Scientific Bulletin of the Petru Maior University of Tirgu Mures
Vol. 6 (XXIII), 2009
ISSN 1841-9267

Computer aided analysis of cylindrical gears geometry with modified profile

Cosmin PETRA
Universitatea PETRU MAIOR din Târgu Mureş
cpetra@upm.ro

Abstract

For the purpose of finding solutions to optimize the classical involute profile, usually use for cylindrical gear, the present study aims as research direction, to modify that profile for improving the overall performance of gears. Gear optimizing will be made using advanced methods of calculation.

1. Introduction

Throughout the world, becoming more gears are manufactured by injection mold plastic, metal powder sintering or by cutting metal. Therefore, restrictions which are increasingly more, can be forgotten since the beginning of the project, when gears are made by traditional methods of cutting and grinding.

Therefore, use computer programs is heady necessary, because bring a clear support in process design, in the decrease time and cost of designing and implementing a project.

Thereby, opens up new possibilities for optimizing gears, endeavoring at a wide range of improvements: reducing noise, increasing of lift gear characteristics. With adequate support came from the calculation and optimization software for gears, this improvements will can be made without any problem. This opens new horizons for improving gears.

In the future will be enhanced following trends (visible even today):

- more metals will be replaced with plastic material
- significant reductions in noise and vibration of gears
- characteristics improvement (design of small gears but with the same or even higher performance than traditional gears)
- the use of specific forms of teeth

2. Addendum optimization

Safety factor against bending strength increases significantly if the connection radius of involute tooth addendum is substantially increased (modified). Even if gear cutting have carefully rounded edges, manufacture of gear using the generation methods by rolling, does not guarantee getting a good tooth fillet radius.

The safety function can be substantially improved by a change of profile, which will be conducted outside the involute. Using software to calculate and determine the stress, improvements can be checked directly.

It is clear that the geometry of the tooth, affecting the strength of the tooth and as a result we need an optimization of the addendum. As a condition it's require increasing tooth resistance. If tooth fillet it's made with non-standard curves, where it is desirable to study addendum stress, not be neglected graphical methods of analysis.

They will be examined three types of fillet:

- fillet with ρ^*_{fP} = 0,38
- fillet with ρ^*_{fP} = 0,45
- fillet optimization (elliptical)

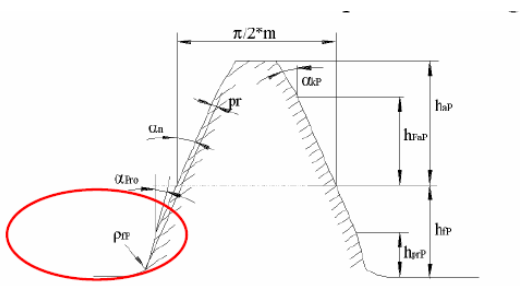


Fig. 1. Reference profile of gear [7]

According to ISO 6336, the critical cross section in the tooth root can be found via the 30° tangent of the root contour. YF and YS are then calculated as shown in formulas (1) and (2) respectively:

Keywords: *gears geometry, analysis, methods of calculation*

$$Y_F = \frac{\dfrac{6 \cdot h_{Fe}}{m_n} \cdot \cos \alpha_{Fen}}{\left(\dfrac{s_{Fn}}{m_n} \right)^2 \cdot \cos \alpha_n} \qquad (1)$$

$$Y_S = (1{,}2 + 0{,}13 \cdot L) \cdot q_S \left[\frac{1}{1{,}21 + \frac{2{,}3}{L}} \right] \qquad (2)$$

The resulting root stress is then calculated according to formula :

$$\sigma_F = \sigma_{FO} \cdot K_A \cdot K_V \cdot K_{F\beta} \cdot K_{F\alpha} \le \sigma_{FP} \quad (3)$$

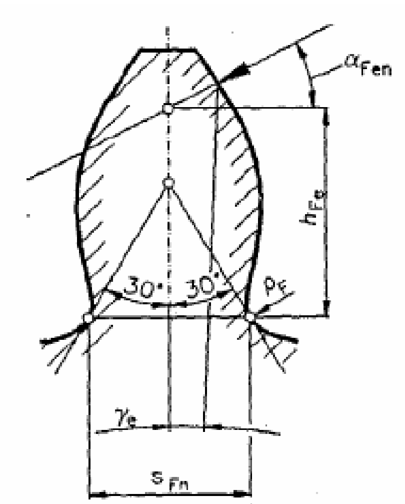
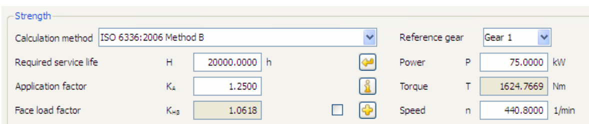


Fig. 2. Calculation of root stress according to ISO6336 [8]

If the tooth base geometry is change, the point defined above may not be the point where tension is highest.

