- Pressione Hertziana
- Nuova Norm. 2736 VDI

KISSsoft
Calculation programs for machine design

Resistance and Lifetime Calculation of Plastic Gears

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SHARING KNOWLEDGE

Basics for Load Capacity Calculation

Basics for load capacity calculation

The basic problem in the calculation of the resistance (safety factors) and the life time of plastic gears is:

1. The measuring of material data (Woehler lines, also called S/N curves) is much more time consuming than for metals.
2. There are practically no material data available for modern plastic materials.

BUT:

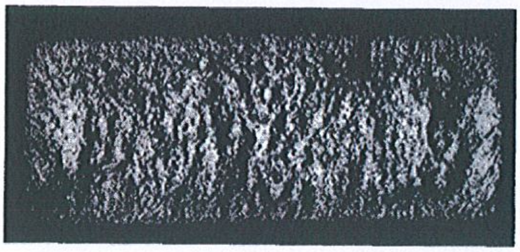
3. It is more complicated, but it is nevertheless possible to design successfully plastic gear reducers!
4. The possibilities to increase the life time of plastic gears are considerable.

Basics for load capacity calculation

IMPORTANT!!!

The basic proceedings for the load capacity calculations are the same for steel, other metals, sintered and PLASTIC gears

Static resistance (Overload breakage or permanent deformation)



Sudden break (brittle break) in a case-hardened spur gear

700:1 electromicroscopic image of the hardened layer of a tooth after breakage.
The fracture surface shows the distribution of intergranular fracture structure and transcrystalline fracture honeycomb structure.

Static resistance (Overload breakage or permanent deformation)

- Check with the peak torque against material static bending strength, some times also against elastic limit (to avoid permanent deformation).
- No notch effect has to be considered (local stress in notch areas is reduced by local flowing of the material).
- For gears: Only tooth breaking has to be checked (wear or Hertzian pressure is not critical if the peak torque is seldom).
- If the frequency of the peak torque is higher than approx. 1000 times in the total life time of the gear, it is a fatigue problem and not a static problem !

Static resistance (Overload breakage or permanent deformation)

What is special in the case of plastic gears ?

- Data is relatively easy to get (bending breaking test is commonly used by the material providers).
- Static bending resistance depends on the temperature.
- The test is not exactly identical to the bending of the gear tooth, therefore the required minimal safety should be > 1.0!
- Many applications, where plastic gears are used, have a low permanent torque and a few times a high peak torque:
- For example seat movers in cars. The seat is moving with a relatively low motor torque, the high peak torque happens, when the seat moves against block.

Static resistance (Overload breakage or permanent deformation)

Calculation method:

- As used for shafts (not documented by ISO gear standard)
- Stress calculation derived from ISO6336, but no YS factor

3. TOOTH ROOT STRENGTH

```
                                                  ------- GEAR 1 -------- GEAR 2 --
Tooth form factor                        [YF]           3.46                 2.97
Contact ratio factor                     [Yeps]                   0.625
Helix angle factor                       [Ybet]                   1.000
Effective facewidth (mm)                 [beff]        13.00                12.00
Nominal bending stress at tooth root (N/mm²)[sigF0]    11.37                10.58
Tooth root stress (N/mm²)                [sigF]        11.37                10.58
* KFa * KFb
Tensile strength (N/mm²)                 [Rm]          56.00                70.34
Safety for tensile stress                [Sb=Rm/sigF]   4.92                 6.65
Calculation formula:          sigF0 = Ft / beff / mn * YF * Yeps * Ybeta
                              sigF  = sigF0 * KA * KV
```

Fatigue resistance

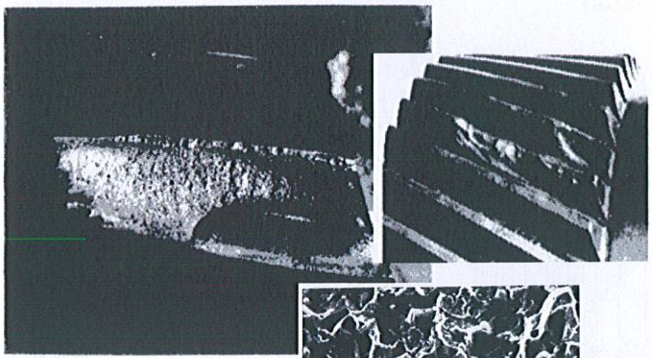


Bild 52 a.
Dauerbruch an einem einsatzgehärtet mit etwa 30 % Dauerbruchanteil

Root fatigue failure

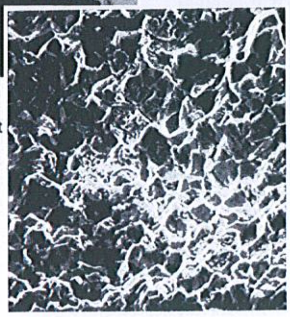


Bild 52 b.
Rasterelektronenmikroskop-Aufnahme 550 : 1 eines Dauerbruchs in der einsatzgehärteten Randzone eines Zahnes. Es liegt ein überwiegend interkristalliner, örtlich auch transkristalliner Bruch vor.

