Layout of profile modifications for symmetric and asymmetric plastic gears

Dr. **A. Pogačnik**, Bauhar s.p., Bled, Slovenia; Dr. **U. Kissling**, KISSsoft AG, Bubikon, Switzerland

This paper was first presented at International Conference on Gears 2019, 3rd International Conference on High Performance Plastic Gears 2019, 3rd International Conference on Gear Production 2019, Garching/Munich (VDI-Berichte 2355, 2019, VDI Verlag GmbH, Page 1461-1473).

Abstract

The behaviour of gears can be significantly improved with appropriate profile modifications. A modification of the gear profile will change the load distribution during a meshing cycle, therefore changing contact pressure, wear, power loss and transmission error. In case of applied tip and/or root relief, the so-called contact shock at the beginning and end of the meshing can be removed, reducing substantially the vibration and noise behaviour of the gear mesh.

The main task of profile modifications is to compensate for the errors in the teeth meshing due to the tooth bending. As the bending of metal gears is very small, a general rule is that the gear manufacturing quality must be high (quality 6 or better according to ISO 1328 [12]), if profile modifications are to be applied. Otherwise the manufacturing tolerances are bigger than the modification amount, therefore practically annihilating the effect of a precise dimensioned modification. Plastic gears produced in injection moulds have usually a quality between 9 - 11, so a conclusion could be, that profile modifications are not useful in this case. But the elasticity modulus of plastic materials is approximatively 100 times lower compared to steel, resulting in much higher bending of the tooth. Therefore, for plastic gears, the use of profile modifications is effective also for lower manufacturing quality.

The layout of profile modifications must be verified by a loaded tooth contact analysis (LTCA), which allows the users to analyse the contact during a meshing cycle step by step. To find the best variant, a parameter varying technique should be used.

If the load is unidirectional most of the time, then asymmetric gears can be considered. The primarily loaded flank can be optimized - normally by increasing the pressure angle - to reduce contact stress and wear. For gears produced in injection moulds, the production cost of an asymmetric gears is the same as for a symmetric gear.

Some examples of gears with symmetric and asymmetric teeth are discussed, where – due to a well applied profile modification - the lifetime could be significantly increased, and/or the noise excitation reduced.

1. Introduction

From automotive industry, to medical equipment, from space industry to simple toys, plastic gears are found in a variety of different areas. The reason is that plastic gears are very suitable for mass production, but also offer other advantages, like lower masses, good dampening properties, lubrication free operation and competitive prices [1,2]. On the other hand, most plastic materials used for gears have moderate allowable operating temperatures, inferior root and flank fatigue strength [1]. Some of these drawbacks can be eliminated by selecting a better plastic material, but usually this results in much higher price and/or more complex manufacturing processes.

With the guideline VDI 2736 [3], the strength (both root and flank) of the plastic gears can be calculated, when the fatigue data is available [4]. But more often, strength is not the main design criteria anymore. NVH (noise-vibrations-harshness) is becoming very important, especially for gears running in sensitive environments, like measuring equipment but also in car actuators, where passenger comfort is a priority. Wear is also very important, since it can induce unwanted noise and vibrations, but also produces wear debris, which is not allowed in some applications (like medical or food).

There are several factors that influence the NVH behaviour in a gear reducer. Most important factors are the motor, the transmission elements and the housing. In this paper, only transmission optimization is discussed.

NVH can be improved in different ways; for instance, selecting an appropriate material combination, where the gear is manufactured in a softer plastic material or increasing the quality of the gears [5]. However, this can increase the complexity and price of the gearbox. The easiest way to influence NVH and wear behaviour is through macrogeometry of the gears (normal module, helix angle, reference profile, profile shift, …) and with appropriate microgeometry design (profile modifications) [6,7]. Since plastic gears are usually manufactured using injection moulding, optimized macro and micro geometry has no influence on the price of the gears.

The main sources of noise in a gear pair are mainly tooth stiffness variations, geometrical errors, surface sliding and tooth meshing impact due to bending [7]. This can be improved by applying profile modifications. The most typical profile modifications for smooth meshing of plastic gears are long tip and root reliefs (linear, progressive or arc like) and profile crowning. Linear reliefs should be avoided, since there is no smooth transition between the involute and the applied tip relief. The effective noise generated by plastic gears (in dB(A)) cannot be easily calculated. When optimizing gears in terms of noise, essentially the same technique as for steel gears is used - the transmission error must be reduced, and the contact shock must be avoided.

