FACTOR AFFECTING THE BENDING STRESS AT CRITICAL SECTION OF ASYMMETRIC SPUR GEAR

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ABSTRACT

The main objective of this review paper is to show how the different parameters are affecting the bending stress at critical section of asymmetric spur gear. Bending stress at critical section is most important parameter in gear design. It must be low as low possible. Our try to minimize it by optimize all affected Parameters of asymmetric spur gear tooth to reduce Bending Stress at critical section of tooth. This reduction can translate into Increased Load Capacity, Size and Weight Reduction, Longer Life, Cost Reduction, Increased Reliability, Noise and Vibration reduction, Increased Gear Efficiency and Maintenance Cost Reduction etc.

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 and tooth peaking effect. These way parameters are affecting directly or indirectly on performance. There are so many parameters are likes Contact ratio, Top land thickness, Pressure angle on drive side profile, Pressure angle on coast side profile, Asymmetry factor, No. of teeth, Interference, Undercut, Centre distance, Gear ratio, Critical section thickness, Profile shift of pinion, Profile shift of gear, Module, Bending stress at critical section, Optimal fillet radius and Balance stress etc. affects the performance. So, it is necessarily to optimize these affected Parameters of asymmetric spur gear tooth to reduce Bending Stress at critical section of tooth.

Key words: Asymmetric spur gear tooth, parameters, critical section thickness.

INTRODUCTION

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. It has to satisfy, among others the above conditions and design accordingly, so that the design is sound as well as economically viable.

The two profiles (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.

An asymmetric spur gear drive means that larger and smaller pressure angles are applied for the driving and coast sides. The two profiles of a gear tooth are functionally different for most gear drives. The workload on one side of profile is significantly higher than the other Gears.

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.

The difference between symmetric and asymmetric tooth is defined by two involutes of two different base circles $D_{bd}$ and $D_{bc}$. The common base tooth thickness does not exist in the asymmetric tooth. The circular distance (tooth thickness) $S_p$ between involute profiles is defined at some reference circle diameter $D_p$ that should be bigger than the largest base diameter.
Kapelevich has developed a method for design of gears with asymmetric teeth. It is found that direct gear design procedure for the synthesis of asymmetric gear pair using an area of existence method and designed the rack cutter parameters to generate these desired asymmetric gear tooth profiles. It is found that asymmetric tooth geometry (with larger pressure angle on drive tooth side) allow for an increase in load capacity while reducing weight and dimensions for same types of gears.[1] Litvin et al. have proposed a modified geometry of asymmetric gear pair as a combination of an involute gear and a double crowned gear. The contact and bending stresses for the driving side (with a larger pressure angle) of an asymmetric spur gear drive are reduced. Modified geometry of asymmetric spur gears for reduction of transmission errors and localization of bearing contact.[2] Alexander L. Kapelevich and Yuriy V. Shekhtman found Optimization of the fillet profile allows reducing the maximum bending stress in the gear tooth root area by 10–30%. It works equally well for both symmetric and asymmetric gear tooth profiles. This bending stress reduction can be traded for Size and weight reduction, longer life, higher load application, Cost reduction (less expensive materials, heat treatment, etc.), lower noise and vibration. [3] Alexander L. Kapelevich and Thomas M. McNamara also observe Direct Gear Design results in 15-30% reduction in stress level when compared to traditionally designed gears. This reduction can be translated into Increased Load Capacity (15-30%), Size and Weight Reduction (10-20%), Longer Life, Cost Reduction, Increased Reliability, Noise and Vibration Reduction (finer pitch, more teeth will result in higher contact ratio for the given center distance), Increased Gear Efficiency (1-2% per stage), Maintenance Cost Reduction and Other benefits Depending on the Application. Direct gear design for asymmetric tooth profiles provides additional opportunity for improvement of gear drives with unidirectional load cycles that are typical for many mechanical transmissions.[4] V. Senthil Kumar et al. have observed With this choice of design, the full optimization with respect to bending stress is not found. The positive effect on the bending stress is then solely related to a root thickness increase of the tooth, but the stress can be further reduced using shape optimization to reduce the stress concentration. The influence of several non-standard asymmetric rack cutters are designed to accommodate different combinations in the values of pressure angle, top land thickness ratio, profile shift, speed ratio and the asymmetric factors on the maximum fillet stress has been analyzed to suggest the optimum values of these parameters that improve the fillet capacity in bending.[5] Singh Vedang et al. have found As the pressure angle on drive side increases, the bending stress decreases and bending load capacity increases. Decision on maximum magnitude of drive side pressure angle is constraint by the safe contact ratio and tooth peaking effect.[6] Alexander L. Kapelevich has developed Direct Gear Design is an alternative approach to traditional gear design. It is not constrained by predefined tooling parameters. It allows for the analysis of a wide range of parameters for all possible gear combinations in order to find the most suitable solution for a particular custom application. This gear design method can exceed the limits of traditional rack generating methods of gear design.