Pitting

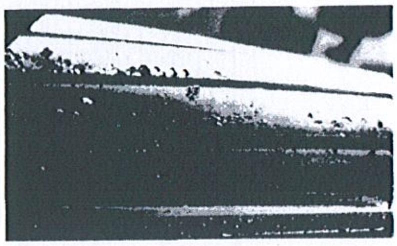
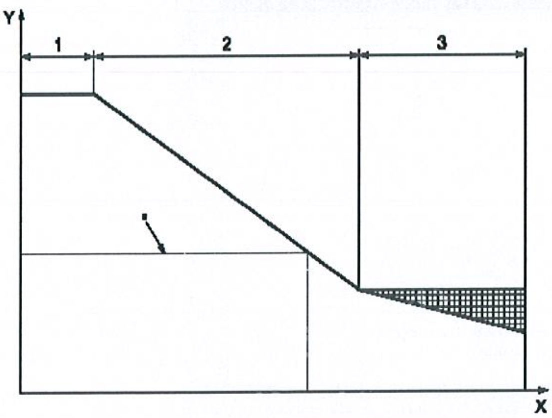


Bild 24.
Fortschreitende Grübchenbildung an den Zahnflanken eines Stirnrades aus Vergütungsstahl. Durch den Ausbruch benachbarter Materialteilchen ist es teilweise zu zusammenhängenden Materialausbrüchen gekommen.
(Maßstab 1 : 1)

Fatigue resistance

Material values depends on the no. of load cycles:
Woehler line (also called S/N curves)

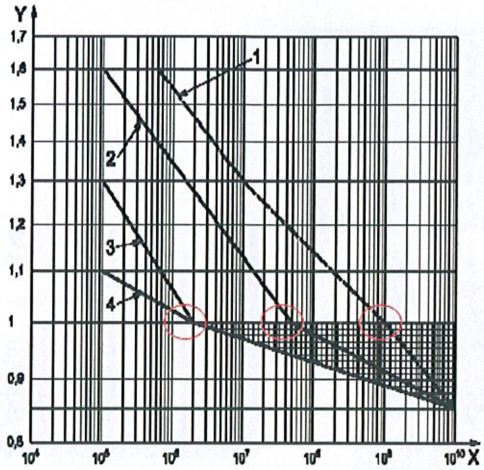
Check with the nominal torque against the admissible material value for bending and for pressure/wear.



Woehler line:
- 1. static domain
- 2. limited life domain
- 3. long life or endurance domain

Fatigue resistance

Z_{NT} for static and reference stresses may be taken from Figure 6 or Table 2.



Key

X number of load cycles, N_L

Y life factor, Z_{NT}

1 St, V, GGG (perl., bai.), GTS (perl.), Eh, IF [a]

2 St, V, GGG (perl., bai.), GTS (perl.), Eh, IF

3 GG, GGG (ferr.), NT (nitr.), NV (nitr.)

4 NV (nitrocar.)

Metals:

The characteristic of the Woehler line is known for different steel types (as case carburized, through hardened, etc.). For a specific material, only the σHlim (for Hertzian pressure) and the σFlim (for bending) must be measured.

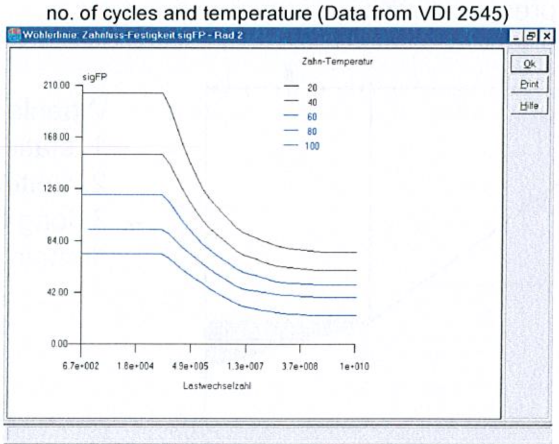
σHlim, σFlim are the material values for the endurance domain of the Woehler line.

Fatigue resistance

What is special in the case of plastic gears ?

- Different Woehler lines for different temperatures !
- Furthermore different Woehler lines for different lubrication!

Woehler lines for bending stress of PA66 depending on no. of cycles and temperature (Data from VDI 2545)



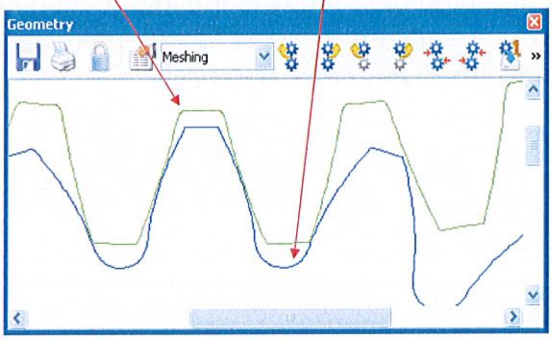
Fatigue resistance: Bending

Local stress increase by notch effects has to be considered.

In Gears: Notch effect in the root area is very important.

In Gears: Notch factor is called 'Stress concentration factor' YS

High notch factor   Low notch factor



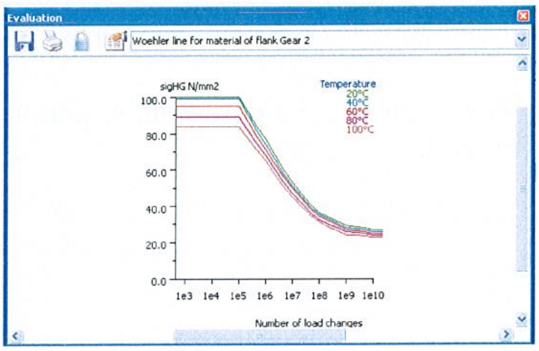
Fatigue resistance: Hertzian pressure / Wear

In metals, Hertzian pressure produces pitting.

In plastics, pitting is known only in special cases (with strong materials such as PEEK).

In plastics wear is the life time limiting criteria. Wear depends on Hertzian pressure, sliding velocity and lubrication.

VDI2545 proposes Woehler lines for admissible pressure to avoid wear.



Fatigue resistance

What is special in the case of plastic gears ?

- Data is VERY complicated to get (Woehler lines must be measured for different temperatures for bending and for wear/Hertzian pressure).
- The only documented data today is the information in the VDI2545. These data was measured in the 1970's in Germany. The data available is only for POM, PA12 and PA66.

