

# Computer aided analysis of cylindrical gears geometry with modified profile

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#### 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  $\rho *_{fP} = 0,38$
- fillet with  $\rho_{\text{fP}}^* = 0.45$
- fillet optimization (elliptical)

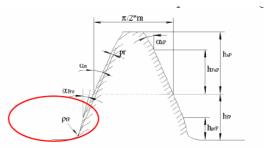


Fig. 1. Reference profile of gear [7]

According to ISO 6336, the critical cross section in the tooth root can be found via the  $30^{\circ}$  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

$$Y_{F} = \frac{\frac{6 \cdot h_{Fe}}{m_{n}} \cdot \cos \alpha_{Fen}}{\left(\frac{s_{Fn}}{m_{n}}\right)^{2} \cdot \cos \alpha_{n}}$$
(1)  
$$Y_{S} = (1, 2 + 0, 13 \cdot L) \cdot q_{S} \left[\frac{1}{1, 21 + \frac{2, 3}{L}}\right]$$
(2)

The resulting root stress is then calculated according to formula :

$$\sigma_{F} = \sigma_{FO} \cdot K_{A} \cdot K_{V} \cdot K_{FB} \cdot K_{F\alpha} \leq \sigma_{FP} \quad (3)$$

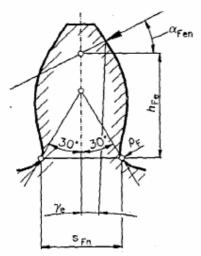


Fig. 2. Calculation of root stress according to ISO6336 [8]

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.

Strength							
Calculation method	ISO 6336:2006 Metho	d B	~	Reference gear		Gear 1 💊	
Required service life	н	20000.0000 h	<b>~</b>	Power	Ρ	75.0000	kW
Application factor	Ka	1.2500	2	Torque	Т	1624.7669	Nm
Face load factor	Кна	1.0618	🗆 😔	Speed	n	440.8000	1/min

Configuration	Reference profile	e gear	~	1	Configuration	Reference profile	gear	~	Ĩ
Processing	Final treatment		~	1	Processing	Final treatment		¥	1
Reference profile	1.25 / 0.38 / 1.0	ISO 53.2 Profi	A 🔽	2	Reference profile	1.25 / 0.38 / 1.0	ISO 53.2 Profi	А 🚩	
Dedendum coefficient	h's	1.2500		00	Dedendum coefficient	h'e	1.2500		00
Root radius factor	p'#	0.3800		44	Root radius factor	o'	0.3800		4
Addendum coefficient	h"	1.0000		40	Addendum coefficient	h"20	1.0000		44
Protuberance height fa	ctor h"##	0.0000		49	Protuberance height fac	tor h <sup>*</sup> pr#	0.0000		04
Protuberance angle	0,00	0.0000	•	40	Protuberance angle	0,07	0.0000	•	00
Tip form height coeffici	ent h"sao	0.0000			Tip form height coefficie	nt h <sup>*</sup> ean	0.0000		
Ramp angle	00	0.0000	•		Ramp angle	0.0	0.0000	•	
topping tool					topping tool				

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

Normal module		ma	6.0000	mm			
Pressure angle at normal section		an i	20.0000	•			
ielix angle at reference circle		β	0.0000	•		A A	A
Center distance		a 30	03.0000	mm	2	NE	M
lumber of teeth	z	2	5	76		PLA-	Part -
acevidth	b	44.000	2	44.0000	mm	×	0
Profile shift coefficient	×	0.248	5	-0.2485			
ooth thickness factor	s*a	1.751	7	1.3899		Transverse c.	1.6686
edendum coefficient	h'e	1.250		1,2500		contact ratio	0.0000
toot radius factor	p*#	0.380	0	0.3800		Overlap ratio z <sub>a</sub> Total contact ratio z <sub>y</sub>	1.6686
ddendum coefficient	h'ar	1.000		1.0000		Flank safety 1.3226	
ip diameter	d,	164.982	0	465.0180			2.5957 2.5078 1.3226 1.3775
Root diameter	d,	137.982		438.0180	mm		3.9185
ip dearance factor	c*	0.250	0	0.2500		Safety against scuffing (flash temperature)	6.8710

