HCR Gears in the Industrial Gearbox

M. Burián, M. Trochta, and J. Havlik

Abstract Gearing with a prolonged contact ratio – so called HCR gearing is used more and more in these days, mainly for the gearboxes in the automobile industry, and for other transport machines. HCR gearing mainly influences the noise reduction in the gearing system. Relative addendum of the HCR gearing usually changes the height of the gear h_a^*, however, this value must be higher than $h_a^* > 1$.

Keywords HCR • Standard • Gearing • Industrial • Gearbox

1 Comparison of a Standard and HCR Gearing

In the Figs. 1 and 2, you can compare a standard and HCR gearing.

Advantages of HCR gearing with contact ratio coefficient $\varepsilon_\alpha \geq 2$

- reduction of gear load because they occupy two and more teeth
- bigger carrying capacity in comparison with standard gearing
- noise reduction due to larger coefficient of gear profile
- there are not spontaneous changes of stiffness as in the usage of $\varepsilon_\alpha < 2$

Disadvantages of HCR gearing

- application and production, special tools for gear production
- smaller possibility of corrections, larger amount of teeth
- higher values of slidings courses [1, 2]

2 Selection of HCR Gearing in the Industrial Gearbox MTC 42 A – 210

We had a discussion with TOS Znojmo company's designers if there should be a financial investment made for the HCR gearing production of the existing front industrial gearbox MTC 42 A – 210, see Fig. 3 [3].

M. Burián (✉) • M. Trochta • J. Havlik
VŠB – Technical University of Ostrava, Ostrava, Czech Republic
e-mail: miroslav.burian@vsb.cz; miroslav.trochta@vsb.cz; jiri.havlik@vsb.cz

