

Geometria degli ingranaggi

Teoria

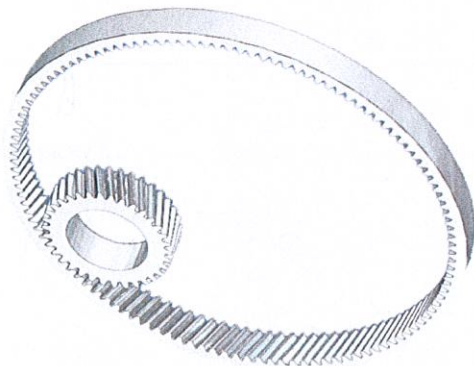
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SHARING KNOWLEDGE

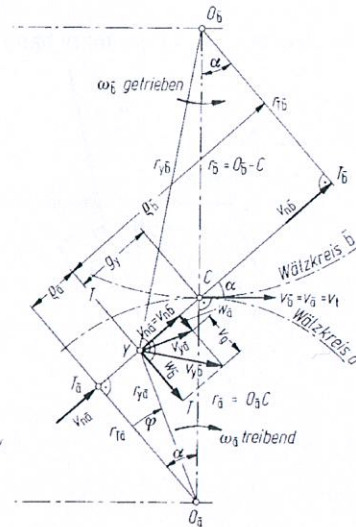
I. Dentatura teorica (senza gioco)

Nella prima parte viene trattata la
dentatura teorica (senza gioco)

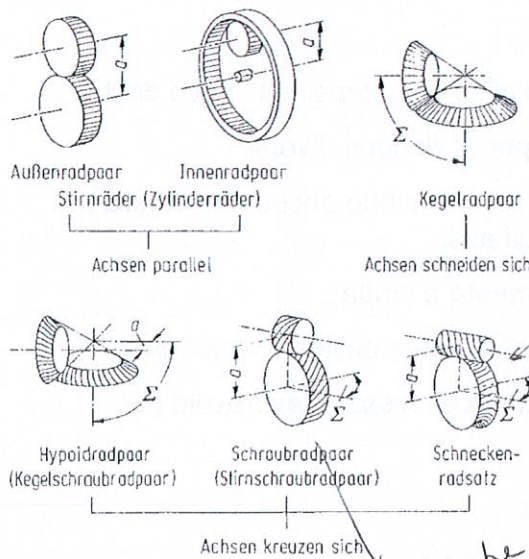


Velocità

- Velocità normali sono identici
- Velocità nella direzione della tangente nel punto di contatto w_a, w_b
- Velocità di strisciamento $v_{ga} = w_a - w_b$
- Strisciamento specifico
 $\zeta_a = v_g/w_a = 1 - w_b/w_a$
 $\zeta_b = v_g/w_b = 1 - w_a/w_b$
vel. in direz della tangente
 $(+3/-3)$
- Fattore di strisciamento $K_g = v_g/v_t$



Combinazioni di ingranaggi

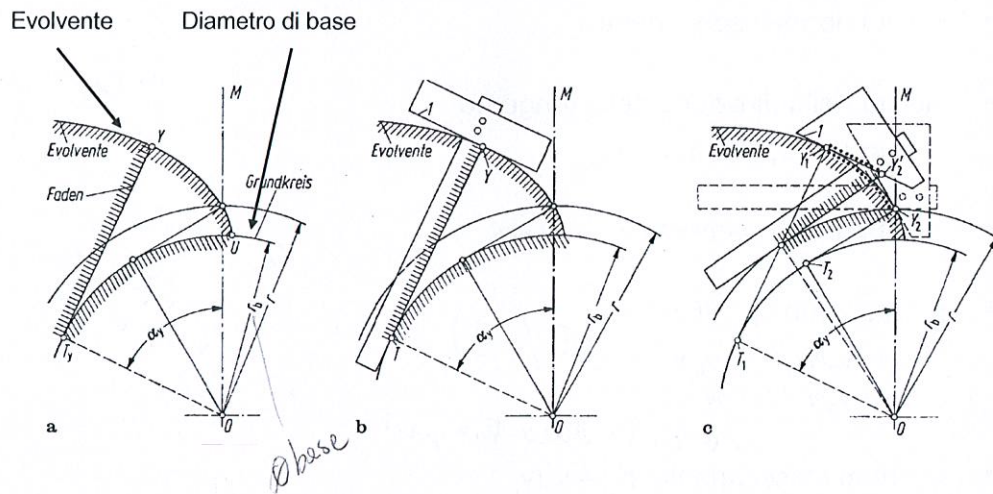


*dist. assi è negativa
 x indicare z della ruota $\frac{-100}{\text{negativo}}$*

*classico → contatto lineare
 globoidale*

sghebbate

Definizione dell'evolvente



L'evolvente è definito unicamente dal cerchio di base!