Fatigue resistance

VDI 2545:

The only world-wide recognized method for designing plastic gears.

Derived from DIN3990 for steel, but includes

temperature dependency and Woehler line for plastics

Was withdrawn a couple of decades ago, but is still used as there is no replacement.

Development of a new edition, with a new number, VDI 2736, was started in 2007.

Fatigue resistance

Differences between the calculation according to VDI 2545 and G. Niemann (Tech. University, Munich)

	Niemann	VDI2545	VDI2545-modif.
Root calculation:			
Y_F	Method C	Method B or C	Method B or C
Y_S	ISO 6336	1.0 fixed	ISO 6336
Y_ε	1.0 fixed	$= 1 / \varepsilon_\alpha$ (*1)	$= 1 / \varepsilon_\alpha$ (*1)
Y_β	1.0 fixed	ISO 6336	ISO 6336
Y_{ST}	2	1	2
σ_{FE}	$= Y_{ST} * \sigma_{Flim} = 2\sigma_{Flim}$	$= Y_{ST} * \sigma_{Flim} = \sigma_{Flim}$	$= Y_{ST} * \sigma_{Flim} = 2\sigma_{Flim}$
Flank Calculation:			
Z_ε	1.0 fixed	ISO 6336	ISO 6336
Z_V	1.0 fixed	1.0 fixed	1.0 fixed
Z_R	1.0 fixed	1.0 fixed	1.0 fixed

Tooth deformation: Very different calculation methods.

(*1): Only for Method C, otherwise $Y_\varepsilon = 1$

→ The new edition of VDI2736 will be similar to the variant VDI2545-modif.

IMPORTANTE

Check static strength or fatigue?

T_p / T_n	Load cycle of T_p	Applicable calculation method
>= 3.5	< 100 ... 1000	Static method with T_p
	> 100 ... 1000	Fatigue method with T_p
2.0 ... 3.5	< 100 ... 1000	Static method with T_p and Fatigue method with T_n
	> 100 ... 1000	Fatigue method (Load spectrum with T_p and T_n)
<= 2.0	< 100 ... 1000	Fatigue method with T_n
	> 100 ... 1000	Fatigue method (Load spectrum with T_p and T_n)

T_p: Peak torque
T_n: Nominal torque
Blue: Usual case
Static method: Verification of static strength against breakage or deformation
Fatigue method: For dry running, proof of wear safety
For oil or grease lubrication, proof of fatigue safety

Proceedings for Load Capacity Calculation

Checking resistance

What can you do, if static resistance has to be checked, and no material data is available ?

→ Ask the material supplier for bending strength data (for different temperatures)

→ Then perform a static resistance calculation

→ For safety against tooth breaking we recommend SFmin = 2

FATIGUE
ETC.

Checking resistance

What can you do, if fatigue has to be checked?

When the material is POM, PA12 or PA66 (or similar), use VDI2545 method and material data.

→ You have to define carefully the required minimum safety factors

If you use other materials and no material data is available ?

(1) You may get material data with a test rig (needs months or years!)

(2) You may use VDI2545 data and adapt the required minimum safety (by guess!)

(3) You may 'extract' the material data from existing design.

(4) You may use ISO6336 (for steel) introducing a approximative value for sHlim, sFlim.

→ (3) is actually the best solution in practice!

Checking resistance

What can you do, if fatigue has to be checked?

When the material is POM, PA12 or PA66 (or similar), use VDI2545 method and material data.

→ You have to define carefully the required minimum safety factors

A calculation method like VDI2545 defines how the theoretical safety factors are defined but not which factors are required.

FATTORI DI SICUREZZA

	Root	Flank / Wear
Module > 2	2.0	1.4
Module 1.0..2.0	1.6	1.2
Module 0.5..1.0	1.2	1.0
Module < 0.5	0.8	0.8

Checking resistance

How to define the required safety factors for use with VDI2545 for your application?

- Determine general factors including material values
- Validate using existing (if possible also critical) gear boxes
- Determine the theoretical safety factors using these results
- Use these theoretical safety factors for the sizing of new gear boxes and incorporate all new experience in the further definition of the factors

Calculation Method + Experience (know how) = Optimal Results

Checking resistance

<u>If you use other materials and no material data is available ?</u>
(3) You may 'extract' the material data from existing design.

- Do have an existing gear box using the same (or similar) material, which performs well or which you have tested on the test rig?
- If YES, then you can use this method:
    - Perform a resistance calculation of the existing gear box, using the same torque as on the test rig and the same number of hours.
    - The resulting safety factors of this calculation are to be used as minimum required safety factors for the new design.
    - Consequence: Your new design will perform as well as the existing design.

Calculation Method + Experience (know how) = Optimal Results

Checking resistance

An important note concerning the use of data from existing design

- If your existing design is over dimensioned, then the resulting required safety factors from the existing design may be too high.
- So also your new design will be over dimensioned!
- We recommend therefore, if you test a gearbox on the test rig, and if the required life time is achieved without any problem, to repeat the test with a higher torque.
- If with higher torque, you get a failure of the gears after a certain number of hours X, this is exactly the information you are looking for: Repeat the calculation with the higher torque and life time X, you will get exactly the minimum required safeties!