V. Dynybyl et al. (eds.), *The Latest Methods of Construction Design*,
DOI 10.1007/978-3-319-22762-7_2

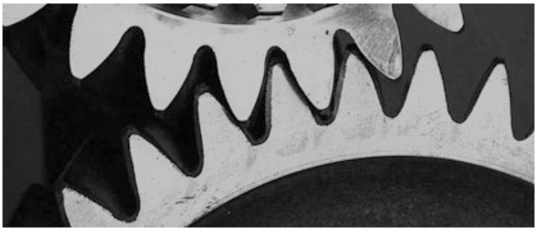

Fig. 1 Gearing with HCR profile

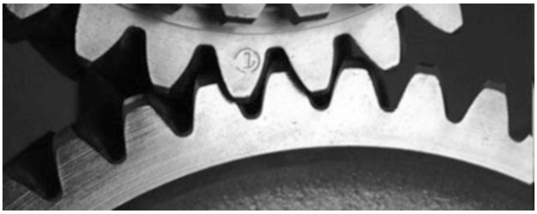

Fig. 2 Gearing with a standard gearing sprocket

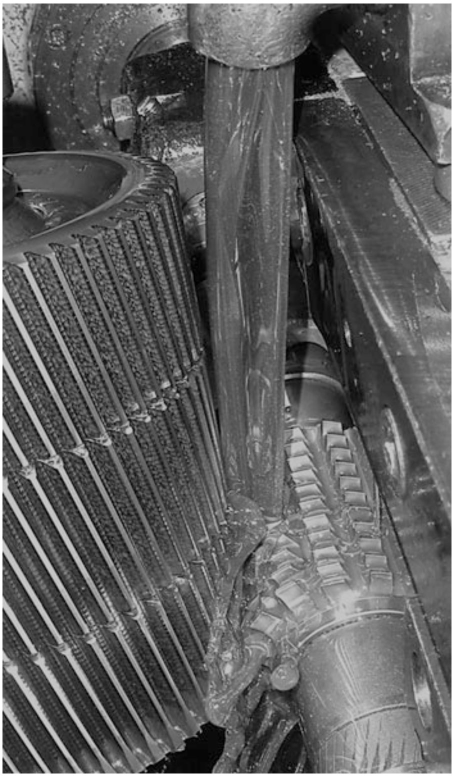

Fig. 3 Front industrial gearbox MTC 42 A – 210

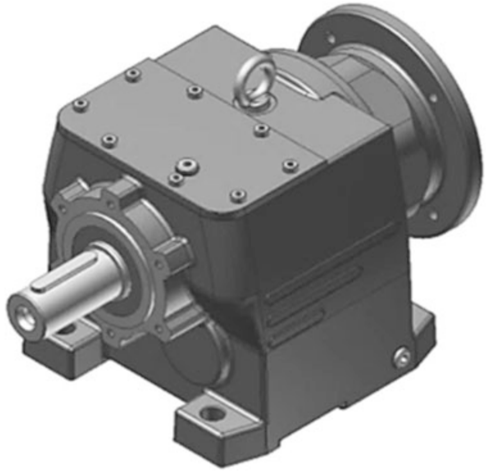

Fig. 4 Gearing production in TOS Znojmo

Table 1 Parameters of the front industrial gearbox MTC 42 A – 210

Number of teeth		Module [mm]	Angle of sprocket's inclination	Wheel's width [mm]		Cog wheel's material		Axial distance [mm]
z_1	z_2	$m_{N1.2}$	$\beta_{1.2}$	1	2	1	2	$a_{W1.2}$
23	91	1	20°	24	18	16 220	14 220	61

Company TOS Znojmo produces all cog wheels itself, see Fig. 4. Therefore if to invest funds to HCR gearing production.

Industrial gearbox is the most produced article in the company. There are no tools for non-standard HCR gearing production. The original size of gearbox, material for cog wheels, and the same gear ratio must remain. The original module, teeth number, and wheel's depth were kept.

3 Strength Calculation for Bend and Contact

I made a strength calculation of gearing for the appropriate bend and contact. Only first gearing is introduced here z_1, z_2. Values for calculation are in the Table 1.

It is impossible to determine which parameters are the best for the calculation while choosing parameters for HCR gearing. It depends on the concrete usage of industrial gearbox, peripheral speed, carrying capacity of wheels and technological possibilities of gearing production.

By the calculation of HCR gearing for gearing 1,2, I changed the relative addendum by both, the teeth of the pinion, and the cog wheel from value 1,02 to value 1,20. It is possible to change relative head clearance c_p^*, relative radius ρ_{fP}^* and mesh angle α. Alteration possibilities are in Table 2. First column is for relative

Table 2 Gear geometry of gearing

		Producer	Calculation of mesh angle α = 20°			Calculation of mesh angle α ≠ 20°			
		I.	II.	III.	IV.	V.	VI.	VII.	VIII.
Input value of the gearing 1,2	h^*_{a1}	1.00	1.05	1.10	1.15	1.20	1.10	1.10	1.20
	h^*_{a2}	1.00	1.05	1.10	1.15	1.20	1.10	1.10	1.20
	ρ^*_{fP1}	0.38	0.38	0.38	0.38	0.35	0.30	0.30	0.30
	ρ^*_{fP2}	0.38	0.38	0.38	0.38	0.35	0.30	0.30	0.30
	α	20°	20°	20°	20°	16°	18.5°	16.5°	16.5°
Calculated parameters of the gearing 1,2	ε_α	1.47	1.54	1.60	1.66	1.88	1.65	1.73	1.86
	ε_β	1.95	1.95	1.95	1.95	1.95	1.95	1.95	1.90
	ε_γ	3.43	3.49	3.56	3.62	3.84	3.61	3.69	3.82
	x_1	0.331	0.355	0.379	0.405	0.549	0.415	0.470	0.533
	x_2	0.017	−0.007	−0.031	−0.057	−0.198	−0.066	−0.119	−0.181

Table 3 Gear strength verification of gearings 1 and 2

		Producer	Calculated values						
		I.	II.	III.	IV.	V.	VI.	VII.	VIII.
Strength calculation of gearing $z_{1,2}$ *for bending*	S_{F1}	3.14	3.17	3.20	3.18	3.20	3.14	3.12	3.17
	$\sigma_{F\ MAX1}$	446	442	437	441	438	446	449	441
	$\sigma_{FP\ MAX1}$	1400	1400	1400	1400	1400	1400	1400	1400
	$\sigma_{F\ MAX2}$	502	481	477	473	512	492	511	502
Strength calculation of gearing $z_{1,2}$ *for contact*	S_{H1}	1.39	1.42	1.44	1.47	1.44	1.43	1.40	1.39
	$\sigma_{H\ MAX1}$	1.296	1.269	1.244	1.220	1.247	1.259	1.284	1.296
	S_{H2}	1.39	1.42	1.44	1.47	1.44	1.43	1.40	1.39
	$\sigma_{H\ MAX2}$	1 296	1 269	1 244	1 220	1 247	1 259	1 284	1 296

addendum tooth $h_a^* = 1$ of standard gearing profile – value of created industrial gearbox MTC 42A – 210. Calculation was done by using software “Geometrie”.