Lead modifications can also have a positive influence on NVH, especially when trying to compensate for axis inclination and deviation errors. Lead crowning could be applied in such cases, but the problem is that such modifications are not easily injection moulded, so it is very seldomly used on plastic gears. Lead modifications are not evaluated in this study.

The main idea of profile modifications is to compensate for the errors in the teeth meshing due to the tooth bending and due to manufacturing errors. A general rule for steel gears is that profile modifications should only be applied, when the quality of the gears is high (Q6 or better according to the ISO 1328). The reason is that profile modifications on steel gears are usually relatively small, so they are annulated at lower quality by the manufacturing errors and have no or very small effect.

So why should profile modifications be applied on plastic gears, when the manufacturing quality is much worse compared to steel gears (maybe Q9-11)? The reason is that for plastic gears, the bending of the teeth is much higher compared to steel gears due to lower Young's modulus. Therefore, for plastic gears, the use of profile modifications is effective also for lower manufacturing quality. Table 1 shows the bending of the teeth compared to the manufacturing errors. It can be seen very clearly that bending of the tooth compared to the tolerance range is similar for plastic gears in quality 12 and case carburised gears in quality 6! Thus, applying profile modifications for plastic gears also make sense.

Table 1: Gear tip bending vs. the tolerance for a gear pair (m_n = 1.375 mm, z = 23/85).

	POM-POM gear (80°C)			CH Steel Gear
Torque (Pinion), Nm	3.3	3.3	3.3	37.2
Bending safety S_F (Pinion)	1.35	1.35	1.35	1.35
Calculation method	VDI 2736	VDI 2736	VDI 2736	ISO 6336-3
Young modulus, MPa	2080	2080	2080	206000
Quality acc. DIN 3961	**10**	**11**	**12**	**6**
Tooth tip bending C_a (by LTCA), µm	**101.5**	**101.5**	**101.5**	**9.1**
Profile slope deviation $f_{H\alpha}$, µm	23	37	59	5
Bending/Slope deviation ($C_a/f_{H\alpha}$)	101/23 = **4.4**	101/37 = **2.7**	101/59 = **1.7**	9.1/5 = **1.8**

The layout of profile modifications should be verified by a loaded tooth contact analysis (LTCA), which allows the users to analyse the contact during a meshing cycle step by step. LTCA can be performed using a finite element approach (FEM) or with semi-analytical method (usually based on the Weber-Banaschek approach [8]). In the gear calculation software KISSsoft [6], the Weber-Banaschek model is used. The procedure is adapted for low Young modulus, so it can be used also on plastic gears, where higher tooth deflections are expected. Additionally, the LTCA is adapted for asymmetric gears according Langheinrich [9].

In this paper, the effect of profile modifications on gear performance will be evaluated through loaded tooth contact analysis using symmetric as well as asymmetric gear geometry. The results show that with an appropriately defined profile modification, the transmission error (noise excitation) and wear can be significantly reduced.

2. Design criteria for plastic gears

Design criteria for plastic gears is usually driven by the product requirements. The most obvious and often used criteria are root and flank safety factors, which can be calculated using the VDI 2736 guideline. The strength calculation in VDI 2736 doesn't consider the tooth elasticity and is thus neglecting the effect of the increased transverse contact ratio under load – a behaviour very common for plastic gears. Figure 1 shows the theoretical path of contact (left) and the path of contact under load (right). For involute gears, the theoretical path of contact is a straight line (theoretical transverse contact ratio of 1.587), limited by the active tip diameters of both gears. Under load however, the theoretical path of contact is extended at the beginning and at the end of meshing, effectively increasing the contact ratio to 2.073. This leads to improper meshing at the beginning and end, since there is flank contact outside of the theoretical path of contact. This is generating the contact shock, which has an influence on the NVH behaviour of the gears.

