Optimization of Addendum Modification for Bending Strength of Involute Spur Gear

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Abstract—

In the present work the effects of different values of addendum modification on the gear root stress in involute spur gears presented. The series of addendum modification co-efficient from higher value that is positive value to lower value that is negative value through zero is considered. Analysis is done theoretically and experimentally concerning the effect of addendum modification on the gear root stresses. Addendum modification calculated as per DIN 3992.Tooth Bending tests performed on spur gears with various amounts of addendum modification, using a universal testing machine of hydraulic type. Gear models with different addendum modification generated using kiss-soft software. Test results validated with the analysis software.

Keywords - **Addendum modification, root stress, universal testing machine, kiss-soft, Ansys**.

I. INTRODUCTION

The displacement form cutter center to gear center is called as profile shift or addendum modification. The addendum modification is called positive when the displacement is away from the gear center and it is considered as negative when in the direction towards the center of the gear [1].

Addendum modification also called as profile shift. One can say of profile shift as merely shifting the gear towards or away so that the active profile of the teeth uses a different sector of the same involute curve. Profile shift does not need new cutters if straight sided hobs are used (the usual case). The hob is shifted towards or away to shift the profiles and the outer diameters of the gears are modified to accommodate the profile shifts. Therefore, the advantages of profile shifting are achieved without any extra cost addition. Only form cutting like as milling requires a different and new cutter for profile shifted gears. It is because milling-cutting or grinding require a cutter or grinding wheel with a profile which matches the shape of the space between teeth. Since profile shifting changes the shape of the tooth space, the shape of the form cutter must change.

The normal pressure angle measured on the generating pitch diameter (the same as the standard pitch diameter or pitch circle diameter) is determined by the cutter pressure angle and is usually 20° for most gears. It is independent of profile shift. Module is also important parameter of spur gear and states that increasing the module results in to increasing dynamic stress [2].One of the most important parameters in any gear design is the number of teeth in the smaller gear, and profile shift has its greatest significance when the pinion has a small number of teeth. In fact, the first use of profile shift was to avoid undercut in pinions with less than 17 teeth.

Gears with minimum teeth are more susceptible to profile shift, and their tooth shape changes more considerably, in another case profile shift changes tooth shape only little for gears having a large of teeth. In fact, the teeth of a rack (with do not change shape as the teeth are profile shifted.

When gears are manufactured by a gear generation, the datum line of the basic rack profile need not necessarily form a tangent to the reference circle of the manufactured gear. The tooth form may be changed by moving the datum line from the tangential position. The involute shape of the gear tooth profile is retained and the effect is merely to use parts beyond from or closer to the origin of the involute.

If profile correction is used only for avoiding undercut, the formulae given in equation (1) can be used. If, contrary, other parameters as stated above are comes in picture which demand profile shifting, the total profile correction should be appropriately distributed between the pinion and the gear. For ease some equations (2),(3) relevant to profile correction are repeated here after required transposing.

Besides the mentioned equations, allocation of gear correction factors can also be done as per the steps given in IS: 3756 and DIN 3992[1] which are basically the same. These standards are valid for spur and helical gears belonging to the standard basic rack and having number of teeth 10 or more.

$$x = 1 - \frac{z}{2}\sin^2 a \quad \text{……. (1)}$$

$$x_1 + x_2 = \frac{(z_1+z_2)\,(inv\ \alpha_{tw} - inv\ \alpha t)}{2\,tan\ \alpha} \quad \text{…… (2)}$$

$$cos\ \alpha_{tw} = \frac{m_t\,(Z_1+Z_2)}{2a}\ cos\ a_t \quad \text{…….. (3)}$$

James Kuria John Kihiu [2] states that increasing the addendum is one of the options for increasing the contact ratio since this can be achieved by simply adjusting the cutter depth. The maximum permissible addendum modification coefficients can be achieved by changing the addendum modification coefficient of the pinion and gear until the top land thickness of gear teeth is equal to the lowest permissible (usually 0.3m).

Ivan Atanasovska1, Vera Nikolić-Stanojević [3] gives influence of addendum modification coefficient value on the load capacity of involute gear by using FEM method. The correct selection of addendum modification coefficients can directly affect on improvement of characteristics of gear pair working that guide to increasing gear life-time, also it reduces vibrations and shocks during working of gear pair.

Mihai Bănică,[4] Dipl. Eng., Lecturer gave The guesstimate of dynamic behavior and gear noise of spur gear. The choosing correct addendum modifications are very important, because these affect heavily on geometrical and resistance parameters of the spur gearing. The addendum modifications have a crucial and significant influence on the dynamic behavior of the involute spur gear.

Satoshi OD and Taktto [5] here represented study on the addendum modification's effects on root stresses of spur gear and of helical gears. Bending fatigue tests were performed on helical gears with several amounts of addendum modification, using a hydraulic bending fatigue testing machine. In the case of involute helical gears, the increase in bending strength because of positive addendum modification is much smaller than that in the case of spur gears measure advantages in this procedure greatly improved without appreciable hanging their dimensions and the undercut of the gears with a small number of teeth can be avoided.