Implementing Data for Plastic Gears in KISSsoft

For static check of tooth breaking and
for fatigue check based on VDI2545 method

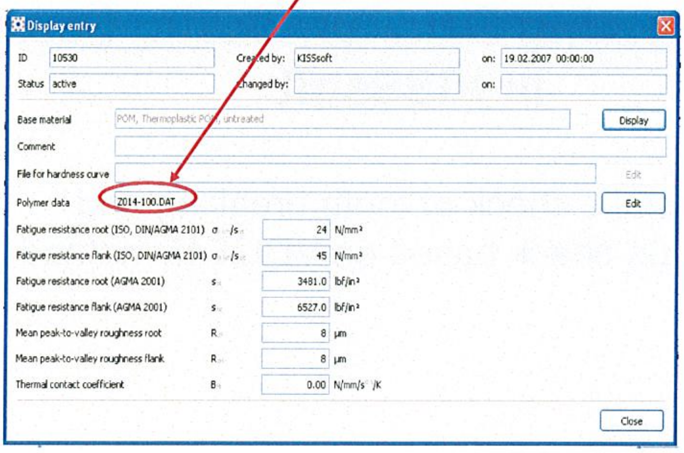
Implementing data for plastic gears

Data for Plastic Gears are depending on temperature.

The Woehler line (S/N curve) of Plastic materials must be measured.
No 'synthetic' Woehler diagram as for steels can be used.
→ Plastic data must therefore be treated completely different with steel data. In KISSsoft all specific data of a Plastic material are defined in a text file.

Implementing data for plastic gears

In the Gear material data base only the name of the file, containing the important data, is indicated.



Implementing data for plastic gears

An example for fatigue:

The table contains data for σ_{Hlim} (admissible Hertzian pressure) in N/mm² from 0 to 10^ 99 load cycles and from 20 to 140 °C.

Intermediate values are interpolated linear with temperature and logarithmic with cycles.

```
-- Comment: PA66, sigHlim (flank) for PA66, grease lubricated
-- Source: VDI2545
--
:TABLE FUNCTION FlankenSigHNFett
      INPUT X ZahnTempFlanke TREAT LINEAR
      INPUT Y Lastwechsel    TREAT LOG
DATA
         20      40      60      80     100     120     140
0       100      99      95      89      84      77      69
1E5     100      99      95      89      84      77      69
1E6      77      74      71      68      65      58      52
1E7      53      51      50      47      45      41      36
1E8      36      35      34      32      31      28      25
1E9      29      28      27      26      24      22      20
1E10     27      26      25      24      23      21      19
1E11     26      25      24      23      22      20      18
1E99     26      25      24      23      22      20      18
END
```


Implementing data for plastic gears

An example for static bending resistance:

The table contains data for static bending resistance in N/mm² from -20 to 120 °C and data for the elastic limit in N/mm² from -20 to 120 °C.

Intermediate values are interpolated linear with temperature.

```
-- Comment: Data for POM, static bending resistance - N/mm2
-- Source: from DUPONT for Delrin 111P NC010
:TABLE FUNCTION SigFb
      INPUT X ZahnTempFuss TREAT LINEAR
DATA
    -20     23     40     90    120
     88     70     62     34     22
END

-- Comment: Data for static deformation limit stress - N/mm2
-- Source: from DUPONT for Delrin 111P NC010
:TABLE FUNCTION SigFs
      INPUT X ZahnTempFuss TREAT LINEAR
DATA
    -20     23     40     90    120
     81     64     56     31     20
END
```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Implementing data for plastic gears

An example for Young's modulus (of elasticity):
The Young's modulus of Plastic materials is depending on temperature.

The table contains data for Young's modulus in N/mm2 from -20 to 120 °C.

```
-- Comment: Data for POM, Young's modulus, N/mm2
-- Source: from VDI2545
:TABLE FUNCTION ElastizitatsModul
      INPUT X ZahnTempFuss TREAT LINEAR
DATA
    -20      0      20     40     60     80     100     120
   4400    3950   3500   2950   2400   1800   1400     950
END
```


Using ISO 6336 for Plastic Gears in KISSsoft

For adaptation of calculation method for steel gears to plastic gears

Using ISO 6336 for plastic gears

Be careful, when using ISO6336 or DIN3990 for plastic materials.

Messages

Calculation of plastics according to DIN3990 or ISO6336:
An exact calculation taking into account the actual stress-cycle diagram (Woehler line) and the temperature dependence can only be executed according to VDI2545.
An approximately determination of service life can be executed in KISSsoft according to DIN3990 or ISO6336.
The permissible material data sigFlim and sigHlim have to be examined,
sigFlim and sigHlim should be valid for a cycle number of approximate 10^8.

(The values in data base are valid for a temperature of 70° and a number of load changes of 10^8.)

MEGLIO VDI2545

Using ISO 6336 for plastic gears

Be careful, when using ISO6336 or DIN3990 for plastic materials.

The material values σHlim and σFlim (for endurance) must be introduced. They have to be according to the effective service temperature, because ISO6336 does not consider changing of material proprieties with temperature.

There is no known correlation between endurance data and static resistance in plastics. Therefore such data is also difficult, if not impossible, to get from the material supplier.

The Woehler line for plastic is different from the Woehler line for steel, there in the life-depending and in the static domain of the Woehler line some errors occur.

Using ISO 6336 for plastic gears

Some hints:

Set Kv, KHβ, KHα Factors to 1.0 (as used in VDI2545).
For accurate calculation of Hertzian pressure, set also the Young modulus according to the service temperature.

Some advantage:

If you would like to compare the dimensions of different gear types (bevel compared to worm or to face gear reduction for example), then using the same material values in different calculation methods can give quick and good results.