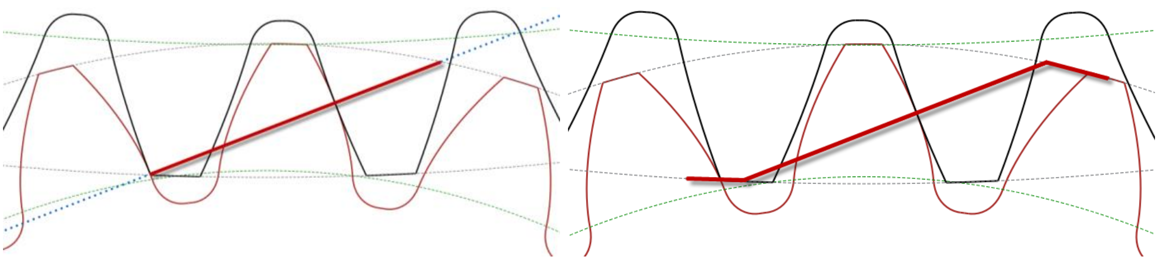

Figure 1: Theoretical path of contact (left) and effective path of contact under load (right).

When gears are optimized for noise behaviour, a good technique is to aim at a high transverse contact ratio ε_α (if possible $\varepsilon_\alpha > 2.0$). Higher contact ratios can be achieved by using a deep tooth form. The problem is, that it is not possible to reach $\varepsilon_\alpha > 2$, if the pinion has a small number of teeth ($z < 22$) because of pointed tip or undercut.

NVH behaviour can also be influenced also by the quality of the gears. The better the quality, the lower the noise. But improving the quality of the plastic gears is usually very expensive (if even possible), so it is not a very convenient way.

NVH behaviour also depends on wear. With progressing wear on the tooth flanks during lifetime, the transmission error will be affected – and normally increase. When optimizing gears in terms of NVH, wear behaviour of the gears should also be checked, especially for dry running gears [11].

It is important to select tribologically appropriate material combinations. However, wear can be reduced also by optimizing macro geometry (specific sliding, …) and/or by applying profile modifications (see Chapters 3 and 4). Even if no profile modifications are applied, it might happen, that “modifications” are generated due to wear during the running in process.

Table 2 shows a comparison of results of VDI 2736 calculation and loaded tooth contact analysis (LTCA) calculation (once using infinite tooth stiffness and once Weber-Banaschek approach). Infinite stiffness assumes, that no tooth bending occurs.

With infinite stiffness, the results are very similar for VDI 2736 and LTCA. This is expected, since also VDI 2736 assumes infinite stiffness for strength calculation. But when considering also the bending of the teeth, the results change significantly. First, the contact ratio increases by 33%, which essentially decreases the normal load on the flanks. This results in significantly lower root stresses (more than 30%) and lower Hertzian pressure (around 20%). It means that the calculation according VDI 2736 can sometimes be too pessimistic. On the other hand, considering the bending of the teeth also results in an increased transmission error (9.31 µm).

Table 2: Comparison of different calculation methods

	VDI 2736	LTCA	LTCA
Stiffness calculation	Infinite	Infinite	Weber Banaschek
Root stress 1, MPa	16.3	15.1	11.3
Root stress 2, MPa	16.1	16.4	9.8
Hertzian pressure, MPa	22.7	24.2	19.2
Contact ratio	1.587	1.582	2.110
Transmission error, µm	0	0	9.31

For the NVH behaviour, the most important parameter in Table 2 is the transmission error. When optimizing in terms of noise, the max-min value of the transmission error (PPTE) should be minimized. PPTE can be reduced by applying appropriate profile modifications.

3. Effects of profile modifications for symmetric gears

For a steel/plastic gear combination, the profile modifications are usually only applied on the plastic gear. If profile modifications are applied also on a steel gear, a custom tool is necessary for manufacturing, therefore costs are increased. However, when using a sinter/plastic combination, modifications can easily (and should) be applied on both gears simultaneously, since the manufacturing costs are the same.

Table 3 shows results from the LTCA (using Weber-Banaschek approach) for different profile modifications applied to gear 1 and 2 (m_n = 1, z = 15/48, α = 20°, ε_{α_th} = 1.31). The material combination is sinter/POM at 50°C. Three different arc tip reliefs (arc) were applied either to only one gear or to both gears simultaneously.

Table 3: LTCA results for with different modifications applied – for symmetric gears.