By the calculation, I found out the timing coefficient of the gear profile ε_α, the timing coefficient gear profile ε_β and the total coefficient of time for the contact ratio $\varepsilon_\gamma = \varepsilon_\alpha + \varepsilon_\beta$. In Table 2 are the correction values of pinion x_1 and the wheel x_2 [4].

From the table, it is visible that the contact ratio of mesh angle $\alpha = 20°$ has increased the gear profile ε_α in comparison with the original value in the first column (standard gearing). Duration of the mesh profile has increased $\varepsilon_\gamma = 3,62$. The highest value of the contact ratio $\varepsilon_\gamma = 3,82$ is for the last column VIII. In addition to the relative teethes addendum h_a^*, I also had to change the mesh angle α and relative radius of the gear heel ρ_{fP}^*. Requested axial distance $a_{w1,2} = 61\,\text{mm}$ showed in Table 1 must be kept. Calculated axial distance doesn’t match the requested one. The difference is evened up by corrections of stabilization of the specific slips.

3.1 Simplified Strength Verification

I made a simplified strength verification of gearing based on ČSN 01 4686 – part 4 with the help of software CSNw. Results of strength verification are in Table 3 [5].

Indexes 1 are for the pinion and indexes 2 for the wheel. Particular quantities (parameters) are:

S_F – safety factor versus formation of fatigue failure in the tooth heel,
$\sigma_{F\ MAX}$ – bending stress of tooth heel in the dangerous section,
$\sigma_{FP\ MAX}$ – allowed stress during the bending = 1400 MPa,
S_{H1} – safety factor versus formation of fatigue failure on the side tooth,
$\sigma_{H\ MAX}$ – touch stress (Herz pressure) in the rolling point,
$\sigma_{HP\ MAX}$ – allowed stress during the contact = 2400 MPa.

Conditions for strength verification according to the formula (1), (2), (3) and (4):

$$\sigma_{FP\ \mathrm{MAX1}} \geq \sigma_{F\ \mathrm{MAX1}} \tag{1}$$

$$\sigma_{FP\ \mathrm{MAX2}} \geq \sigma_{F\ \mathrm{MAX2}} \tag{2}$$

$$\sigma_{HP\ \mathrm{MAX2}} \geq \sigma_{F\ \mathrm{MAX1}} \tag{3}$$

$$\sigma_{HP\ \mathrm{MAX2}} \geq \sigma_{F\ \mathrm{MAX2}} \tag{4}$$

Particular safeties, bending stress and stress in the contact haven't changed much. All conditions were fulfilled according to the formulas (1), (2), (3) and (4). Allowed stress is always higher than applied stress. There is no sense to use nonstandard HCR hearing for small modules. The Contact timing ratio of teeth profile hasn't changed much compared to original value of the standard gearing. Carrying capacity of cog wheels hasn't changed much in comparison with standard cog wheels of standard tooth profile. It is too expensive to use these teeth for this industrial gearbox. It is necessary to purchase a special tool for the production of the gearing, which the TOS company doesn't have available. Existing gearing is satisfactory.

4 Gearing from Modules 1 up to 9 mm HCR

Further, I devoted my time to HCR gearing usage during module changes. How big stress in the bending or contact will then influence (with the change of the module) imaginary axis distance? Apart from module and tooth number, the values are the same as in Table 1. I changed only relative addendum. The other parameters are the same as in the standard gearing.

4.1 Comparison of the Standard and HCR Gearing

In Fig. 5 is the comparison of bending stress in perilous (hazardous) place of pinion tooth of both standard and HCR gearing.

In the Fig. 5 is indicated the stress increase and decrease. And that at all the time by the module 2,75 mm.

Because during the calculation of bending stress in the perilous profile of the tooth heel in the pinion $\sigma_{F\ MAX1}$ and in the wheel $\sigma_{F\ MAX2}$, the coefficient slope sprocket is changing Y_{β}. Based on the Fig. 6 this coefficient is dependent on slope sprocket $\beta_{1,2} = 20°$ and on the contact ration of the profile ε_{β}.