KISSsoft
Calculation programs for machine design

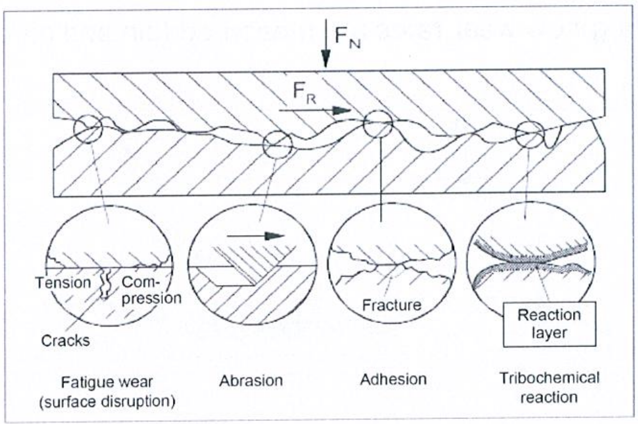
Wear on gears

Predicting of the worn tooth form and the consequences on NVH and lifetime

SHARING KNOWLEDGE

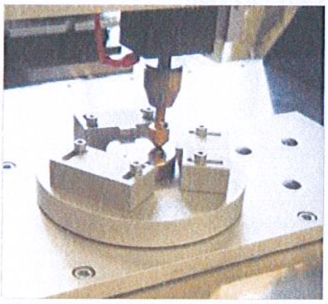
1. Theory of wear

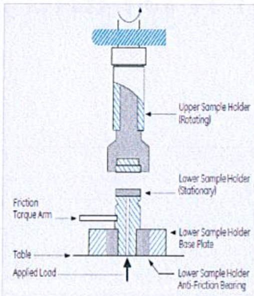
The term "wear" is used to describe the progressive removal of surface material due to mechanical and/or chemical stress.
The four main wear processes defined in DIN 50323 (DIN 50323, 1995) are adhesion, abrasion, surface disruption and tribochemical reaction.

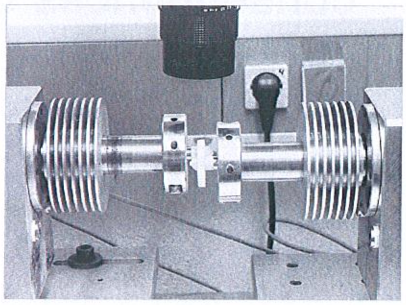


2. Determining the wear coefficient

The simplest method of measuring wear is to press a pin made of the material being investigated against a rotating ring. In the plastics industry this method is known as the "pin and disk test rig" test.
When investigating plastics, this ring is usually made of metal with a low surface roughness R_z. A "thrust washer apparatus" is also used. In this test two disks are pressed together.





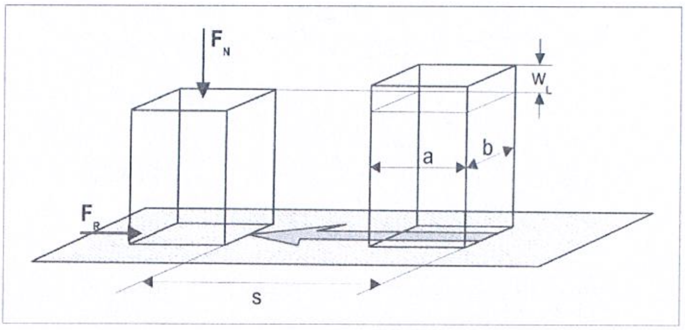


2.1 Determination of the wear coefficient

The wear coefficient k_w derived by using the pin and disk test rig – is defined as follows:

$$w = k_w \cdot p \cdot v \cdot t \qquad (1)$$

Diagram showing how wear rates are measured (pin and disk test):



2.1 Determination of the wear coefficient

As the wear coefficient is very small, it is expressed in $mm^2/N \cdot 1e\text{-}9$ instead of mm^2/N.

More commonly in literature the coefficient is expressed in mm^3/Nm. This corresponds to

$$\frac{1\ \mathrm{mm}^2(1e-9)}{1N} = \frac{1\ \mathrm{mm}^2(1e-9)}{1N}\left(\frac{1000\ \mathrm{mm}}{1000\ \mathrm{mm}}\right) = \frac{1\ \mathrm{mm}^3(1e-6)}{1\ \mathrm{Nm}}$$

This unit for k_w is used in VDI 2736-1.

2.2 Calculation of the local wear on the tooth flank

The formula for wear (1) can now be converted as follows for use in the tooth flank for an infinitesimal period of time Δt:

Surface pressure $p = F_n / A$
Momentary area $A = b \cdot v_p \cdot \Delta t$

Applied in (1):

$\Delta W = k_w \cdot F_n / (b \cdot v_p \cdot \Delta t) \cdot v_g \cdot \Delta t$

Converted:
$\Delta W = k_w \cdot F_n / b \cdot v_g / v_p$; as defined in DIN ISO 21771 v_g/v_p is the specific sliding ζ; so then the local wear on the tooth flank for N load cycles is

$$w_{lokal} = \frac{F_{n.lokal}}{b} \cdot N \cdot \zeta \cdot k_w \qquad (2)$$

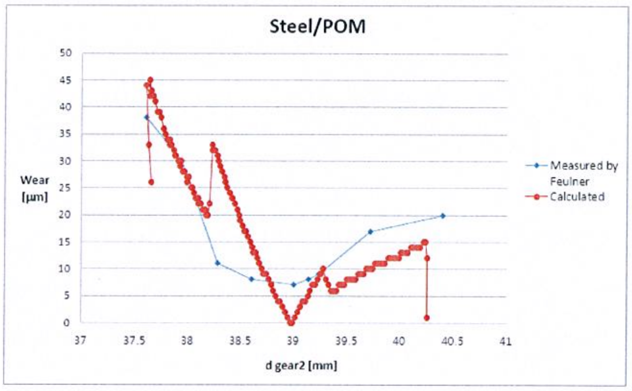
2.3 Calculation of mean wear and safety against wear

For layout purposes, it would be helpful to calculate directly the mean wear on the tooth flank.

Clearly, this could be obtained by calculating the mean value of the local distribution of wear across the tooth flank.

Result for mean wear:
Measured 16.4 µm,
calculated 17.8 µm.

However, this is a very complicated approach!