3. Strength rating according to ISO6336

First determine of the safety factor will be in ISO 6336, method B. In this case, a standard reference profile (1.25 / 0.38 / 1.00) according ISO 53.2. Profil A is used.



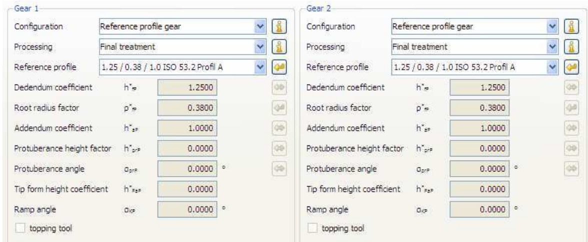


Fig. 3. Initial settings according to ISO 6336, method B

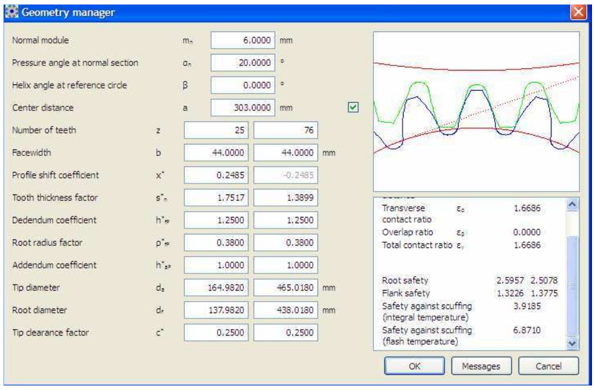


Fig. 4. The value of safety factor, geometric, ρ * $_{fp}$ = 0.38

The next step is setting a different value of the radius of fillet, ρ * fp = 0.45. It will repeat all previous steps, resulting from the performance calculation following report:

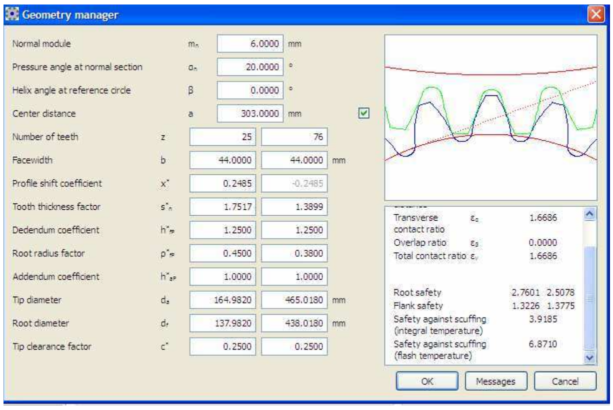


Fig.5. The value of safety factor, geometric, ρ * $_{fp}$ = 0.45

It is noted that the value of safety factor increased with increasing radius of fillet

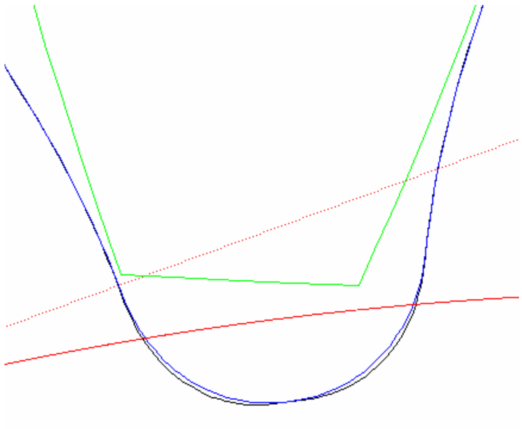


Fig. 6. Superimposed representation of the two profiles

4. Strength rating using graphical method

The possibility to setting the gear is an undeniable advantage when trying to design an optimal gear. Enabling graphical method is performed as follows:

