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Research Paper

CONSTRAINTS AFFECT THE OPTIMIZATION OF BENDING STRESS AT CRITICAL SECTION OF ASYMMETRIC SPUR GEAR TOOTH

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The main objective of this paper is to show that various constraints affect the optimization of bending stress. In this paper various constraints are discussed. For specific gear tooth parameter analytical calculation are done and give recommendation as per IS standards.

Keywords: Contact ratio, Critical section thickness, Asymmetric spur gear and constraints, Tip thickness

INTRODUCTION

In gears are the most common means of transmitting power in the modern mechanical engineering world. A gear can be defined as a machine element used to transmit motion and power between rotating shafts by means of progressive engagement of projections called teeth. Gears have a wide variety of applications which vary from a tiny size used in watches to the large gears used in lifting mechanisms and speed reducers. They form vital elements of main and ancillary mechanisms in many machines such as automobiles, tractors, metal cutting machine tools etc. In recent times, the gear design has become a highly complicated and comprehensive subject. A designer of a modern gear drive system must remember that the main objective of a gear drive is to transmit higher power with comparatively smaller overall dimensions of the driving system which can be constructed with the minimum possible manufacturing cost, runs reasonably free of noise and vibration, and which required little maintenance. He has to satisfy, among others the above conditions and design accordingly, so that the design is sound as well as economically viable.

ASYMMETRIC SPUR GEAR TEETH

An asymmetric spur gear drive means that larger and smaller pressure angles are applied for the driving and coast sides. The two profiles

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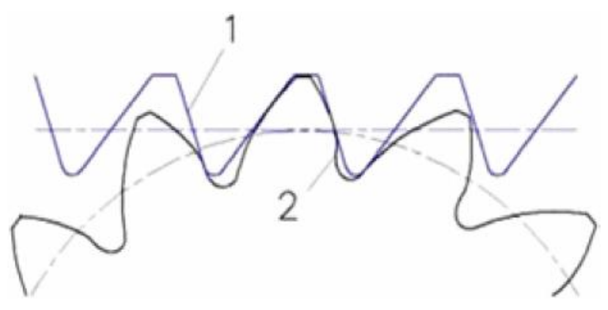
(sides) of a gear tooth are functionally different for many gears. The workload on one profile is significantly higher and is applied for longer periods of time than for the opposite one. The design of the asymmetric tooth shape reflects this functional difference.

The design intent of asymmetric gear teeth is to improve the performance of the primary contacting profile. The opposite profile is typically unloaded or lightly loaded during relatively short work periods. The degree of asymmetry and drive profile selection for these gears depends on the application.

Figure 1: Asymmetric Spur Gear



CONSTRAINTS WHICH AFFECT OPTIMIZATION OF BENDING STRESS

As the pressure angle on drive side increases, the bending stress reduces at critical section of asymmetric spur gear. But Decision on maximum magnitude of drive side pressure angle is constraint by the safe contact ratio. As the pressure angle on drive side increases, the bending stress reduces at critical section of asymmetric spur gear and Decision on maximum magnitude of drive side pressure angle is constraint by the safe contact ratio and Gear standard procedures such as IS, recommended that, its value is 1.1. Bellow this value, the loading period of a single gear tooth pair significantly increases, which is undesirable under cyclic loading conditions (Singh Vedang and Senthilvelan, 2007; Niels, 2010; Moya *et al.*, 2010; Alexander Kapelevich and Yuriy Shekhtman, 2010; Konstandinos *et al.*, 2010; and Prajapati and Vaghela, 2013). As the pressure angle on drive side increases, the bending stress reduces at critical section of asymmetric spur gear and Decision on maximum magnitude of drive side pressure angle is constraint by the top land tip thickness and gear standard procedures such as IS, recommended that, its value is should be greater than equal to 0.2 times the module for the hardened gears. Bellow this value, tip thickness decreases and tip becomes too sharp, more and more pointed (Singh Vedang and Senthilvelan, 2007; Niels, 2010; Moya *et al.*, 2010; Alexander Kapelevich and Yuriy Shekhtman, 2010; Vaghela and Patel, 2011a, 2011b and 2011d; and Prajapati and Vaghela, 2013). As the pressure angle on drive/coast side increases, the bending stress reduces. (Mallesh *et al.*, 2009; and Niels, 2010). The largest reduction in the bending stress can be found with (pressure angle on drive side profile) ~*d* > ~*c* (pressure angle on coast side profile).

ANALYTICAL CALCULATION OF CONSTRAINTS

For this analysis solutions following gear tooth parameters are used. Gears are used to transmit a power of 18 KW at 1600 rpm.

Contact ratio is a measure of the average number of teeth in contact during the period in which a tooth comes and goes out of contact with the mating gear. Contact Ratio is the ratio of the arc of action to the circular pitch.