2.3 Calculation of mean wear and safety against wear

A much simpler method is to apply a mean value formula, such as the one derived from investigations carried out by Plewe:

$$\delta_W = \frac{\pi \cdot (F/b) \cdot N \cdot H_V \cdot k_w}{(l_{Fl}/m_n)} \tag{3}$$

where the loss factor is H_V and the length of the active tooth flank is l_{Fl}:

$$l_{Fl} = \frac{d_b}{4} \cdot \left(\tan^2\left(\arccos\left(\frac{d_b}{d_{Na}}\right)\right) - \tan^2\left(\arccos\left(\frac{d_b}{d_{Nf}}\right)\right)\right) \tag{4}$$

with d_b as the base diameter; d_{Na} tip active diameter; d_{Nf} root active diameter. These equations are documented in VDI 2637.

2.3 Calculation of mean wear and safety against wear

This is a simple method to obtain the safety against wear. The calculation is performed in a similar way to the method for calculating wear for steel-bronze worm gear pairs as defined in ISO 14521.

The resulting wear safety S_W is then the quotient from the permitted wear δ_{Wlim} to the occurring wear δ_W during the required service life.

WEAR

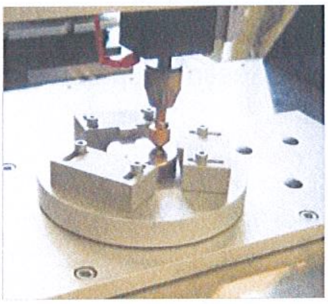
Line load at reference diameter (N/mm) [w]	7.13			
Line load at reference diameter (N/mm)	$[K_A{*}K_\gamma{*}K_V{*}K_{H\beta}{*}K_{H\alpha}{*}w]$		7.13	
Loss factor	$[H_V]$		0.178	
Length of active flank (mm)	$[l_{Fl}]$	2.70		2.60
Wear factor ($mm^3/Nm/10^6$)	$[k_W]$	0.60000		1.03000
Data for kw2 from file	Z014-100.DAT			
Normal tooth thickness in pitch circle (mm)	$[s_n]$	2.12		1.97
Maximum permissible wear (%)	$[W_{limit}]$		20.00	
Permissible wear on flank (mm)	$[\delta_{Wlimn}]$	0.42		0.39
Wear removal (mm)	$[\delta_{Wn}]$	0.24927		0.14419
Wear removal (mg)	$[=lFL{*}b{*}z{*}ro{*}\delta Wn]$	204.0		494.2
Safety against wear	$[S_W]$	1.70		2.73

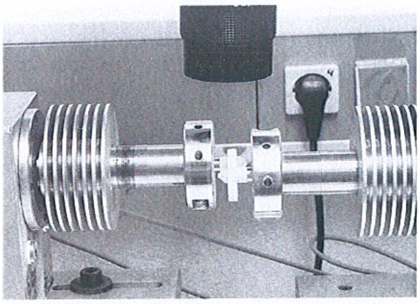
2.4 Determining the wear coefficient k_W

Wear can be measured quite easily by using a pin and disk test rig.

Using gear testing apparatus to measure wear involves significantly more time and effort.

But: Are k_w-values obtained by pin-disk rig reliable ?





2.4 Determining the wear coefficient k_W

Feulner measured k_w with both methods:

Material pairing	Data Gear test rig	k_w Gear test rig	Data Pin and disk test rig	k_w Pin and disk test rig	k_w Pin and Disk test rig at R_z 0.45 µm *
Steel/POM	1500-3000 rpM R_z 0.45 µm	1.03-1.34	v_g 0.5 m/s R_z 1.5 µm	3.4	1.0
Steel/PBT	As above	3.66-3.69	As above	7.8	n/a

Nearly equal

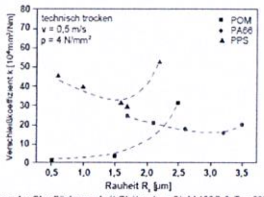


Bild 3.2 Einfluss der Oberflächenrauheit Gleitpartner Stahl 100Cr6, T_U = 23°C [3]

* Estimated in accordance with the surface roughness shown in Figure 3.2 in [4]

The roughness R_z (pin and disk) of 1.5 µm to R_z (test rig) of 0.45 µm has a significant effect, as shown in the figure.

2.4 Determining the wear coefficient k_W

Even if this first check is positive -> Today, not enough researched data are available to decide if a measurement taken using a pin/disk-rig can be used to determine useful values for gear calculations.

It must not be forgotten that a difference of "only" a factor of 2 between the measured coefficients is actually a good result because it can be used to estimate the service life to an accuracy of +-50%.

However, if the effect of lubricants are to be considered when measuring wear coefficients, it is almost certain that gear testing apparatus must be used.

2.5 Lubrication

Wear happens in not lubricated reducers.

Oil lubricated gears fails normally by tooth breaking; if a bit of oil is involved, wear can be observed only in rare cases.

Wear rate on a PA12 spur gear measured by Fürstenberger:

Lubrication	Wear coefficient k_w, $mm^3/Nm \cdot 1e\text{-}6$
Dry running	4.20
Grease	0.30
Oil	0.14

Wear coefficient in an oil bath is about 30 times lower than when dry-running!

2.5 Lubrication

Most gear drives with plastic gears are grease-lubricated.

For grease a prognostic is more difficult:

The conditions during the life cycle of the gear-reducer can change heavily:

From well lubricated -> dry running.

In most cases grease lubricated reducers do not have a high risk of wear.

