gears.

$$u_g = \omega_g R_g \tag{12}$$

ω_p and ω_g are the angular velocity of the pinion and gear

Finite Element Model: The FEM model has been developed with the given gear parameters m=1, α_o=20°, z_p=20, i=1.5, ha=1m, Np=150 rpm and F_{nt}=10 N for non-standard gear drive as shown in Figure 4. The contact and bending stresses has been evaluated using ANSYS 12 software. The 2D finite element model of spur gear displayed in Figure 4(a) is a three teeth full rim multi-pair contact model, which is used to carry out the analysis. A 2-D PLANE 82 element having two degrees of freedom per node has been used to mesh the gear model. Figure 4(b) shows the magnified view of the teeth contact.

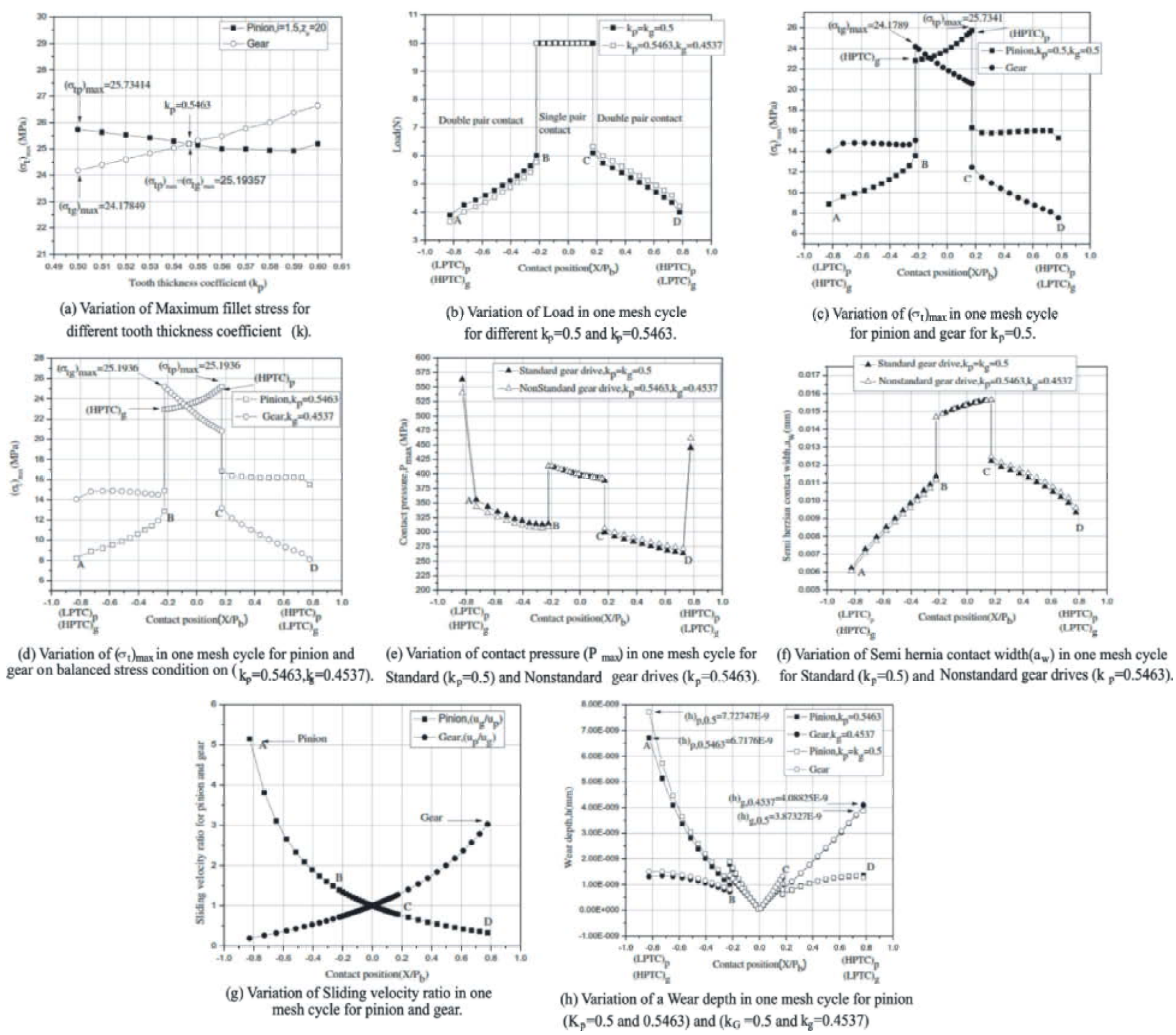
The following conditions and material properties are made in the present work.

- Assuming a uniform load distribution along the face width of the gear tooth, a 2D FE analysis is implemented based on the plane strain condition.
- The material is a linear elastic isotropic and homogeneous one.
- Tooth models for this analysis have been generated using a full round rack cutter.