Solution nr.	1	2	3	4	5	6	7	8	9	10	**11**
Tip relief gear 1 Value, µm Length factor	/ / /	/ / /	/ / /	/ / /	Arc 10 1	Arc 20 1	Arc 30 1	Arc 10 1	Arc 20 1	Arc 30 1	**Arc 23 2**
Tip relief gear 2 Value, µm Length factor	/ / /	Arc 10 1	Arc 20 1	Arc 30 1	/ / /	/ / /	/ / /	Arc 10 1	Arc 20 1	Arc 30 1	**Arc 23 2**
Root stress 1, MPa	43.5	43.9	44.4	44.9	44.2	44.2	44.1	43.9	44.4	44.9	**44.3**
Root stress 2, MPa	43.1	43.1	43.1	43.1	43.4	46.6	46.7	43.4	46.5	46.7	**45.5**
Hertzian pr., MPa	83.4	83.4	83.4	83.4	83.4	83.7	86.5	83.4	83.7	86.4	**90.4**
Contact ratio ε_{α_eff}	1.75	1.73	1.72	1.70	1.73	1.72	1.70	1.71	1.68	1.65	**1.67**
$\varepsilon_{\alpha\_eff} - \varepsilon_{\alpha\_th}$	0.44	0.42	0.41	0.39	0.42	0.41	0.39	0.40	0.37	0.34	**0.36**
PPTE, µm	27.0	27.0	27.0	27.0	23.6	23.3	26.5	23.5	22.8	25.3	**16.8**
Max wear, µm	60.8	60.5	60.1	61.2	57.7	52.6	40.7	55.9	49.6	44.2	**46.0**

The results from Table 3 show, that applying tip modifications only on gear 2 (solutions 2-4) produces no improvement in terms of transmission error and wear behaviour. However, if tip modifications are applied only on gear 1 (solutions 5-7), an improvement in wear behaviour can be seen (reduction by 35%). Transmission error is not affected. When applying modifications to both gears (solutions 8-10), an improvement both in transmission error (15%) and in wear (20%) can be seen. In all cases, applying profile modifications decreased the effective contact ratio ε_{α_eff}. If ε_{α_eff} is bigger than the theoretical contact ratio

ε_{α_th}, a contact shock occurs during the start of the contact, generating noise. Therefore, a reduction of the difference ε_{α_eff} - ε_{α_th} indicates also an improvement of the NVH behaviour.

Solutions 1-10 were obtained by just guessing the values for the modifications. However, with a modification sizing tool in KISSsoft [6] it is possible to determine the optimal modification, as shown in Figure 2. With this tool, modification parameters are automatically varied in a defined range to find the best possible combination for a given gear set and operating conditions. In this case, solution 11 (in *Table 3*) was obtained by modification sizing. From all solutions, no. 11 shows the minimum transmission error (reduction by 40% to unmodified tooth) and low wear (reduction by 25%). Other important parameters must always be checked to avoid unwanted problems. In this case, the root stresses did not change significantly, while the Hertzian pressure increased by less than 10%.

