Prediction of Theoretical Wear in High Contact Ratio Spur Gear Drive

G Muthuveerappan, Rama Thirumurugan

Abstract

Gear contact surface wear is one of the important failure modes in gear systems. The procedure implemented in this work includes finite element multi pair contact modeling to predict the load sharing between the simultaneously contacting pairs and mathematical programming method to estimate the gear surface wear in high contact ratio gear drives for particular number of cycles. A method to find out the new tooth profile and the worn out volume after some particular number of wear cycles is also presented. The parametric study based on the pre established center distance and also for the fixed pitch circle radius are discussed. The variation of wear depth at various contact point along the path of contact shows the maximum wear depth at the beginning and end of the contact in gear drive, but the highest wear occurs at the dedendum flank portion or root portion of tooth surface.

Keywords: Wear, Finite element analysis, High contact ratio, Spur gear.

1 Introduction

Wear and the associated material loss can lead not only to gear tooth failure, but also it can lead to changes in vibration and noise behavior, produces non uniform gear rate, alter the dynamic effects and decreases the efficiency. In addition, wear can change the patterns of gear contact such that the altered load distributions and contact stresses will accelerate the occurrence of other failure modes such as pitting and scoring. Gear wear debris can also be detrimental to the performance of bearings or other components of a drive system.

Many researchers have attempted to predict the gear wear using analytical and numerical tools. For most of the studies, the well-known Archard's wear equation in conjunction with gear contact model and relative sliding calculations was used.

Andersson[1] appeared to be the first one to make the detailed study on the gear tooth mild wear. He derived the expressions to calculate the slip distance of a point on the pinion flank when it slides against the opposite mating flank during a mesh action under boundary lubrication or dry running condition. These expressions were used by Flodin and Andersson [2] to calculate the sliding distance from involute spur gear profile geometry. They have used single point observation method and simplified Winkler's mattress model to predict the wear in the normal contact ratio (NCR) gear tooth in which the contact ratio (ϵ) is less than two. They have later extended their spur gear wear prediction methodology to helical gears by slicing the helical gear in the face direction into narrow spur gear segments staggered according to the helix angle[3].

Rama Thirumurugan (Corresponding author), E-mail:rama.thirumurugan@gmail.com
G. Muthuveerappan, E-mail:jmvr@iitm.ac.in
Machine Design Section, Department of Mechanical Engineering,
Indian Institute of Technology Madras, Chennai-36.

Wu and Cheng[4] derived the formula for equivalent wear rate and tooth wear profiles along the line of action considering gear dynamics and rough elastohydrodynamic lubrication (EHL). Results show that most materials are removed from both the addendum and dedendum tooth surfaces and that the highest wear occurs at the beginning of an engagement. This high wear region corresponds to the root of the driving (pinion) teeth and the tip of the driven (gear) teeth. Brauer and Andersson [5] presented a mixed FE and analytical method to simulate wear in NCR spur gear tooth. FE model was used by them to calculate the gear contact load, then Hertzian equation was used to find the contact pressure and contact width.

Bajpai et al. [6] proposed a wear model for NCR spur and helical gears that combines a FE based gear contact mechanics model that predicts contact pressures, a sliding distance computation algorithm and Archard's wear formulation to predict wear of the contacting tooth surfaces.

Despite the abundant literature available on the evaluation of the wear in the NCR gear drives, the calculation of wear in the high contact ratio (HCR) gear drives in which $\epsilon > 2$ has been rarely studied. This work explores the effect of various gear parameters on the wear of HCR spur gears. The typical contact points along the path of contact and the evaluation of load sharing between simultaneously engaging pairs using finite element method are done based on the author's previous works([7] and [8]). The sliding wear at the contact point of the two mating surfaces are calculated based on the assumption that the surfaces are mixed or boundary lubricated, the hardness values of the material is constant throughout the wear cycle and the stiffness of the contacting pair at a particular point is not affected as the wear progress.

2 Wear Simulation Model

The sliding wear in local contact can expressed by the Archard's wear equation and it is given as[2]

$$\frac{V}{s} = K\frac{F}{H} \tag{1}$$

where, V is the volume of the worn out material, s is the sliding distance of a point i slides against an interacting tooth flank during one mesh, F is the applied normal load, H is the hardness of the contacting surfaces and K is the dimensionless wear coefficient.