RESULTS AND DISCUSSION

An accurate estimation of contact and fillet stresses require to enhance the performance of the gear drives. In many literature, It was found that the tooth load in the double pair contact (AB and CD) regions was taken as the half of the total normal load (F_{nAB}=F_{nCD}=50% F_{nt}). However, in the present work the actual tooth load shared by different contact pairs (single and double pairs contacts) and the respective contact and fillet stresses have been estimated through multi pair contact finite element analysis. The calculated results are shown in the Figure 5 (a –h). In the standard gear drive, the maximum fillet stress in the pinion is higher than that of the gear (k_p=k_g=0.5, $(\sigma_{tp})_{max}$= 25.73 MPa and $(\sigma_{tg})_{max}$=24.18 MPa). This unbalanced maximum fillet stresses developed in the standard pinion and gear are vanished in the non-standard gear drives for k_p=0.5463 and k_g=0.4537.

($(\sigma_{tp})_{max}$=$(\sigma_{tg})_{max}$=25.1936 MPa, Figure 5 (a)).



(a) Variation of Maximum fillet stress for different tooth thickness coefficient (k).

(b) Variation of Load in one mesh cycle for different k_p=0.5 and k_p=0.5463.

(c) Variation of $(\sigma_t)_{max}$ in one mesh cycle for pinion and gear for k_p=0.5.

(d) Variation of $(\sigma_t)_{max}$ in one mesh cycle for pinion and gear on balanced stress condition on (k_p=0.5463,k_g=0.4537).

(e) Variation of contact pressure (P_{max}) in one mesh cycle for Standard (k_p=0.5) and Nonstandard gear drives (k_p=0.5463).

(f) Variation of Semi hernia contact width(a_w) in one mesh cycle for Standard (k_p=0.5) and Nonstandard gear drives (k_p=0.5463).

(g) Variation of Sliding velocity ratio in one mesh cycle for pinion and gear.

(h) Variation of a Wear depth in one mesh cycle for pinion (K_p=0.5 and 0.5463) and (k_G =0.5 and k_g=0.4537)

Fig. 5: Influence of tooth thickness coefficient on Load, $(\sigma_t)_{max}$,a_w,sliding velocity ratio and wear depth in one mesh cycle for various contact position.

The variation of load shared by the pair during one mesh cycle for standard and non-standard spur gear drives are shown in Figure 5 (b). It is found from Figure 5 (b) that in the double pair contact regions (AB and CD), the trailing pair (at A) shares the load of 4 N (40% F_{nt}) and the simultaneous leading contact pair (at C) takes the remaining load of 6 N (60% F_{nt}) in the standard gear drives. It is also found that the only one gear pair shares the full normal load of 10 N in the single pair contact region BC. As the tooth thickness coefficient k_p increases, the load shared by the trailing pair decreases in the region AB and increases in the another simultaneous contact pair region CD. The variation of maximum fillet stress developed in the fillet region on standard and non-standard gears are shown in Figure 5 (c and d). It is noticed that the maximum fillet stresses developed in the fillet region between the pinion and gear are balanced in the non-standard gear drives, when the gear drive is loaded at the highest point of single tooth contact (Figure 5 (d)). As k_p increases, the contact pressure (P_{max}) nd half contact width (a_w)decreases in the region AB (flank of the pinion makes contact with face of the gear) and increases in the region CD (face of the pinion makes contact with flank of the gear), which is mainly due to decrease in the load share in the region AB and increases in the region CD (Figures 5 (e, f and b)).

The sliding velocity ratio for pinion and gear are plotted in the Figure 4 (g). From the Figure 4 (g), it is noted that the sliding velocity ratio is always higher in the flank regions of the pinion and gear. It is also inferred that the sliding velocity ratio is maximum in the flank of the pinion (at A, beginning of the contact) than that of the gear flank (at D, ending of the contact). This is because of that the maximum sliding distance exists at A than that of D from the pitch point. Based on the finite element study made on wear, it is observed that the maximum wear depth occurs at the beginning and the end of the contact (Figure 5 (h)) because of the maximum sliding distance. Due to increase in the k_p values, the maximum wear depth decreases at the flank region of the non-standard pinion, because of the lower values of load share and the corresponding contact pressure at the region. Hence, the non-standard gear drive is an advantageous one than that of standard gear drives related to their bending strength, contact strength and wear behaviour.

CONCLUSION

In the present study, a detailed investigation has been carried out on the contact stress, wear depth and the balanced maximum fillet stress for standard and non-standard symmetric spur gears using multi pair contact model. Through this study on non-standard pinion and gear, it is concluded that the use of unequal tooth thickness at the pitch circle of the pinion and gear is justified as one of the possible solutions for the enhancement of fillet bending strength, contact strength and reduction wear depth of the NCR spur gear drives.

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