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

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

Normal module	ſ	n. 6.00	000 mm		
Pressure angle at normal section		in 20.00	• 000		
Helix angle at reference drde	f	0.00	000		000
Center distance		303.00	000 mm		NDA
Number of teeth	z	25	76	]	
Facewidth	b	44.0000	44.0000	mm	X
Profile shift coefficient	x*	0.2485	-0.2485	]	
Tooth thickness factor	s'.	1.7517	1.3899	]	Transverse s. 1.6686
Dedendum coefficient	h's	1.2500	1,2500	]	contact ratio
Root radius factor	ρ*=	0.4500	0.3800	1	Overlap ratio ε₂ 0.0000   Total contact ratio εν 1.6686
Addendum coefficient	h'ar	1.0000	1,0000	1	
Tip diameter	d,	164,9820	465.0180	mm	Root safety 2.7601 2.5078 Flank safety 1.3226 1.3775
Root diameter	d,	137.9820	438.0180	mm	Safety against scuffing 3.9185 (integral temperature)
	e*	0.2500	0.2500	1	Safety against scuffing 6.8710 (flash temperature)

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

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

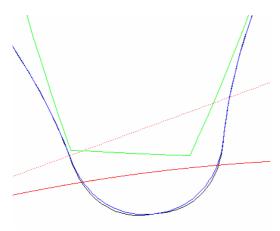


Fig. 6. Superimposed representation of the two profiles

### 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:

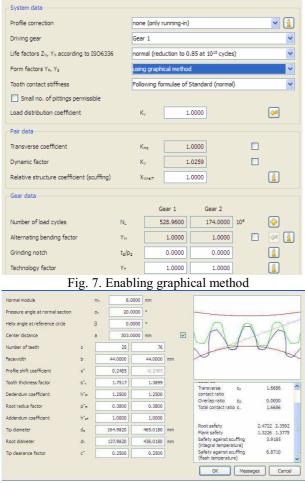


Fig. 8. Safety factor,  $\rho^*_{fp} = 0.38$ , graphical method



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.

Normal module		m. 6.00	000 mm				
Pressure angle at normal section		on 20.00	° 000				-
Helix angle at reference circle		β 0.00	° 000		O F	A	esi
Center distance		a 303.00	000 mm		NL	M	
Number of teeth	z	25	76		- D	-U	
Facewidth	Ь	44.0000	44.0000	mm	×	C	×
Profile shift coefficient	x*	0.2485	-0.2485				
Tooth thickness factor	5°n	1.7517	1.3899		Transverse E-	1.6686	
Dedendum coefficient	h"ø	1.2500	1.2500		contact ratio		
	h"e p"e	0.4500	1.2500 0.3800		contact ratio Overlap ratio ε <sub>2</sub> Total contact ratio ε <sub>2</sub>	0.0000	
Dedendum coefficient Root radius factor Addendum coefficient					Overlap ratio ε <sub>2</sub> Total contact ratio ε <sub>ν</sub>	1.6686	-
Root radius factor	p*ø	0.4500	0.3800		Overlap ratio 80		-
Root radius factor Addendum coefficient	p*ø h*₂ø	0.4500	0.3800	] ] mm	Overlap ratio 8ª Total contact ratio 8, Root safety	1.6686 2.8672 2.3592	

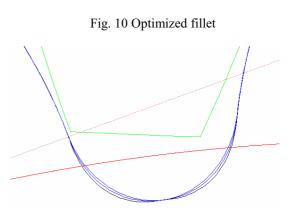


Fig. 11 Superimposed representation of the three profiles

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.

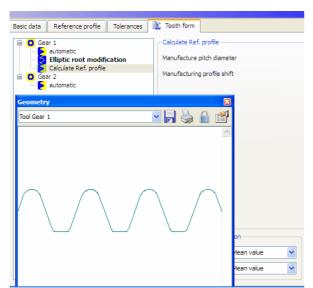


Fig. 12. Basic rack

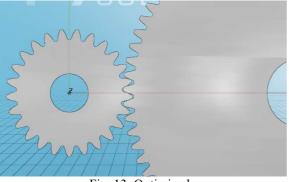


Fig. 13. Optimized gears

### 5. Conclusion

Tabel 1. Value of factor safety

	Safety factor	Safety factor
	according	according ISO
	ISO 6336	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%
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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

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