The above equation is modified by dividing the both sides of the equation by the apparent contact area

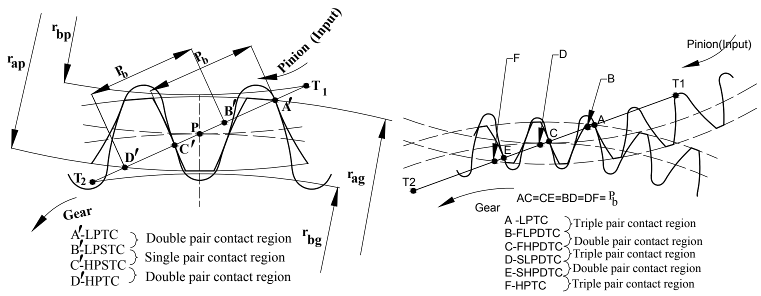
$$\frac{h}{s} = k_w p_H \tag{2}$$

where, h is the wear depth, p_H is the contact pressure and k_w is the dimensional wear co efficient and it is equal to K/H

In this work, the single point observation procedure proposed by Andersson and Eriksson [9] is used to find the wear depth and k_w is assumed as constant through out the mesh cycle while also the p_H is constant at the particular observation point. In this procedure, the mean contact pressure ($(p_m)_i$), the half contact width (a_i) and the sliding distances are calculated for the observed contact point (i) in the mesh cycle. The accumulated wear depth is calculated using the following equation.

$$h_{i,new} = h_{i,old} + k_w (p_m)_i 2a_i \left\{ \frac{[(v_p)_i - (v_g)_i]}{(v_p)_i} \right\} \quad (3)$$

where v is the sliding velocity,subscript p and g denotes pinion and gear respectively The above equation is used to carry out the wear simulation and parametric study on spur gears. The typical contact points during the mesh cycle for NCR and HCR gear drive is given in Figs.1a and 1b.



(a) NCR gear (b) HCR gear

Figure 1: Typical contact points along the path of contact.

2.1 Wear depth calculation for NCR and HCR gears

The input speed and power considered in this study are 150 rpm and 5kW respectively. The wear coefficient k_w is taken as $5X10^{-16}m^2N^{-1}$ [2]. The mean contact pressure is taken as 3/4 of the maximum pressure [10]. Total number of wear cycles considered in this study is 50000. The gear parameters considered for this study is given in Table.1. The wear depth calculated for each 1000 cycles for a NCR gear drive is given

Table 1: Gear parameters

Parameters	Value	Parameters	Value
Gear ratio (i)	1	Addendum modification (x)	0
Tooth number (z)	50	Dedendum (h_f)	h_a+0.25m
Module (m)	1	Cutter tip radius (ρ_r)	0.3m
Pressure angle (α_o)	20^o	Rim thickness (t_R)	5m
Addendum (h_a)		Material elastic modulus (E)	210 GPa
—NCR Gear pair	1m	Poissonś ratio (ν)	0.3
—HCR Gear pair	1.223m		

in the Fig.2a. Based on the theoretical study made on the wear, the maximum wear depth occurs at the beginning and end of contact in NCR gear tooth, but the highest wear occurs at the lowest point of tooth contact (LPTC) of pinion tooth surface because of the maximum sliding distance. These trends agree well with the experimental evidence of Dhanasekaran and Ganamoorthy[11], Walton and Goodwin[12], etc., as long as mild abrasive wear is prevalent. Also there is an increase in the wear depth at highest point of single tooth contact (HPSTC) and lowest point of single tooth contact (LPSTC) are noted for NCR gear drive because of the sudden increase in the load at these points. At the pitch point region there is no wear observed due to zero sliding.

The simulated wear depth in HCR gear drive is plotted in Fig. 2b. From this simulated wear depth, like in NCR gear, a similar trend of higher wear depth at the tooth root flank surface is observed compared to tooth tip face. But maximum wear depth occur only at first lowest point of double teeth contact (FLPDTC (B)) for HCR gear tooth. There is a reduction in wear depth observed on both the sides around the pitch point, which is due to the triple pair contact and it is zero at the pitch point.