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Caratteristiche della dentatura a evolvente

- Fabbricazione semplice con utensile a fianco dritto
- Un unico utensile per n° di denti diversi
- Trasmissione del moto continuo anche con variazione della distanza tra gli assi
- La linea d'ingranamento é dritta
- Direzione e modulo costante della forza normale
- Fianchi con curvatura convessa (sfavorevole per pressione Hertz)

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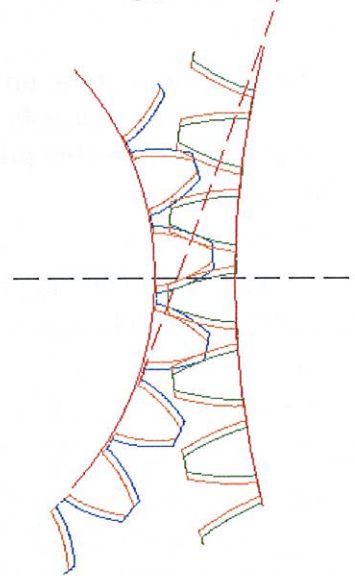
la cicloide ha un più alto errore di trasmissione

le conseguenze di un errore di trasam. si ripercuotono su tutta la trasmissione.
Dentatura a evolvente

Dal diametro di base d_b vengono generati degli evolventi equidistanti con passo p_b .

La linea d'ingranamento è tangente ai due cerchi di base.

I diametri dei cerchi di base sono nel rapporto del n° dei denti.



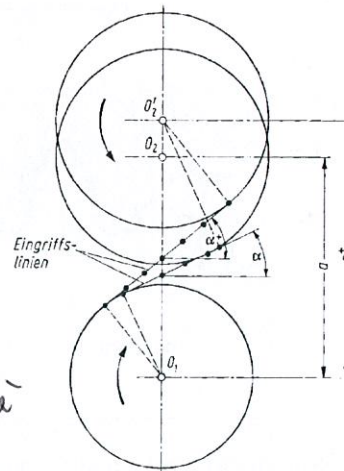
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Dentatura a evolvente

Se si cambia la distanza tra gli assi, cambiano soltanto la linea d'ingranamento e gli angoli d'ingranamento.

La dentatura ad evolvente è insensibile alla variazione della distanza tra gli assi.

l'errore di trasmissione è la variabile di non continuità del moto tra 2 ingranaggi



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Dentatura ad evolvente

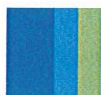
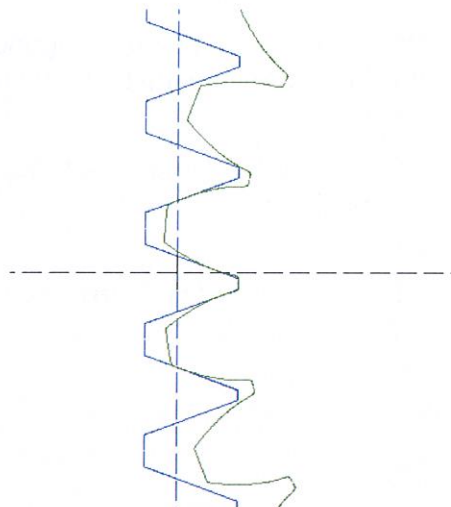
Per la fabbricazione, un utensile di tipo cremagliera con angolo di profilo α viene ingranato sul cerchio primitivo