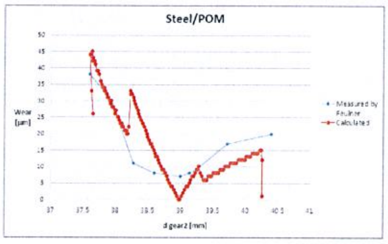
3. Determining the progression of wear

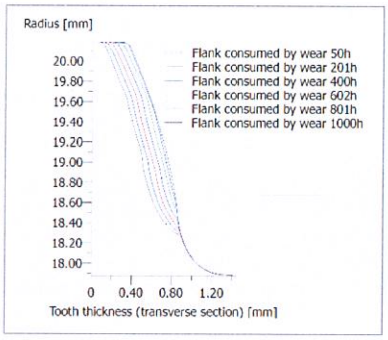
The wear on the tooth flank can be calculated using the formula for local wear in a LTCA.

But the result is 'wrong'!

No usable results, if the wear is calculated in a single step.

The wear characteristics must be calculated step by step because the tooth form changes as it becomes worn, and therefore the load distribution will change across the meshing.



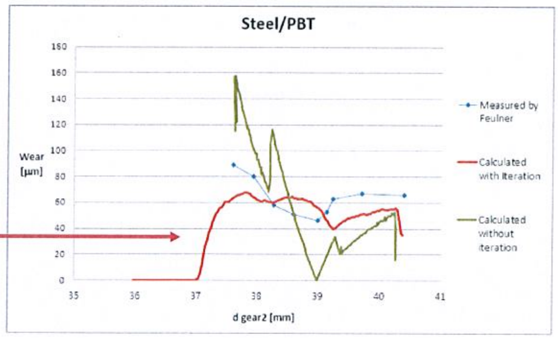


3. Determining the progression of wear

The algorithm is implemented in the LTCA (loaded tooth contact analysis) of KISSsoft.

The maximum permitted wear per iterative step must be predefined so that the iterative progression of wear can be calculated. This predefined value is critical for achieving realistic results.

Result by Iteration

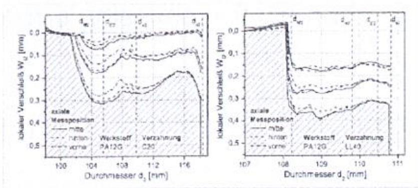


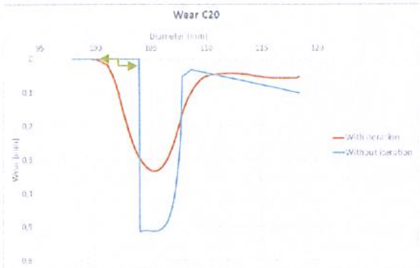
3. Determining the progression of wear

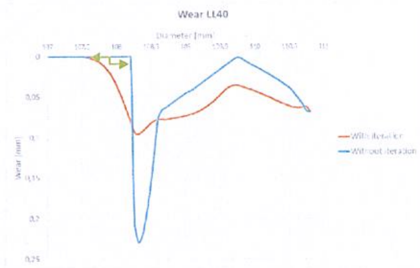
A recently published measurement was made by Fürstenberger at the FZG.

Right: Worn profiles of two different PA12 gears, measured by Fürstenberger.

Below: Result by simulation (brown: with simulation of wear progress, blue: without). The prolongation of the active flank in the root area is marked with a green arrow.



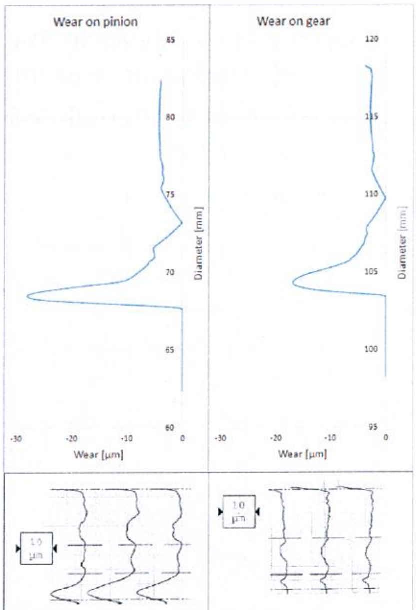




3. Determining the progression of wear

Further research showed that the method as described here has also a similar good correlation with worn tooth forms measured on slow running steel gears.

Wear measurements taken from case-hardened gears (module 4.5 mm) using the FZG gear test rig revealed a good match both with the distribution of wear on the tooth flank and with the amount of wear.

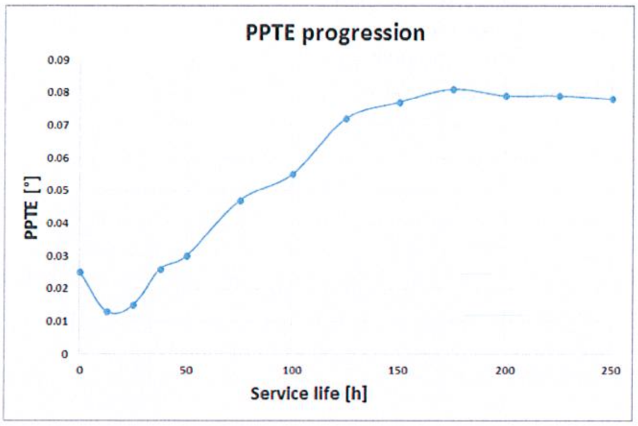


Profile of pinion
Tooth 1, tooth 6, tooth 11

Profile of gear
Tooth 1, tooth 9, tooth 17

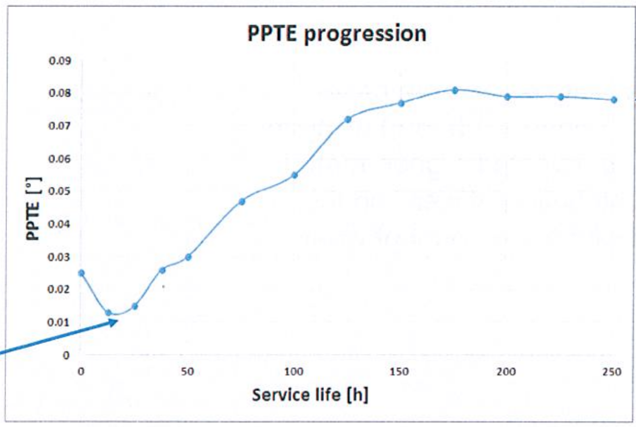
4. Effects of the wear progress

As the tooth form can now be predicted with reasonable accuracy, more detailed analyses, for example, defining the change in load distribution or the increase in transmission error due to wear, can be predicted.