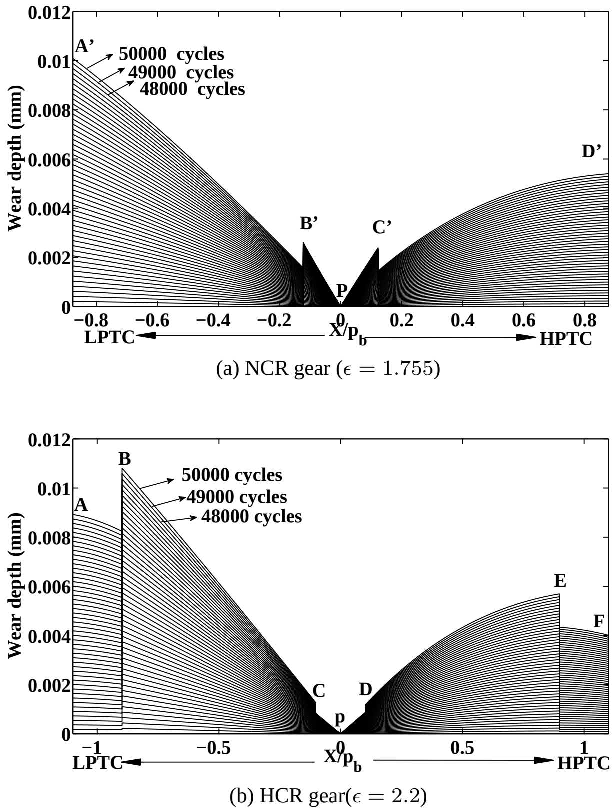


Figure 2: Wear depth for each 1000 cycles for NCR and HCR gears.

2.2 Wear volume calculation

The worn out volume or material loss is calculated from the estimated wear depth for 50000 cycles. Each contact point has been moved by a distance of wear depth along the line of action. The coordinates of the new point $i'(x'_i,y'_i)$ on the worn out profile are calculated from the tooth layout shown in Fig.3 and it is given as

$$x'_i = xi - hcos(\theta'_i) \tag{4}$$

$$y'_i = y_i - hcos(\theta'_i) \tag{5}$$

The new profile after 50000 cycles along with the original gear profile is constructed

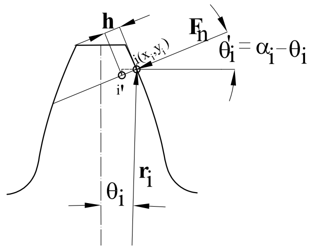


Figure 3: Tooth layout to calculate the worn out profile

in a CAD package and the area between these two curves has been calculated. Wear volume is calculated by multiplying gear face width with the calculated area.

3 Parametric Study on HCR Gear Drive with Fixed Center Distance

In this work the center distance(C_o) is fixed as 50mm and the effect of addendum (h_a), pressure angle (α_o) and their combined effect on wear depth are discussed.

3.1 The effect of addendum

For the different values of h_a ranging from 1m to 1.29m, the maximum wear depth and the amount of material loss are compared and discussed. The maximum wear depth occurs at the point LPTC (A') for the gear tooth with $\epsilon \leq 2$ where as for the $\epsilon > 2$, the maximum wear depth occurs at the point FLPDTC (B) (Fig.4a). It is observed that the wear depth is increasing with respect to an increase in the h_a up to $\epsilon \leq 2$, then it is decreasing for the further increase in h_a due to the reduction in the load shared by the pair at this point, which causes the reduction in maximum contact pressure. It is also observed that the wear volume is increasing for an increase in h_a(Fig.5a).

3.2 Effect of pressure angle

For different values of α_o from 14.5^o to 22.5^o, the calculated maximum wear depth and wear volume are compared. It can be observed from the figure that an increase in

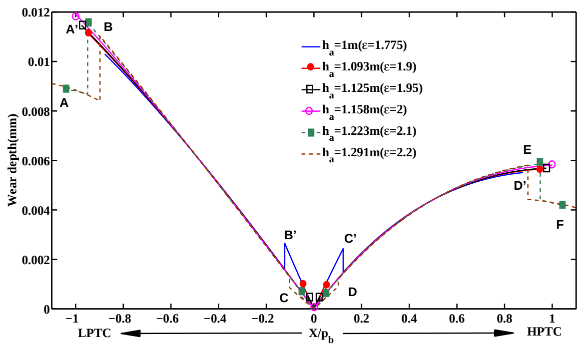
α_o reduces the wear depth. The maximum wear depth occurs at LPTC (A) for a gear tooth with lower α_o where as, it occurs at FLPDTC (B) for a gear tooth with higher α_o (Fig.4b). A decrease in wear volume is observed as α_o increases (Fig.5b)