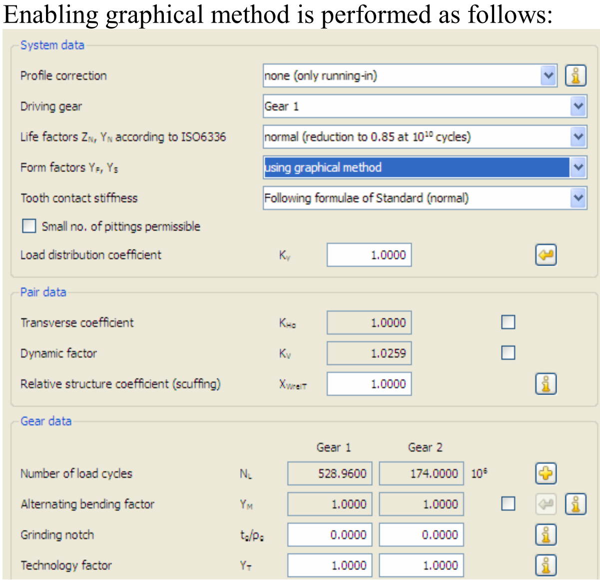


Fig. 7. Enabling graphical method

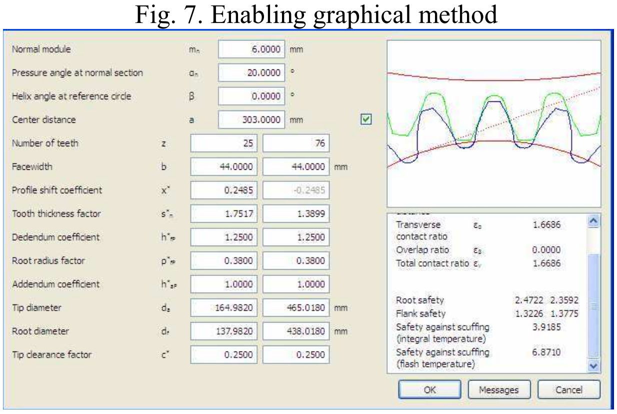


Fig. 8. Safety factor, $\rho^*_{fp} = 0.38$, graphical method

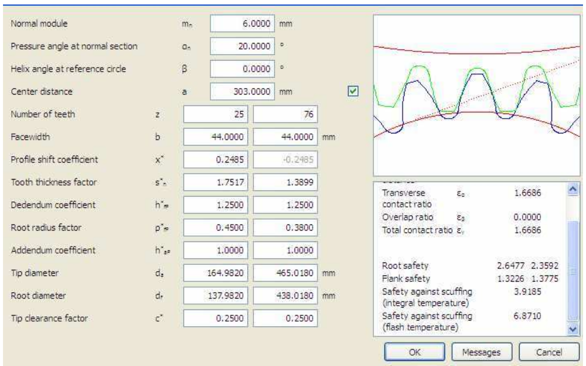


Fig. 9. Safety factor, $\rho^*_{fp} = 0.45$, graphical method

Still trying to improve tooth resistance to requests by involute connection with an arc of ellipse in the base at the tooth.

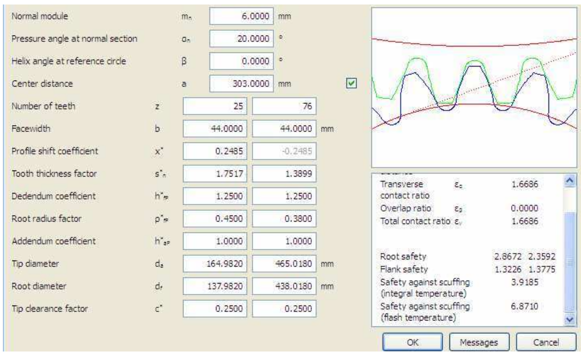


Fig. 10 Optimized fillet

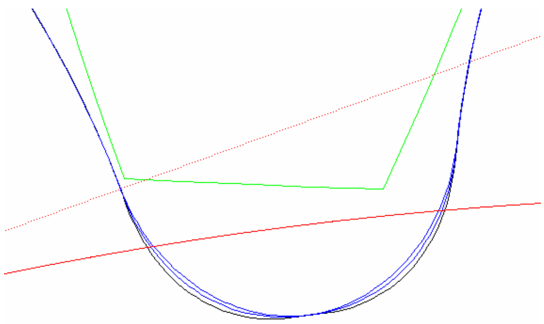


Fig. 11 Superimposed representation of the three profiles

Profile of basic rack that will process the toothed wheel which was amended in the addendum of tooth, there isn't one standard and therefore it's recommended designing such tools only for large production. These things have a financial justification, because the total costs of achieving those rack is reduced.