4.1 Progression of transmission error

Increasing vibrations in the meshing over a long period of time can reduce the endurance limit. For this reason, being able to predict the progression of transmission error in advance is a very interesting result.

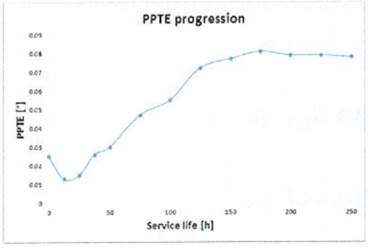


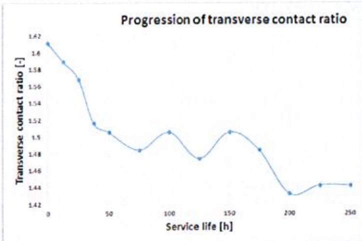
High plateau

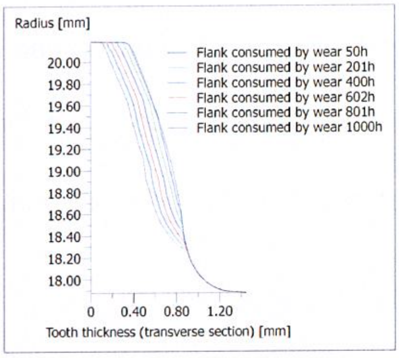
"running-in phase"

4.1 Progression of transmission error

Top: Progression of PPTE over the service life.
Bottom: Change in the transverse contact ratio.



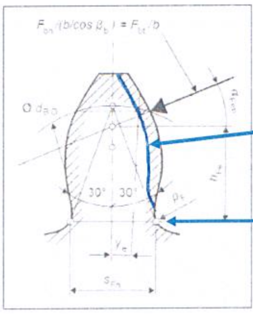




Worn tooth flank in several steps up to 1000 h

4.2 Lifetime

Due to the decreasing tooth thickness, the stresses will grow during wear progress. The calculation of the bending stress as proposed by VDI2736 (the method, described in ISO6336-3, is checking the bending stress in the section where the 30° tangents contact the root fillet) is not appropriate.



Worn tooth flank

'Critical' section according VDI2736

A better approach is to calculate the nominal shear stress in the tooth, starting in the section corresponding to the diameter of the (upper) single contact point $d_{B/D}$ down to the section of the active root diameter d_{Nf}.

4.2 Lifetime

Shear Stress Determination

A proposition is to use the formula for the allowable shear stress number
$\tau_{SE} = 0.577 * \sigma_{FE}$

τ_{nom} : Highest value found between section $d_{B/D}$ and d_{Nf}.

A mean notch factor for shear of 1.25 is appropriate.

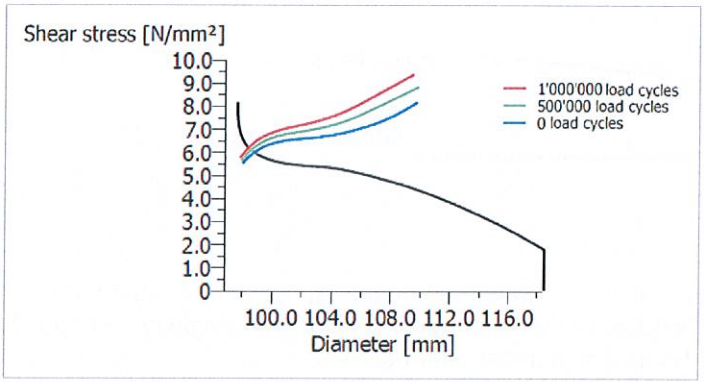
The safety factor shear stress is calculated according

$$S_\tau = \frac{\tau_{SE}}{(\tau_{nom} * 1.25)} = \frac{\sigma_{FE} * 0.577}{(\tau_{nom} * 1.25)} \leq S_{\tau\min}$$

A minimum safety factor $S_{\tau min}$ 1.5 is recommended.

4.2 Lifetime

Shear stress distribution in the tooth after 0, 0.5 and 1 mio cycles (in the meshing position with the highest load). The admitted stress [eq. 5] is 27 N/mm².



4.2 Lifetime

Fürstenberger proposes a simpler method, which uses the mean wear formula (eq. 3). The shear stress is calculated in the section of the active root diameter d_{Nf}, reduced by the mean wear value. The permissible shear stress is deduced from the permissible bending stress at 10^5 cycles.
As this method is relatively inaccurate:

- maximum shear stress is not in the d_{Nf}-section (normally higher up)
- the effective no. of cycles should be considered

a high minimum safety factor of 2.5 is recommended.

Summary

Two calculation methods are now available for calculating wear:

- An analytical method, which uses simple formulae to determine the mean wear when designing gear systems.
- A more complex method is integrated in contact analysis and is used to ascertain the progression of wear. The wear characteristics must be calculated step by step because the tooth form changes as it becomes worn, and therefore the load distribution changes across the meshing.

When these calculation methods are compared with measurements taken on test rigs and with results from real life situations, it can be seen that these methods produce useful, realistic results.
Therefore, it is now possible to predict the effect of a worn tooth form on the load distribution, transmission error and reduction of lifetime.