$$d = d_b / \cos(\alpha)$$

Dal passo sul cerchio primitivo p_t viene definito il modulo

$$m_t = p_t / \pi$$

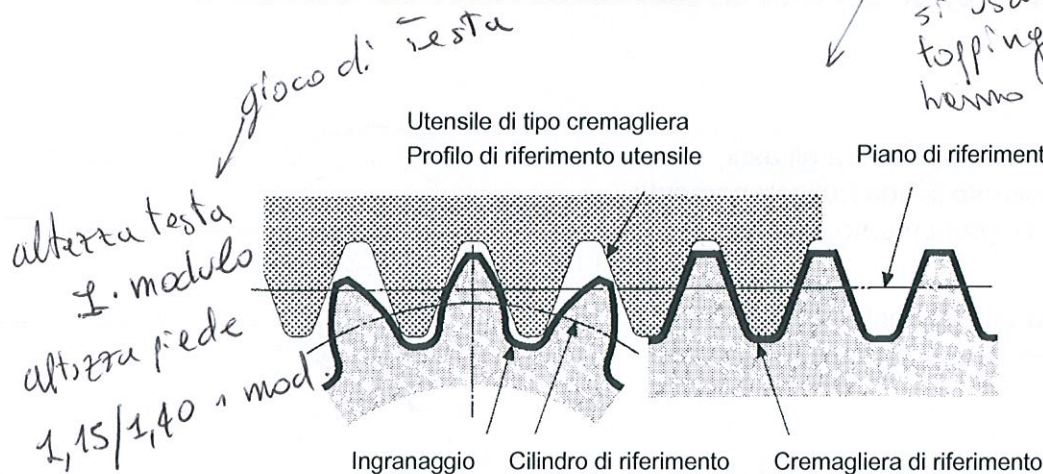
$$d = z \cdot p_t / \pi = z \cdot m_t$$



profilo di rif ← *utensile ingranaggio*

Profilo di riferimento

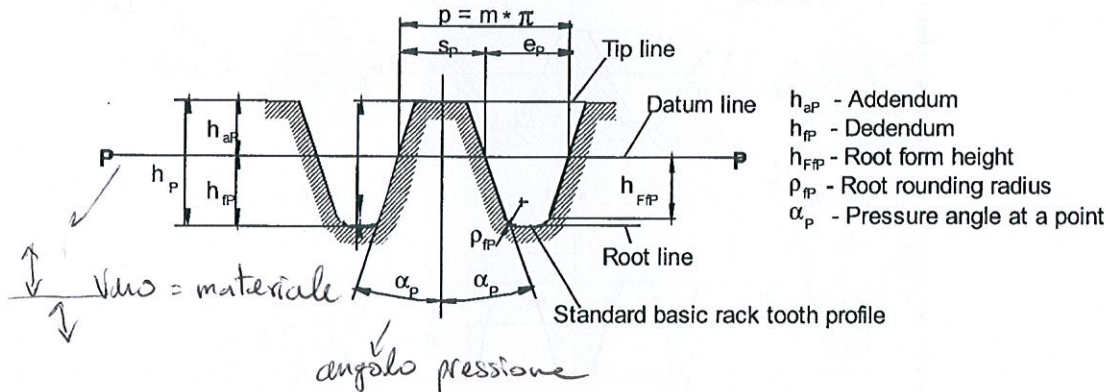
Fino da modulo 2 in su da modulo 2 in giù si usano utensili topping che non hanno giochi di Testa



Relazione tra ingranaggio, profilo di riferimento e utensile



Profilo di riferimento del ingranaggio 15053



Normalmente: $h_{aP}=1$, $h_{fP}>h_{aP}$ per l'ingranaggio

Per il profilo di riferimento dell'UTENSILE, altezza testa e piede sono scambiati, cioè h_{fP} (ingranaggio) uguale h_{aP0} (utensile)

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Profilo di riferimento secondo DIN 867

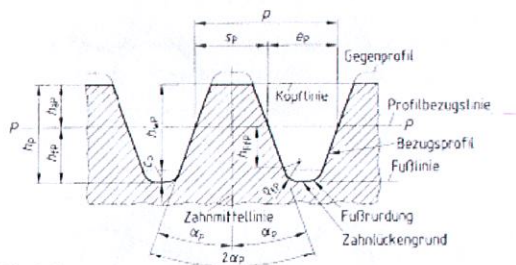
unica cosa definita

Fattore altezza testa $h_{aP} = 1$

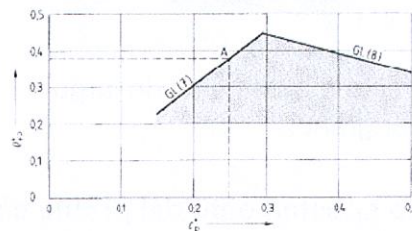
Fattore altezza piede $h_{fP} = 1 + c_p$