Direct Gear Design results in a 15-30% reduction in stress level when compared to traditionally designed gears. This reduction can be translated into Increased Load Capacity (15-30%), Size and Weight Reduction (10-20%), Longer Life, Cost Reduction, Increased Reliability, Noise and Vibration reduction, Increased Gear Efficiency and Maintenance Cost Reduction.[7] Th. Costopoulos, V. Spitas et al. have In this paper the concept of the asymmetric one-sided -involute gear teeth was introduced and studied using FEA. Due to this concept increase load carrying capacity and combine the good meshing properties of the driving involute and the increased strength of non-involute curves. The geometry of the proposed gears was investigated and the generation process was modeled using the Theory of Gearing. The novel design incorporates two main innovative features, which are the substitution of the standard trochoidal fillet with a circular fillet for the reduction of the stress concentration at the driving side and the use of a fully rack or hob-generated, addendum geometry for the coast side. The increase in load carrying capacity can reach up to 28% compared to
the standard 20-involute teeth.[8] Niels L. Pedersen was investigated. In the papers on asymmetric tooth design there seems to be two choices: either αd<αc, i.e., the drive side pressure angle is smaller than the coast side pressure angle or αd>αc. The choice made is αd>αc is stated that this choice is made because it reduces the mesh stiffness that has a positive influence on noise and vibration levels. The choice made is αd<αc and here it is shown that this only has a little influence on the mesh stiffness. The present paper does not discuss mesh stiffness and both choices of relative pressure angles are shown in the examples in order to explore the possible advantages. When increasing pressure angles the negative result is that the top land thickness becomes smaller and in the limit it becomes pointed, with a further increase in the pressure angle the teeth become shorter. A shorter teeth decrease the contact ratio, which is undesirable. The minimum top land thickness limit is sa>0.25 M, for carburized gears the limit is reported to be higher sa>0.4 M. The largest reduction in the bending stress can be found with αd>αc. With a drive side pressure angle, αd=36° (AE1), the bending stress reduction compared to the standard ISO tooth is about 40% independent of the number of teeth on the gear. Maximum difference as compared to an optimized tooth is 3% and With a coast side pressure angle, αc=34° (AE2), the bending stress reduction compared to the standard ISO tooth is about 18% independent of the number of teeth on the gear. Maximum difference as compared to an optimized tooth is 5%. [9] Moya J.L et al. found a theoretical analysis of a procedure to determine the Lewis Factor, which can play a major role in the fracture of asymmetric plastic gear teeth. The Lewis factor for different coefficient of asymmetry is calculated for different number of teeth and it is found that Lewis factor increases with coefficient of asymmetry and number of teeth. [10] Alexander Kapelevich and Yuriy Shekhtman presents a unique approach and methodology to define the limits of selection for gear parameters. The area within those limits is called the “area of existence of involute gears”. This paper presents the definition and construction of areas of existence of both external and internal gears. The isograms of the constant operating pressure angles, contact ratios and the maximum mesh efficiency (minimum sliding) isograms, as well as the interference isograms and other parameters are defined. An area of existence allows the location of gear pairs with certain characteristics. Its practical purpose is to define the gear pair parameters that satisfy specific performance requirements before detailed design and calculations. An area of existence of gears with asymmetric teeth is also considered. [11] Alex Kapelevich describes an alternative Direct Gear Design approach for the asymmetric gear design, demonstrating the basic gear tooth and mesh parameter definition. It also familiarizes with a proprietary tooth fillet profiles optimization technique, providing minimum bending stress concentration. [12] G. Mallesh et al. Estimate the critical section for different pressure angles and backup ratios using computer programme and compare the results obtained by other researchers. Developed programme is used to create a finite element model for symmetric and asymmetric spur gear tooth to study the effect of bending stress at the critical section for different backup ratios. The rim thickness was varied and the location and magnitude of the maximum bending stresses were reported and results obtained were compared with the Lewis bending equation.