Table 1: Gear Tooth Parameter

S. No.	Description	Value
1.	Pressure angle, coast side	200 fixed
2.	Pressure angle, drive side	200-400 increment by 20
3.	Number of teeth	25 and 47
4.	Module	4 mm

Figure 2: Positions of Tooth Load Variation

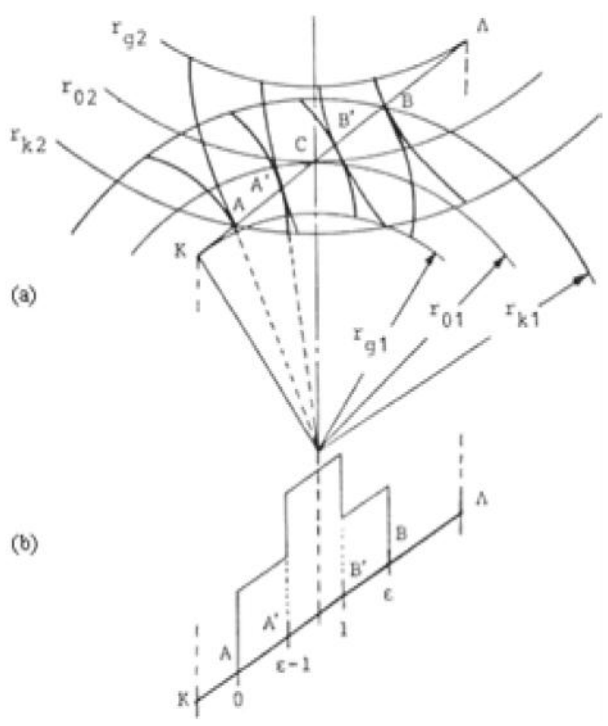


Figure 2 show that position of tooth load variation. A'B' show that this duration only a single pair of gear teeth is in contact which is subject to the maximum load. If contact ration is very low this duration becomes high which is not desirable.

$$CR = \frac{\sqrt{r_{t1}^2 - r_{b1}^2} + \sqrt{r_{t2}^2 - r_{b2}^2} - C.\sin ¤d}{p.\cos ¤d}$$

Contact ratio for different pressure angle on drive side profile of asymmetric gear tooth was achieved by above equations.

Table 2: Contact Ratio of Asymmetric Spur Gear Teeth for Different Pressure Angle on Drive Side Profile

Pressure Angle on Drive Side Profile of Asymmetric Spur Gear in Degree	Contact Ratio
20	1.679016
22	1.5940664
24	1.5214836
26	1.4596723
28	1.407323
30	1.3633682
32	1.3269453
34	1.297367
36	1.2740965
38	1.256730
40	1.2449829

To calculate tip thickness for different pressure angle on drive side profile of asymmetric spur gear tooth following equation are used. Thickness of tooth at any radius *r* thickness of tooth is calculated by following equations.

$$S_r = S_{rd} + S_{rc}$$

where,

$$S_{rd} = r.\sim$$

$$S_{rc} = r.\sim$$

$$\sim = \frac{®}{2.z} + inv¤ - inv¤_r$$

$$¤_r = \cos^{-1}\left(\frac{r_p}{r}.\cos ¤\right)$$

With help of above equation obtained the tip thickness of asymmetric spur gear tooth which are listed in bellow table.

The largest reduction in the bending stress can be found with (pressure angle on drive