Ivana ATANASOVSKA and Dejan MOMCILOVIC [6] studied the comparative diagram of tooth profile in terms of spur gear tooth bending strength. The described procedure suggests research of tooth profile parameters impact on bending strength of involute gears tooth. The diagrams are used for building the groups of comparative diagrams which will give simultaneously selection of optimal addendum modification coefficients and root curvature radius for a specific gear pair with aspect of tooth bending strength. The tooth profile parameters that have noteworthy impact at tooth root stresses and tooth bending strength are: addendum modification and root curvature radius. They have developed a method for selection of addendum modification for pinion and wheel with optimum bending stresses.

V. Atanasiu, I. Doroftei [7] presents a dynamic tooth load analysis of spur gears with addendum modifications. In his Study, the dynamic model includes the non-linear time varying mesh stiffness, variable tooth profile errors, and tooth profile modifications in mandate to get consistent data for the forecast of gear dynamic loads. A study of the two group of the assessment methods of the mesh stiffness used in the dynamic analysis of spur gear pairs.

II. Gear Root stresses

According ISO 6336-3-2006, [8] Method B the local tooth root stress is determined as the product of nominal tooth root stress and a stress correction factor. This method includes the theory that the determinant tooth root stress happens with application of load at the outer point of single pair tooth contact of spur gears or of the virtual involute spur gears. However, in the latter case, the "transverse load" shall be replaced by the "normal load", applied on the face width of the actual gear of interest.

For gears having virtual contact ratios in the range $2 \leq \varepsilon\alpha n \leq 2.5$, it is presumed that the determinant stress occurs after application of load at the inner point of triple pair tooth contact. In ISO 6336 [8], this assumption is taken into consideration by the deep tooth factor, Y_{DT}. In the case of involute helical gears, the factor, Y_β, taken in to accounts for deviations from these assumptions .Method B is suitable for broad calculations and is also suitable for computer programming and for the analysis of pulsator tests (at a given point of load application).The total tangential load in the case of gear trains with numerous transmission paths (planetary gear trains,) is not quite equally spread over the individual meshes (depending on design, tangential speed and manufacturing accuracy).This is to be taken into consideration by introducing a mesh load factor, $K\gamma$, to follow K_A in Equation (4), in order to adjust as necessary the average load per mesh.

$$\sigma_F = \sigma_{F0} \, K_A K_V K_{F\beta} K_{F\alpha} \qquad \text{……….. (4)}$$

$$\sigma_{F0} = \frac{F_t}{b m_n} \, Y_F Y_S Y_\beta Y_B Y_{DT} \qquad \text{……….. (5)}$$

III. Experimental setup

Figure 1 shows the representative test setup for determining the gear root stresses. Gradual load will be applied and corresponding deformation is recorded. The load from the load cells present on the Universal Testing Machine will be applied gradually. Display attached to the machine will give corresponding results.

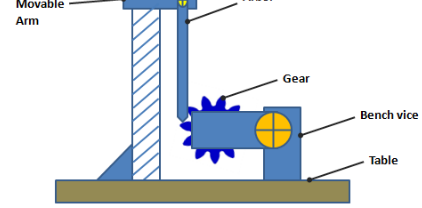

Figure 1 Experimental setup

Experimental Setup Information-

The specs for the UTM (Universal Testing Machine) used for the Test are as below:

Make: Star Testing System (India) - Software based
Model No: SPS 248
Type: DC Servo Control
Measuring capacity – 600 KN
Range -0-600 KN
Least count – 0.006 KN
Distance between columns – 600mm
Piston stroke – 250mm
HP – 2.5
Speed for loading: 5mm/min to 500mm/min
Plunger dia. - 50mm
Overall dimension – 200X800X2400 mm
Weight – 3000 Kg

Strain Gauge
Type - Fixed Gauge
Gauge length - 25 to 50mm
Strain gauge positioned with the terminal

Table 1 Gear specification

Module	5 mm
No of teeth	25
Pressure Angle	20 °
Face width	50 mm
Gear Rim ID	30 mm
Material	18CrNiMo 7-6
Yield strength	850 Mpa
Tensile strength	1200 Mpa
Young's modulus	206000 Mpa
Poisson's ratio	0.3
Hardness	65±1 HRC

The test specimen is prepared as per DIN standard on gear hobbing machine .Table 1 shows the gear specification. Only the common specification are mentioned in table 1.Value for different addendum modification have mentioned in table 2. Each test specimen has same module and number of teeth. Test specimen differs in gear teeth addendum modification. Different addendum modifications are achieved by changing machine setting. In this project to manufacture the test specimen conventional hobbing machine is used. Test specimens have been manufactured using 18CrNiMo 7-6 grade material. This is low carbon steel and are be hardened by case carburizing to case depth of 0.8 to 0.9 mm. Manufactured specimens are be tested on Universal testing machine to get tooth bending strength. Gear is mounted in bench vice. A forged mandrel are be used to fix in moving arm of Universal testing machine. Gear tooth and mandrel are having line contact. Using hand calculation the maximum load is decided and according to that, given load will be applied on universal testing machine. Using experimental results a numerical model will be developed by using ANSYS and then Experimental and Numerical results will be compared. To get spur gear model to use in Ansys with different addendum modification will be generate using kiss-soft software.