[13] G. Mallesh et al. study the effect of bending stress at the critical section for different pressure angles on the drive side along with the profile shift. Comparison has been made for symmetric and asymmetric spur gear tooth using Lewis equation and Finite element analysis software. With the effect of positive shift there is an increase in the tooth thickness at the critical section. With the effect of positive shift there is a reduction in the bending stress at the critical section. [14] Dr. Alexander L. Kapelevich presents definitions of main inspection dimensions and parameters for directly designed spur and helical, external and internal gears with symmetric and asymmetric teeth. [15] G. Mallesh et al. Generate asymmetric spur gear tooth geometry for different pressure angles on drive and coast side using computer programme. Developed programme is used to create a finite element model of gear tooth to study the effect of bending stress at the critical section for different pressure angles, different number of teeth and module. To study the effect of above asymmetric spur tooth parameters Finite Element Analysis software ANSYS was used. As the number of teeth and module increases the bending stresses
decreases, while the other parameters are unchanged. This is due to the fact that with increase of module, the pitch diameters of the gear tooth increases causing the tooth to become bulkier and stronger. Again with the increase of module, the fillet radius of the tooth increases which would cause less impact in the root region (critical section) of the gear tooth. The increase of module means the increase of the tooth width from top to bottom, as a result the stress is observed to be less in the wider tooth for the same loading. The bending stresses decreases with increases in the number of teeth and pressure angle on drive side, with the application of the same load for all gear teeth and one with more teeth, i.e. the bigger one will be stressed lesser. It is observed that as the pressure angle on the drive side increases bending stress decreases.[16] A. Kapelevich and Y. Shekhtman found the gear tooth fillet is an area of maximum bending stress concentration. This paper presents a fillet profile optimization technique for gears with symmetric and asymmetric teeth based on FEA and a random search method. It allows achieving substantial bending stress reduction in comparison with traditionally designed gears. The bending stress reduction provided by the fillet optimization can be converted into other gear performance benefits, such as contact stress reduction and increased gear mesh efficiency higher load capacity, longer lifetime, lower noise and vibration and cost reduction.[17] Konstandinos G. Raptis et al. shows how to calculate highest point of single tooth contact - HPSTC using both numerical and experimental methods. At this point single pair of gear teeth is subjected to the total load which is most unfavorable contact point because maximum stress at gear tooth root are generate.[18] Vaghela P A and Patel D A shows in this paper is to Improvement of Load Carrying Capacity by increases the pressure angle on drive side profile of asymmetric spur gear constraint by the safe contact ratio and tooth peaking effect. Bending stress at the critical section reduces drastically with increases in the pressure angle on the drive side for certain limit. Load carrying capacity of gear increasing from 2150 N to 3820 N without compromising a bending stress 41.73 MPa at critical section of spur gear tooth root.[19] Vaghela P A and Patel D A observed that after the modification by increases the pressure angle on drive side profile of asymmetric spur gear constraint by the safe contact ratio and tooth peaking effect. Bending stress at the critical section reduces drastically with increases in the pressure angle on the drive side for certain limit. Axial thickness is optimized from 40 mm to a 26.25mm without compromising a bending stress 41.73 MPa at critical section of spur gear tooth root.[20] Vaghela P A and Patel D A obtain optimal line, on this line point gives optimize value for improvement of load carrying capacity and reduction in weight without compromising bending stress at critical section of spur gear tooth root. Bellow this optimal line zone is feasible or safe and above this optimal line zone is not feasible or unsafe. On this optimal line, left most point on load 2150 N gives only axial thickness reduction without improvement in load carrying capacity and right most point on load 3820 N gives only improvement in load carrying capacity without axial thickness reduction. Intermediate point between these two points gives a partially reduction in axial thickness of gear (weight reduction) and partially improvements in the load carrying capacity in such a way that it satisfy limit of bending stress not more than 41.73 MPa at critical section of spur gear tooth root. [21]