Fattore altezza gioco di testa
 $c_p = 0.1 \div 0.4$

Angolo di pressione $\alpha_P = 20^\circ$



Indicare xk cambia resistenza dentre



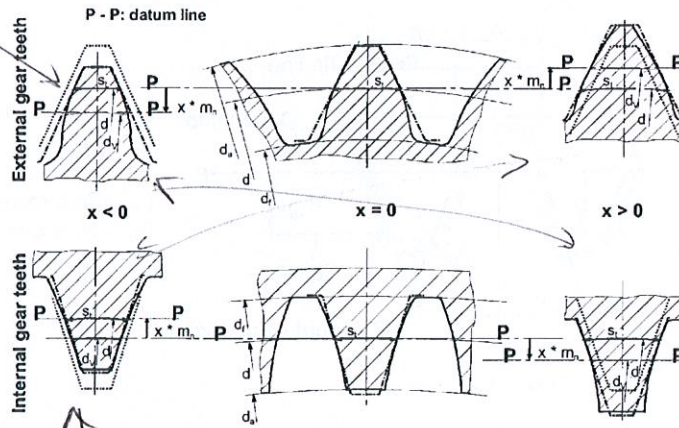
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Spostamento del profilo

non è la correzione!!

Mi avvicino con utensile

Mi allontano con utensile



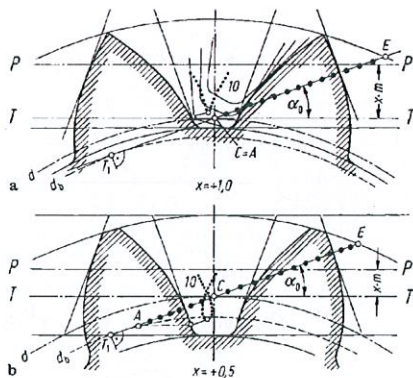
per l'interno si usa spesso negativo
L'utensile viene spostato di $x \cdot m_n$ nella fabbricazione.



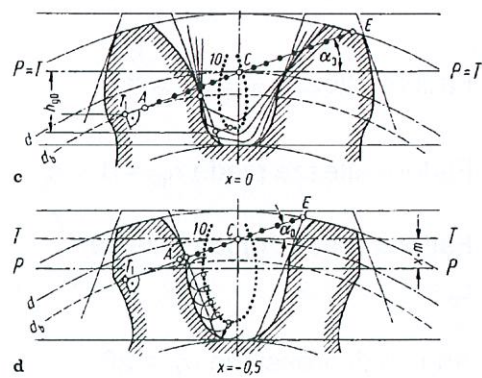
con la plastica?

Spostamento del profilo (x)

meglio di $x=0$



Dente a punta e piede largo per grandi x .



Interferenza di taglio per piccoli x .

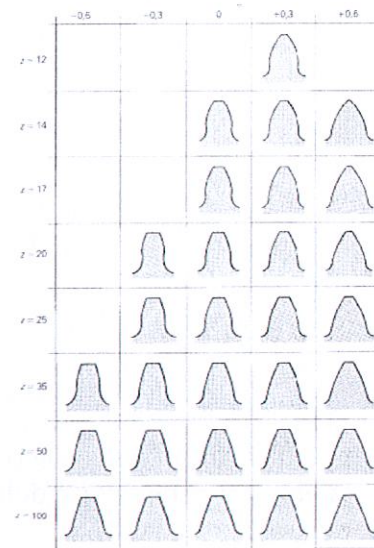
Lo spostamento del profilo cambia la forma del dente



la correzione e lo spostamento sono 2 cose diverse

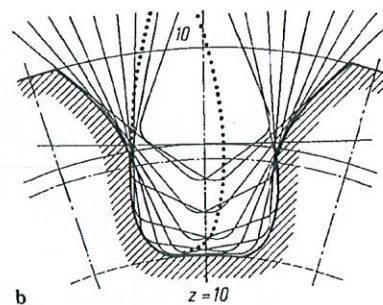
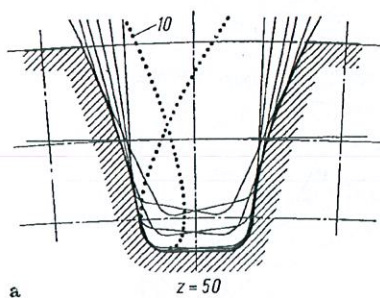
Spostamento del profilo

Lo spostamento del profilo cambia la forma del dente in maniera più evidente nelle ruote con pochi denti.