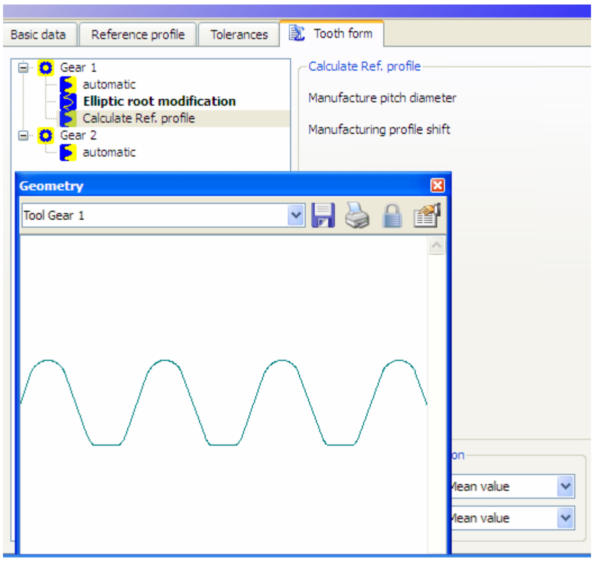


Fig. 12. Basic rack

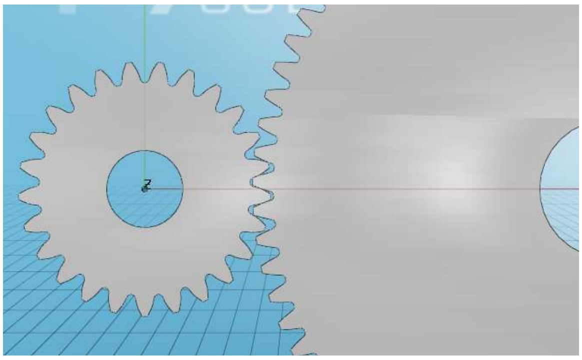


Fig. 13. Optimized gears

5. Conclusion

Tabel 1. Value of factor safety

	Safety factor according ISO 6336	Safety factor according ISO 6336, graphical method
$\rho^*_{fP} = 0{,}38$	2,5957	2,4722
$\rho^*_{fP} = 0{,}45$	2,7601	2,6477
Eliptical	2,7601*	2,8672
Improvement	6%	15%

It is clearly visible that by optimizing the geometry of the tooth, the factor of safety against breakage by bending can be increased by 15%. The only problem would be that, for processing the teeth, it takes a special tool (modified gear hob) so will only yield to large batch production.

6. References

[1] BOLOŞ, V., BOLOŞ, C., Aspecte privind geometria angrenajelor cilindrice cu profilul danturii de tip Concurve. Buletinul Ştiinţific al Universităţii “Petru Maior”, Târgu Mureş, , 2001

[2] PETRA, C., Spur gears with modified tooth profile for providing constant relative curvature – Determining the path of contact , pag. 367- 371, Proceding of the 8-th International Conference MTeM, Cluj-Napoca,4-th - 5-th October 2007, ISBN 973-9087-83-3

[3] PETRA, C., Spur gears with modified tooth profile for providing constant relative curvature – Determining the line of action, Interdisciplinarity in engineering – INTER-ING, Târgu Mureş, noiembrie 2007

[4] PETRA, C., Consideration concerning radius of curvature in contact of mating flanks of spur gears,pag. 93-96, la Proceeding of the 7-th International Conference MteM, Cluj- Napoca 2005, , ISBN 973-9087-83-3

[5] DONIN, R. : Contribuţii la generarea danturii roţilor cu profil ne-evolventic. Teză de doctorat, Universitatea „ Transilvania ” din Braşov, 2001

[6] FLOREA, R. : Contribuţii laproiectarea angrenajului optimal. Teză de doctorat, Universitatea „ Transilvania ” din Braşov, 1997

[7] ISO 6336-1:2006, Calculation of load capacity of spur and helical gears - Part 1: Basic principles, introduction and general influence factors

[8] ISO 6336-3:2006, Calculation of load capacity of spur and helical gears –Calculation of tooth bending strength