KISS-soft program is used for sizing, optimizing and recalculating designs for different components of machine such as shaft, gears and bearings, clutch, welding joints springs, joining elements and belts, screws. In addition to strength verification in line to the related standards numerous optimization functions are also presented. Naturally all important geometry calculations are also executed, reference dimensions are provided for manufacturing, and form of the tooth is decided in two and three dimensions.

IV. RESULT

The gears with different modification modeled in kiss-soft. Exported file of kiss-soft will resemble exact size and shape of gear with all parameters. Gear from kiss-soft exported to STEP file and simulated in Ansys .Table 2 shows the value for root stresses of gears with different addendum modification. Root stresses are determined through kiss-soft and Ansys. Root stress evaluated from kiss-soft for respective gear addendum modification is plotted on graph in Figure 2.

Table .1 Root stress

Addendum modification (mm)	**Root stresses (Mpa) (Kiss-soft)**	**Root stresses (Mpa) (Ansys)**
-0.3	10.63	10.81
-0.1	9.82	9.90
0	9.53	9.63
0.1	8.98	8.70

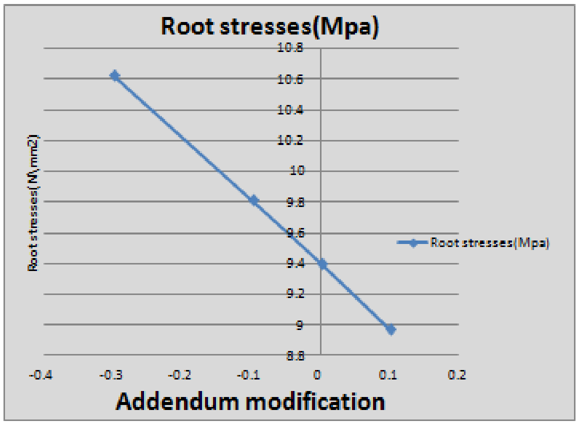

Figure 2 Addendum modification
Vs Root stress

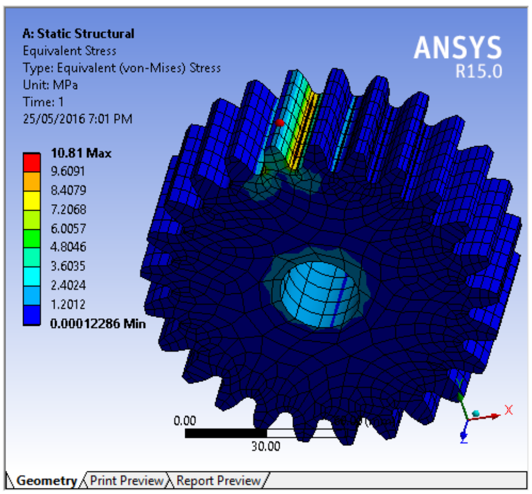

Figure 3 Ansys Result for Addendum modification -0.3

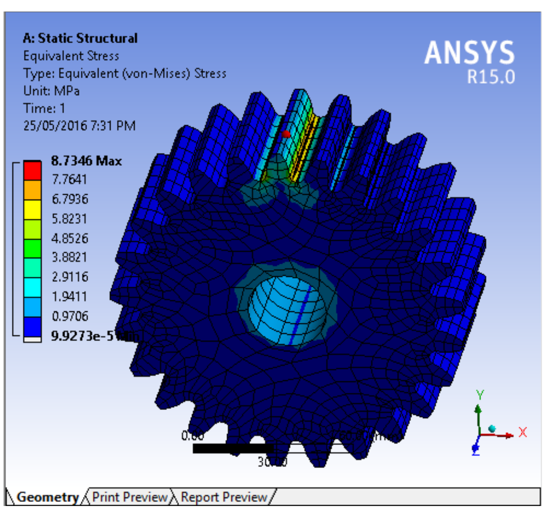

Figure 4 Ansys Result for Addendum modification 0.1

V. CONCLUSION

Root stress depends upon the Addendum modification coefficient can be seen in the graph. The effect of addendum coefficient on root stress when only the driven gear is corrected is found. Then Root stress is also found when both driver gear and the follower gear are modified at the same time. The results have been discussed in detail. It is necessary to typically increase the addendum modification coefficient of the pinion in order to achieve the maximum efficiency and also to increase the root size of the teeth of the pinion.

1. The root stress decreases significantly with an increasing addendum modification coefficient also it decreases with an increase in pressure angle.

2. The tooth thickness at critical section becomes higher with positive addendum modification coefficient. As the tooth thickness increases at critical section the load carrying capacity of gear increases considerably.

3. The root stress decreases further when both the driver and the follower wheel are modified at the same time. Root stress decreases to the lowest value when driver wheel gets positive maximum correction whereas the follower gets the negative correction.

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