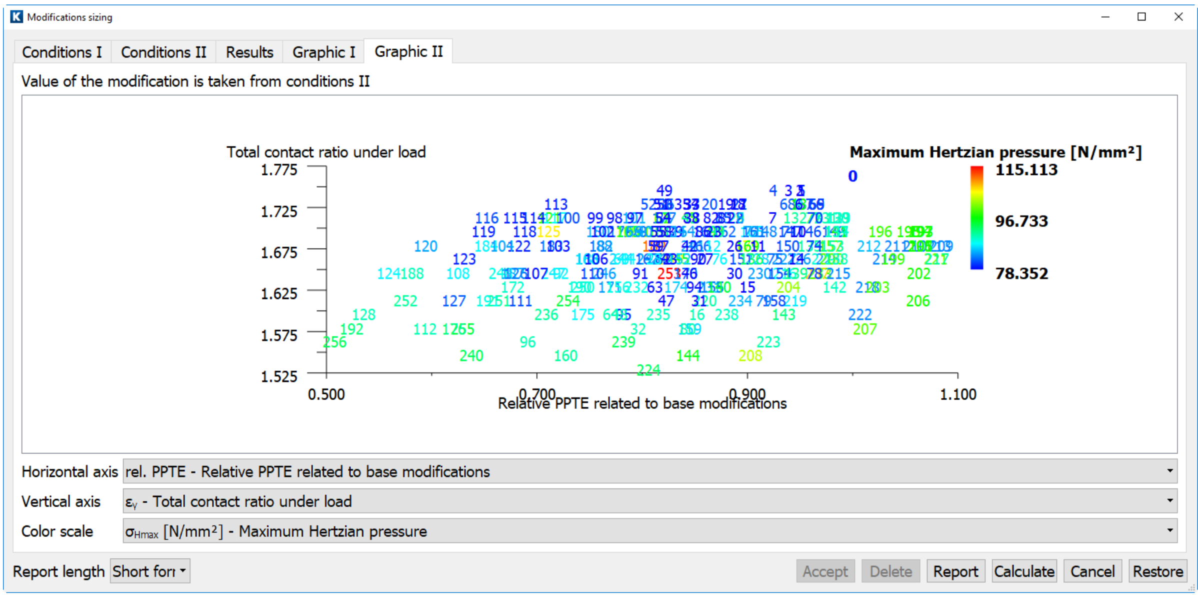

Figure 2: LTCA results from modification sizing functionality in KISSsoft.

Profile modifications for plastic gears can only be fully optimized for a specific load and temperature condition. This means that if the temperature and torque level will change, also the ideal profile modification will change! Thus, it is very important to perform LTCA at different load and temperature conditions and then applying profile modifications, that fits best to all operating conditions.

When optimizing gears in terms of NVH and wear, it is very important not to neglect root and flank strength. Since the effective contact ratio under load is reduced when applying profile modifications, the root and flank stresses may increase. As in Table 3, the increase in root stress in this case is very small (a few %). The flank stress increase is below 10%.

4. Effects of profile modifications for asymmetric gears

Procedure for applying modifications on asymmetric gears is the same as for symmetric gears. For asymmetric gears, the stiffness approach according Weber-Banaschek in the LTCA procedure must be adapted (Langheinrich [9]). Langheinrich approach is partly analytical (modifying the root stress calculation from ISO 6336) and partly FEM based [10].

Table 4 shows results from the LTCA (using Langheinrich approach) for different profile modifications applied to gear 1 and 2. For this analysis, the same material combination and gear geometry was used as before (Chapter 3). For the asymmetric gear, only the pressure angle was changed to 15° (left) and 30° (right). The theoretical contact ratio ε_{α_th} for the right flank in contact is 1.15.

Table 4: LTCA results for with different modifications applied – for asymmetric gears.

Solution nr.	1	2	3	4	5	6	7	8	9	10	**11**
Tip relief gear 1 Value Length	/	/	/	/	Arc 10 1	Arc 20 1	Arc 30 1	Arc 10 1	Arc 20 1	Arc 30 1	**Arc 23 2**
Tip relief gear 2 Value Length	/	Arc 10 1	Arc 20 1	Arc 30 1	/	/	/	Arc 10 1	Arc 20 1	Arc 30 1	**Arc 23 2**
Root stress 1, MPa	48.1	49.0	50.4	51.8	48.1	48.1	48.1	49.0	49.6	50.5	**45.6**
Root stress 2, MPa	45.3	44.2	44.2	44.2	44.9	45.6	46.2	44.9	45.5	46.2	**46.8**
Hertzian pr., MPa	84.8	84.9	85.0	85.2	77.6	78.0	78.3	77.6	78.6	84.3	**81.9**
Contact ratio ε_{α_eff}	1.58	1.55	1.53	1.48	1.57	1.55	1.53	1.53	1.50	1.45	**1.50**
ε_{α_eff} - ε_{α_th}	0.43	0.40	0.38	0.33	0.42	0.40	0.38	0.38	0.35	0.30	**0.35**
PPTE, µm	26.8	26.5	26.3	26.1	23.5	23.3	23.7	22.3	20.2	19.0	**17.4**
Wear, µm	47.2	48.2	47.2	48.9	41.9	35.3	24.7	41.7	35.3	28.0	**35.1**

Results from Table 4 show, that applying tip modifications only on gear 2 has almost no effect on the results (solutions 2-4). However, applying modifications only on gear 1 (solutions 5-7) can reduce transmission error by 10 % and wear by almost 50%. Additionally, Hertzian pressure is also reduced by almost 10%. In both cases, the root stresses are almost not changing from the unmodified tooth form. Simultaneously applying modifications on both gears (solutions 8-10) will result in lower transmission error (reduction by 30%) and wear reduction (by 40%). However, in this case, the root stress of gear 1 increases by about 5% while the Hertzian pressure remains the same. Solution 11 was again obtained by modification sizing. From all solutions, no. 11 shows the smallest transmission error (reduction by 35% compared to the unmodified tooth) and low wear (reduction by 25%). The root and flank stresses do not change significantly.