Up to the module 2.5 mm, the contact coefficient ratio of the profile ε_{β} is bigger than "1". Therefore the coefficient of sprocket's angle Y_{β} is constant up to this value, see Fig. 6. From the module 2,75 mm, the contact ratio of the profile ε_{β} is smaller then "1". Therefore there is a change of the coefficient slope sprocket Y_{β}, which influences bending stress in the dangerous profile of tooth heel in the pinion $\sigma_{F\ \mathrm{MAX1}}$ and on the wheel $\sigma_{F\ \mathrm{MAX2}}$.

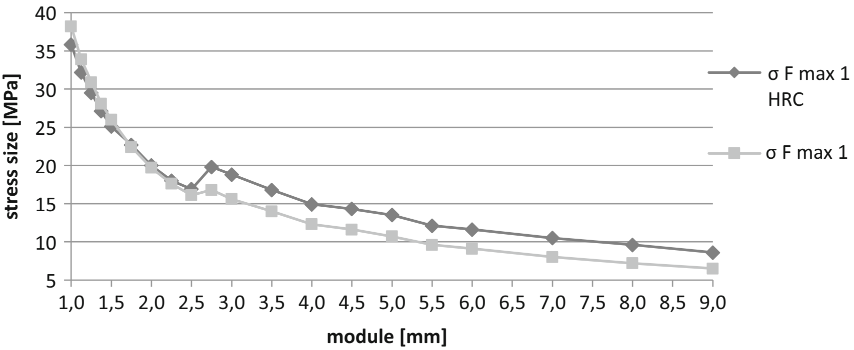

Fig. 5 Bending stress in the dangerous profile of tooth heel in the pinion

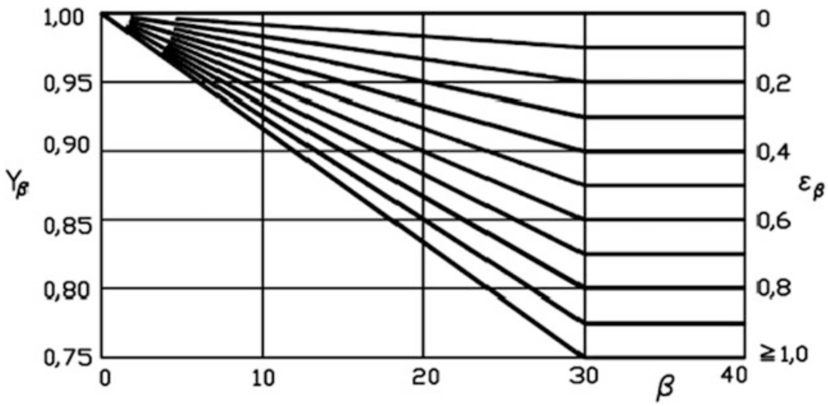

Fig. 6 Coefficient of sprocket's angle Y_{β}

5 Conclusion

Usage of HCR gearing is beneficial. Larger safety factor versus creation of fatigue failure in the tooth heel in the pinion and on the wheel. And to decrease of safety factor versus creation of fatigue failure of the side teeth on the wheel and in the pinion. With increasing module, there is a decrease of bending stress in the perilous profile of the tooth heel in the wheel and on the pinion. The disadvantage for the the gear production is the production of a non-standard tool. There is no sense to use non-standard HCR gearing for seller modules. It is possible that the company TOS Znojmo will not produce gearing with a small module. Depending on concrete usage, larger modules may be used.

Acknowledgments Contribution has been done in connection with project "Posouzení vlastností ozubených převodů z hlediska jejich geometrie a přesnosti výroby", REG. NO. SP2014/24

References

1. M. Němček, *Vybrané problémy geometrie čelních ozubených kol*. Stříbrná technická řada (Montanex, Ostrava, 2003), p. 143 s. ISBN 80-722-5111-2
2. V. Moravec, *Konstrukce strojů a zařízení II: čelní ozubená kola, teorie, výpočet, konstukce, výroba, kontrola* (Montanex, Ostrava, 2001), p. 291 s. ISBN 80-722-5051-5
3. Čelní převodovka MTC 42 A – 210, *Čelní převodovky* [online] (2014), Available via: http://www.tos-znojmo.cz/produkce/mtc/cz/index.htm. Cited 19 Mar 2014
4. GEOMETRIE. *Kontrola geometrie ozubených kol, verze 4.0* © M. NĚMČEK 2013
5. CSNw. *Pevnostní výpočet čelích ozubených kol, verze 3.0*. © M. NĚMČEK 2011