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Come si crea l'interferenza di taglio



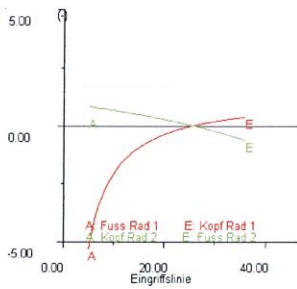
Si ha interferenza di taglio con un basso n° di denti.
Può essere evitata aumentando lo spostamento.

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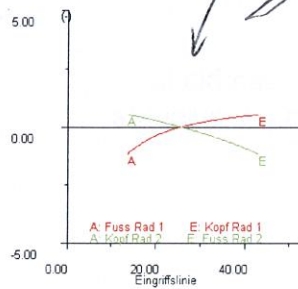
menu → grafica → analisi → scorrim. specifico

Strisciamento specifico

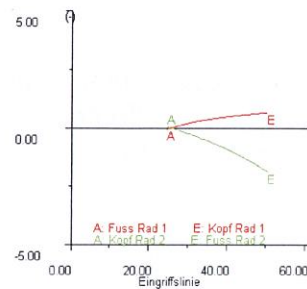
strisciamento specifico bilanciato



$x = -0.3$



$x = 0.27$



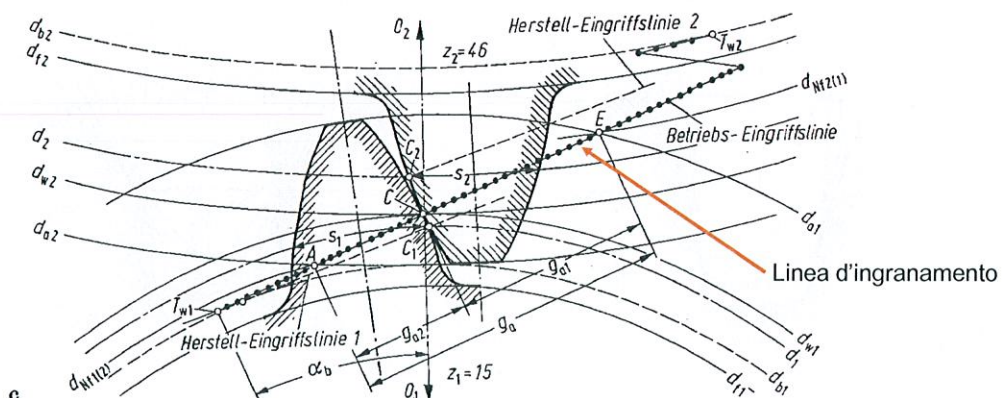
$x = 0.95$

Lo strisciamento specifico è il rapporto tra la velocità di strisciamento e la velocità in direzione della tangente alla forma del dente. E' il criterio per valutare l'usura.



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Linea d'ingranamento

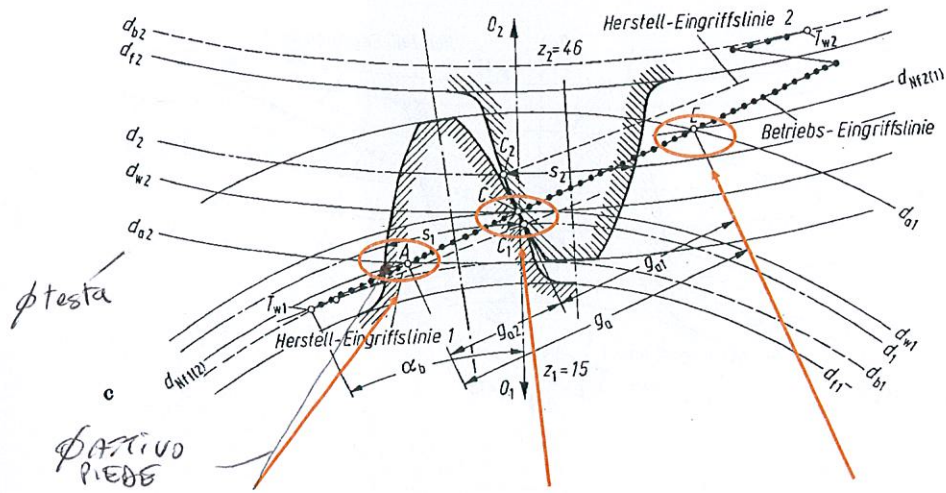


La linea d'ingranamento è un segmento tangente ai due cerchi di base.