**CONCLUDED REMARKS**

Bending stress at critical section is most important parameter in gear design. It must be low as low possible. Our try to minimize it by optimize all affected Parameters of asymmetric spur gear tooth to reduce Bending Stress at critical section of tooth. This reduction can be translated into Increased Load Capacity (15-30%), Size and Weight Reduction (10-20%), Longer Life, Cost Reduction, Increased Reliability, Noise and Vibration reduction, Increased Gear Efficiency and Maintenance Cost Reduction. For custom gear following parameters are affects its performance so it is necessary to optimizing it. **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 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. [6, 9, 10, 11, 18]Top land tip thickness: - 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 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. [6, 9, 11, 19, 20, 21]Pressure angle on drive side profile: - As the pressure angle on drive side increases, the bending stress reduces at critical section of asymmetric spur gear. [9, 13, 14, 16]Pressure angle on coast side profile: - As the pressure angle on drive side increases, the bending stress reduces at critical section of asymmetric spur gear. [9, 13, 14, 16]The largest reduction in the bending stress can be found with (pressure angle on drive side profile) \( \alpha_d > \alpha_c \) (pressure angle on coast side profile). Because, With a drive side pressure angle \( \alpha_d = 36^\circ \) the bending stress reduction compared to the standard ISO tooth is about 40% independent of the number of teeth on the gear and With a coast side pressure angle, \( \alpha_c = 34^\circ \) the bending stress reduction compared to the standard ISO tooth is about 18% independent of the number of teeth on the gear. [9]Asymmetry factor: - The concept of the asymmetry factor was incorporated to cater for asymmetry. It is the relation between the driving side profile and the coast side profile angles. Asymmetry factor= driving side profile angle/ coast side profile angles. [10]No. of teeth: - The minimum number of teeth which a pinion can have to mate with a rack without interference and undercutting can be calculated using equation. \( z = \frac{2}{\sin \alpha^2} \) If minimum number of teeth condition is not satisfied interference and undercut problems occurs. As the pressure angle on drive side increases, it allow to reduce the minimum number of teeth so at critical section the bending stress reduces. [16]Interference :-When the gear tooth tries to dig below the base circle of mating gear then the gear tooth action shall be non conjugate and violate the fundamental law of gearing this non conjugate action is called the interference. [11, 14]Undercut: - A condition in generated gear teeth when any part of the fillet curve lies inside of a line drawn tangent to the working profile at its lowest point. [11, 14]Critical section thickness:-Thickness of the critical section increases with increase in pressure angle, the bending stress reduces at critical section of asymmetric spur gear as critical section thickness increases. The gear becomes bigger and having more resistance to the load as critical section thickness increases. [14]Profile shift of pinion:-With the effect of positive shift there is an increase in the tooth thickness at the critical section. Increasing the pressure angle, the bending stress at the critical section decreases for a given profile shift value. With the effect of positive shift there is a decrease in the bending stress at the critical section. [14]Profile shift of gear:-With the effect of positive shift there is an increase in the tooth thickness at the critical section. Increasing the pressure angle, the bending stress at the critical section decreases for a given profile shift value. With the effect of positive shift there is a decrease in the bending stress at the critical section. [14]Module:-As module increases the gear becomes bigger and having more resistance to the load. As the module increases the bending stresses decreases, while the other parameters are unchanged. This is due to the fact that with increase of module, the pitch diameters of the gear tooth increases causing the tooth to become bulkier and stronger. Again with the increase of module, the fillet radius of the tooth increases which would cause less impact in the root region (critical section) of the gear tooth. The increase of module means the increase of the tooth width from top to bottom, as a result the stress is observed to be less in the wider tooth for the same loading. Thickness of the critical section increases slightly with increase in pressure angle, and directly depends on the module. It is the fact that as module increases the gear becomes bigger and having more resistance to the load. [14, 16]Optimal fillet radius: - It is also known as a Fillet Profile Optimization. In this our aim is to minimizing bending stress concentration along the fillet. The gear tooth fillet is an area of maximum bending stress concentration. A fillet profile optimization technique for gears with asymmetric teeth based on FEA and a random search method us to minimizing bending stress concentration along the fillet. It allows achieving substantial bending stress reduction in comparison with traditionally designed gears. [3, 4,
Balance stress:- Bending Stress Balance - achieving equally strong gears by adjusting the tooth thicknesses at the operating pitch diameters. An iteration method combined with FEA is used. Bending stress balance allows equalizing the tooth strength and durability for the pinion and the gear. Bending stress at critical section: - Bending stress at critical section is most important parameter in gear design. It must be low as possible. Our try to minimize it by optimize all affected Parameters of asymmetric spur gear tooth to reduce Bending Stress at critical section of tooth. This reduction can be translated into Increased Load Capacity, Size and Weight Reduction, Longer Life, Cost Reduction, Increased Reliability, Noise and Vibration reduction, Increased Gear Efficiency and Maintenance Cost Reduction etc.

So, it is necessarily to optimize these affected Parameters of asymmetric spur gear tooth to reduce Bending Stress at critical section of tooth.

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