Figure 3: Representation of Angle ~ at Tip Radius

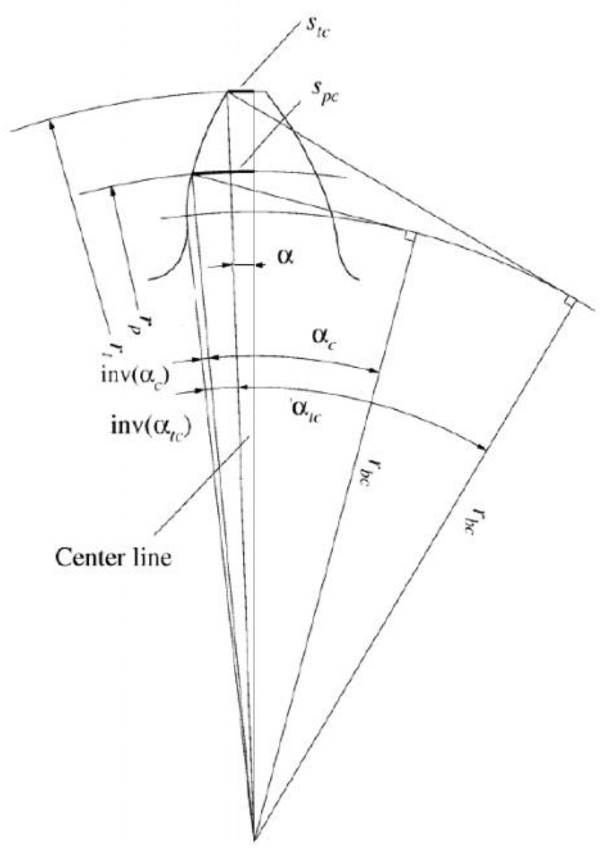


Table 3: Tip Thickness Obtained from Analytical Calculation

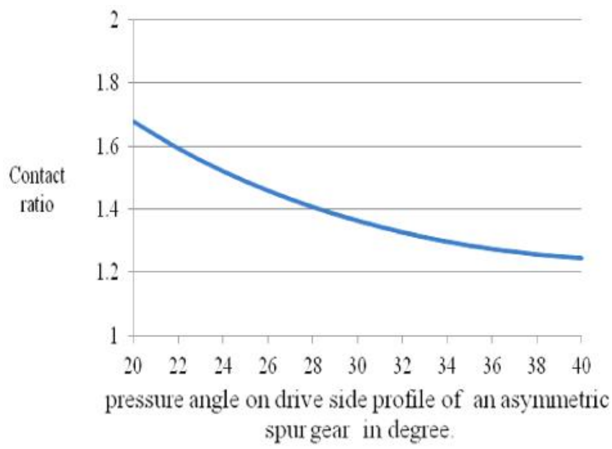
S. No.	Asymmetric Gear of Different Pressure Angle of Drive Side Profile	Tip Thickness of Spur Gear in mm
1.	20	2.662812431
2.	22	2.530329326
3.	24	2.390176814
4.	26	2.242216251
5.	28	2.086155863
6.	30	1.921555886
7.	32	1.747827740
8.	34	1.564227332
9.	36	1.369842319
10.	38	1.163572744
11.	40	0.944104088

side profile) ~*d* > ~*c* (pressure angle on coast side profile) which is also constraint to optimization of bending stress at critical section of asymmetric spur gear tooth.

RESULTS AND DISCUSSION

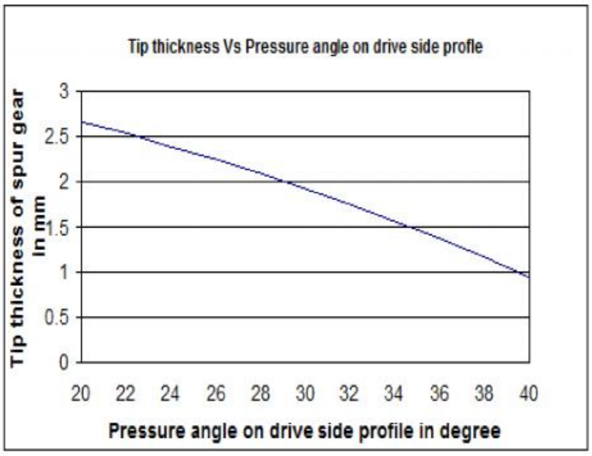
From above analytical analysis obtain graph pressure angle on drive side profile of asymmetric spur gear vs. contact ratio is shown bellow.

Figure 4: Pressure Angle on Drive Side Profile of Asymmetric Spur Gear vs. Contact Ratio



As the pressure angle on drive side profile of asymmetric spur gear increases, a contact ratio also decreases which is also not desirable things. If contact ratio is very less than loading period on single tooth is long which is not desirable. The loading period of a single gear tooth pair significantly increases, which is undesirable under cyclic loading conditions. Gear standard procedures such as IS, recommended that the contact ratio should be greater than equal to 1.1. Higher (up to certain limit) pressure angle on drive side profile of asymmetric spur gear is beneficial for bending stress at critical section of asymmetric spur gear.

Figure 5: Graph Gear Tip Thickness vs Pressure Angle on Drive Side Profile



From above analytical analysis obtain graph for different pressure angle on drive side profile of asymmetric spur gear vs. tip thickness is shown bellow.

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 of gears. On the other hand, tooth flank becomes more curved as the pressure angle increases. Result table and graph shows the decrease in top land width with increase in pressure angle up to the upper limit. The results obtained are for the following parameters, module = 4, number of teeth in the pinion = 25 and pressure angle on coast side = 20°.

For this case module is 4 mm and as par IS, recommended that the tip thickness should be greater than equal to 0.2 times the module of gears. So, tip thickness should be greater than equal to 0.8 mm. From the result table and graph it is observed that above the pressure angle 40° tip thicknesses becomes bellow 0.8 mm. Hence for this case limitation of increasing the pressure angle is 40°.

CONCLUSION

Various constraints affect the optimization of bending stress which is discussed. For specific gear tooth parameter analytical calculation are done and give recommendation as per IS standards. It is necessary to optimize all affected constraints otherwise create diverse effect.

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