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Linea d'ingranamento



A: Inizio del contatto

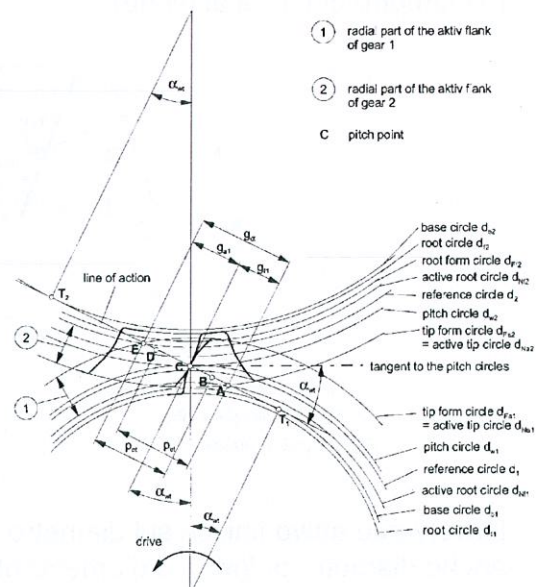
C: Punto di rotolamento

E: Fine del contatto



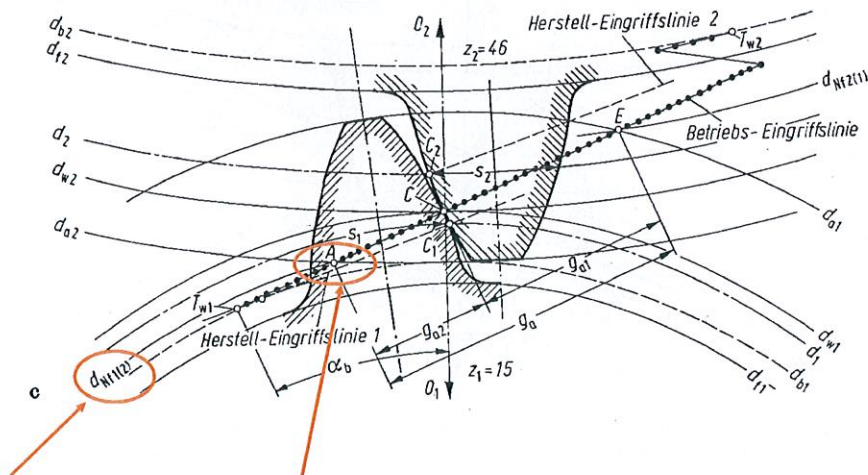
Fianco attivo

Il fianco attivo va dal cerchio attivo al piede fino al cerchio attivo alla testa.



Diametro attivo al piede d_{NF}

(è il diametro dove inizia il contatto con la ruota coniugata)



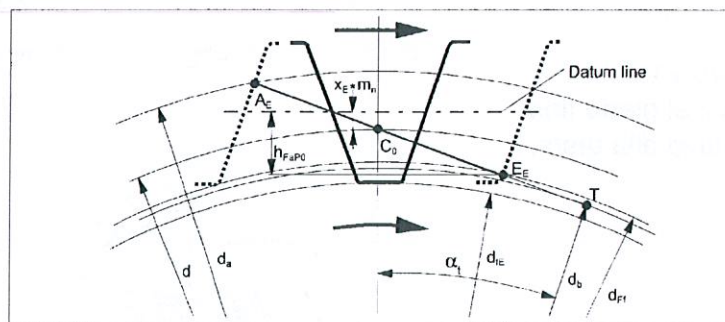
d_{NF1} si trova tramite l'intersezione di d_{a2} e la linea di contatto

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*φ attivo è il φ dove comincia il contatto
se φ forma piede < del φ attivo allora*

Diametro di piede utilizzabile

(o diametro di forma al piede)