3.3 Effect of pressure angle and addendum

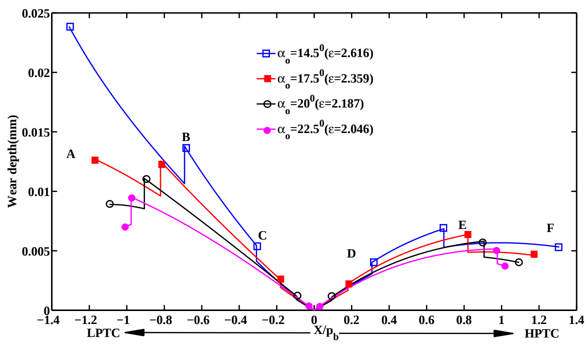
For a fixed contact ratio 2.1 and different combination h_a and α_o, the simulated wear depths and wear volume for a few cases considered in this study are shown in Figs.4c and 5c. Reduction in wear depth and volume are observed for a simultaneous increase in h_a and α_o. The maximum wear depth is noted at FLPDTC (B) for all the cases.

4 Effect of Teeth Number(z)

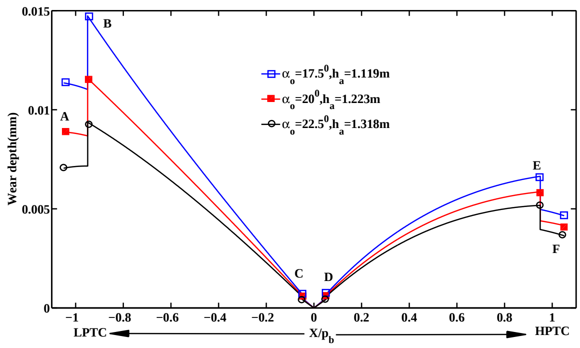
By keeping the pitch circle as 50mm, the teeth numbers are changed and the wear depth and volume are determined for the three cases. The decrease in z increases the wear depth and volume (Figs.4d and 5d).