Compared to the symmetric design (Table 3), results for asymmetric gears (Table 4) show higher root stresses, but much lower wear and transmission error.

5. Influence of manufacturing errors

During the injection moulding process unintended manufacturing errors are produced (profile and helix deviations). Therefore, it is important to consider also the manufacturing errors in the modification selection process. Theoretical manufacturing errors depend on the required quality of the gears and may be substantial compared to the amount of the proposed modification. What if a good solution has a tip relief of 15 μm, but profile deviation according the gear quality is in the range of ±10 μm?

An optimal solution, as found in the previous chapter, must be checked for stability of the main parameters when profile and flank line errors are added.

To consider the manufacturing tolerances, again the modification sizing functionality can be used. This time the modifications of the final design are kept constant, but additionally profile and helix angle modifications are varied to simulate manufacturing errors. The procedure was described by Kissling [13].

6. Conclusions

Modifications are often applied on gears, both for steel gears as well as for plastic gears. For steel gears, profile and lead modifications are usually applied only in cases, where the quality of the gears is 6 or better. Otherwise, the amount of modification is in the same range as manufacturing errors – the effect of modifications would thus be eliminated. But for plastic gears, the applied modifications are much higher, therefore modifications should be applied also on gears in quality 10 or 11.

NVH is very important in gear design. Improvements in terms of noise and vibrations can be achieved by selecting appropriate material combinations, by increasing the microgeometry but also with applying profile modifications. If wear is present in the system, it can also change the NVH behaviour of the gears, so it is equally important in the gear analysis process.

The results from the LTCA shows, that by applying an appropriate profile modifications, wear and transmission behaviour can be significantly reduced both for symmetric and asymmetric gear designs, without significantly increasing the root and flank stresses. If tip relief is used, modifications should be distributed between both gears.

It is important to note, that modifications can be fully optimized only for one operating condition (torque, temperature, …). If the gearbox is operating under variety of different conditions, an “average” modification should be applied, which gives good results in all conditions.

During the injection moulding process unintended manufacturing errors are produced. Therefore, as last step in the design process, it is important to check how errors may affect the performance of the gear mesh in practice.

References

[1] Erhard, G.: Designing with Plastics, Hanser Gardner Publications 2006.

[2] Pogačnik, A., Tavčar J.: An accelerated multilevel test and design procedure for polymer gears, Materials and Design (2015), 65:8, 961-973.

[3] VDI 2736, Thermoplastic gear wheels, Part 1-4, 2014.

[4] Pogačnik A., Beermann S., Measuring permissible root/flank stress of plastic gears according to the VDI 2736, VDI Gear conference, 2017.

[5] Kim H.K., et. all, Noise reduction of a high-speed printing system, using optimized gears based on Taguchi’s method, Journal of Mechanical Science and Technology 24 (12), 2010.

[6] KISSsoft gear calculation software, 2019, www.KISSsoft.com.

[7] Koop, M., Melnikov E., Merz V.: Design investigations and indications for acoustical optimized gear meshes using plastic gears, VDI-Report, 2294, 2017.

[8] Weber, C.and Banaschek, K., 1953, Formänderung und Profilrücknahme bei gerad- und schrägverzahnten Rädern; Schriftenreihe Antriebstechnik, Heft 11, Friedr. Vieweg & Sohn, Braunschweig.

[9] Langheinrich, A: Geometrie, Beanspruchung und Verformung asymmetrischer Stirnradverzahnungen, Dissertation TU München, FZG-Nr. 201/2014, 2014.

[10] Mahr B., Langheinrich A., Pogačnik A., Derivation of tooth stiffness of asymmetric gears for loaded tooth contact analysis, VDI Gear conference, 2019.

[11] Kissling U., Hauri S., Verschleiss von Zahnrädern: Berechnung Verschleissfortschritt und Prognose der verschlissenen Zahnform, Antriebstechnik Marktübersicht, 2015.

[12] ISO1328-1, Cylindrical gears — ISO system of flank tolerance classification — Part 1: Definitions and allowable values of deviations relevant to flanks of gear teeth, 2013.

[13] Kissling U., Sizing of profile modifications for asymmetric gears, AGMA technical paper, FTM, 2019.