C_0 - Pitch point of the generating gear
 A_E - Starting point of meshing
 E_E - End point of meshing

T - Contact point between the generating line of action and the base circle of the gear
 h_{FaPQ} - straight part of the tip flank of the tool generating profile

L'evolvente attivo finisce sul diametro attivo al piede d_{Ff} (chiamato anche diametro di forma o diametro utilizzabile d_{NF0})

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Formule di base

- Diametro primitivo (o di riferimento) $d = z \cdot m_t = z \cdot m_n / \cos(\beta)$
- Diametro di testa $d_a = d + 2 \cdot m_n \cdot h_{aP} + 2 \cdot m_n \cdot x + 2 \cdot m_n \cdot k$
- Diametro di piede $d_f = d - 2 \cdot m_n \cdot h_{fP} + 2 \cdot m_n \cdot x$
- Diametro di rotolamento $d_{w1} = 2 \cdot a \cdot z_1 / (z_1 + z_2)$
- Diametro di base $d_b = d \cdot \cos(\alpha_t)$
- Angolo d'ingranamento apparente $\tan(\alpha_t) = \tan(\alpha_n) / \cos(\beta)$
- Angolo di pressione $\cos(\alpha_{yt}) = d_b / d_y = d \cdot \cos(\alpha_t) / d_y$
- Spessore del dente $s_n = m_n \cdot (\pi / 2 + 2 \cdot x \cdot \tan(\alpha_n))$

Interasse

- Interasse di riferimento a_d : (per $\Sigma x = 0$)
 $a_d = (z_1 + z_2) / 2 \cdot m_n / \cos(\beta)$

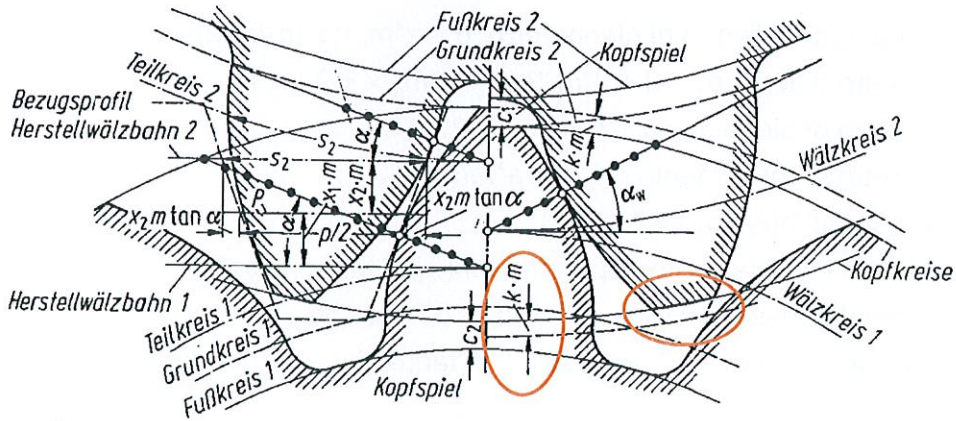
- Interasse a :
 $a = (d_{w1} + d_{w2}) / 2 = a_d \cdot \cos(\alpha_t) / \cos(\alpha_{wt})$
 $< a_d + \Sigma x \cdot m_n$

angolo contatto effettivo

L'interasse risulta dalla somma dei raggi di rotolamento. La variazione dell'interasse è più piccolo della variazione della somma di spostamenti.

- profilo rif. → modifica addendum nota dentata

Modifica del diametro di testa

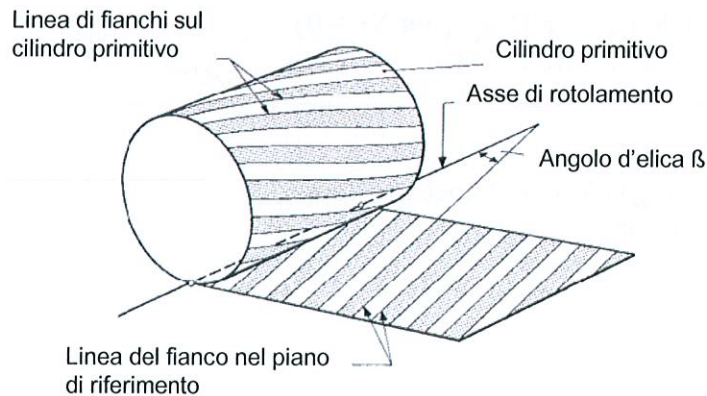


Il diametro di testa viene ridotto per mantenere il gioco di testa nel caso di somma di spostamento (Σx) positiva.