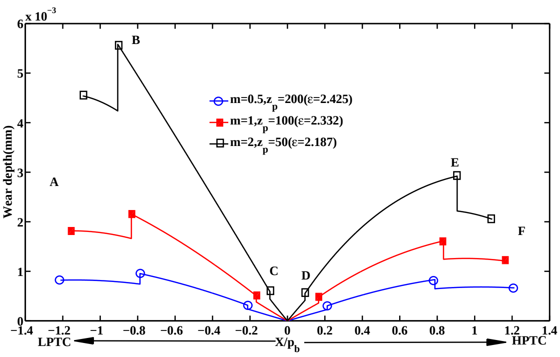
(a) Effect of addendum



(b) Effect of Pressure angle

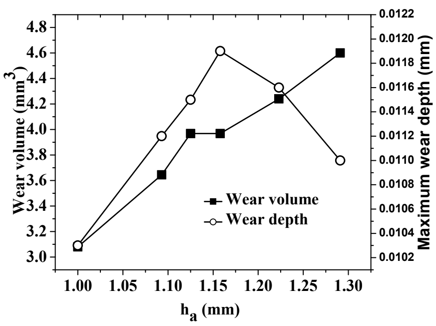


(c) Combined effect of Pressure angle and addendum

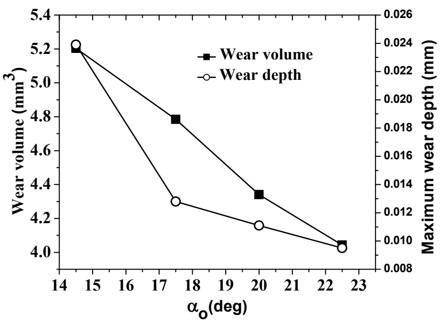


(d) Effect of teeth number

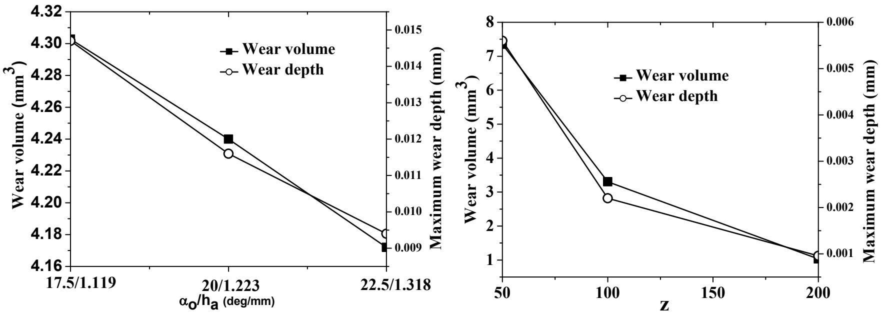
Figure 4: Effect of gear parameters on wear for 50000 cycles



(a) Effect of addendum



(b) Effect of Pressure angle



(c) Combined effect of Pressure angle and addendum (d) Effect of teeth number

Figure 5: Calculated maximum wear depth and wear volume

5 Conclusions

Based on the theoretical study made on the wear, the maximum wear depth occurs at the beginning and end of the contact in NCR gear drive, but the highest wear occurs at the dedendum flank portion. Also an increase in the wear depth at HPSTC and LPSTC is noted for NCR gear drive because of the sudden increase in the load at these points. When HCR gear is achieved by increasing the h_a, the maximum wear depth is noted only at FLPDTC and it is also noted the depth of wear is reduced around the pitch point region due to the triple pair contact. The wear is more for gear tooth having higher h_a due to higher sliding. The maximum depth of wear occurs at LPTC for lower α_o and it is at LPDTC for higher α_o of HCR gear. The HCR gear drive with higher α_o and h_a has shown less wear compared to gear tooth with lesser α_o and h_a. The HCR gear drive with a fixed pitch circle radius, gear with lesser z generates higher wear. An increase in α_o decreases the amount of wear.

References

[1] S. Andersson, *Partial EHD theory and intial wear of gears*. PhD thesis, Royal institute of Technology, Stockholm, 1975.

[2] A. Flodin and S. Andersson, "Simulation of mild wear in spur gears," *Wear*, vol. 207, no. 1-2, pp. 16–23, 1997.

[3] A. Flodin and S. Andersson, "Simulation of mild wear in helical gears," *Wear*, vol. 241, no. 2, pp. 123–128, 2000.

[4] S. Wu and H. Cheng, "Sliding wear calculation in spur gears," *Journal of tribology*, vol. 115(3), p. 493, 1993.

[5] J. Brauer and S. Andersson, "Simulation of wear in gears with flank interference– a mixed fe and analytical approach," *Wear*, vol. 254, no. 11, pp. 1216–1232, 2003.

[6] P. Bajpai, A. Kahraman, and N. Anderson, "A surface wear prediction methodology for parallel-axis gear pairs," *Journal of tribology*, vol. 126, no. 3, pp. 597–605, 2004.

[7] R. Thirumurugan and G. Muthuveerappan, "Maximum fillet stress analysis based on load sharing in normal contact ratio spur gear drives," *Mechanics based design of structures and machines*, vol. 38, no. 2, pp. 204–226, 2010.

[8] R. Thirumurugan and G. Muthuveerappan, "Critical loading points for maximum fillet and contact stresses in normal and high contact ratio spur gears based on load sharing ratio," *Mechanics Based Design of Structures and Machines*, vol. 39, no. 1, pp. 118–141, 2011.

[9] S. Andersson and B. Eriksson, "Prediction of the sliding wear of spur gears," *Proceedings of NORDTRIB*, vol. 90, 1990.

[10] F. Karpat and S. Ekwaro-Osire, "Influence of tip relief modification on the wear of spur gears with asymmetric teeth," *Tribology Transactions*, vol. 51, no. 5, pp. 581–588, 2008.

[11] S. Dhanasekaran and R. Gnanamoorthy, "Gear tooth wear in sintered spur gears under dry running conditions," *Wear*, vol. 265, no. 1-2, pp. 81–87, 2008.

[12] D. Walton and A. Goodwin, "The wear of unlubricated metallic spur gears," *Wear*, vol. 222, no. 2, pp. 103–113, 1998.