→ somma spostamenti
di corona e pignone

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Dentatura elicoidale



L'angolo d'elica β viene indicato sul cilindro primitivo.

Formula per altri cilindri: $\tan(\beta_y) = \tan(\beta) \cdot d_y/d$

angolo elica

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Dentatura elicoidale

Le dimensioni in direzione dell'altezza del dente sono uguali nella sezione apparente e normale.

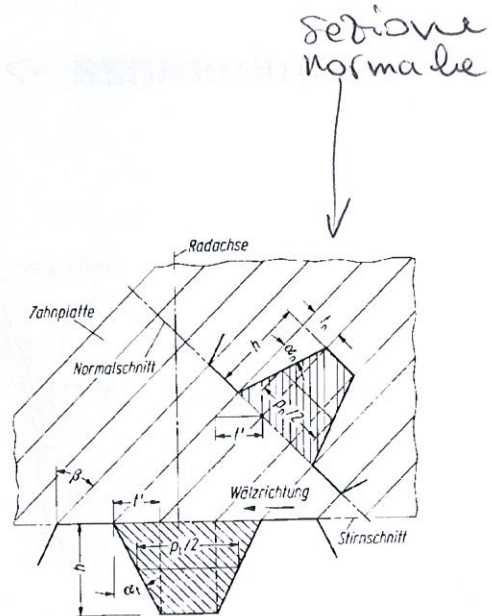
In direzione dello spessore si cambiano di $\cos(\beta)$:

$$m_t = m_n / \cos(\beta)$$

$$p_t = p_n / \cos(\beta)$$

$$\tan(\alpha_t) = \tan(\alpha_n) / \cos(\beta)$$

$$d = z * m_n / \cos(\beta)$$



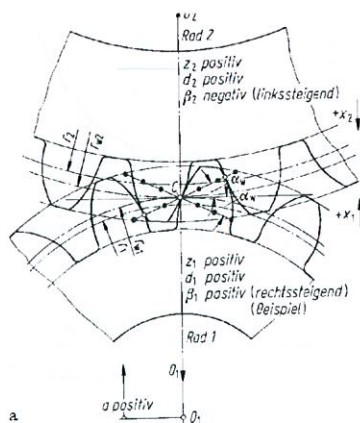
↑ sezione normale
↓ sezione apparente

ISO 21771 → prova normativa x ingr interni
dimensionamento

Dentatura interna

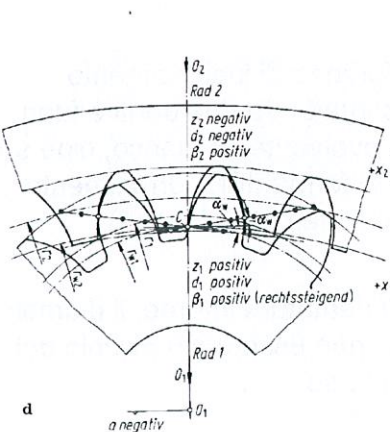
Esterno

- $z > 0$
- $a > 0$
- $d > 0$
- $d_a > 0$
- $d_f > 0$



Interno

- $z < 0$
- $a < 0$
- $d < 0$
- $d_a < 0$
- $d_f < 0$



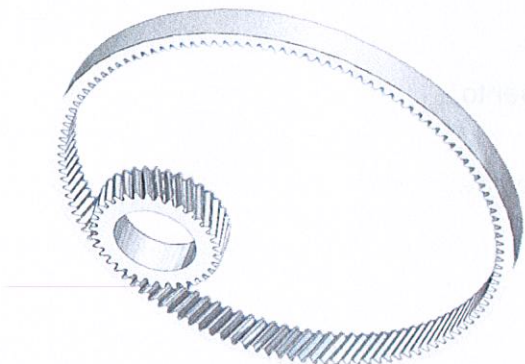
Dati negativi

Per dentature interne

l'interasse, i diametri ed il n° denti sono negativi.

II. Dentatura con gioco

Nella seconda parte viene presentata la dentatura con gioco.



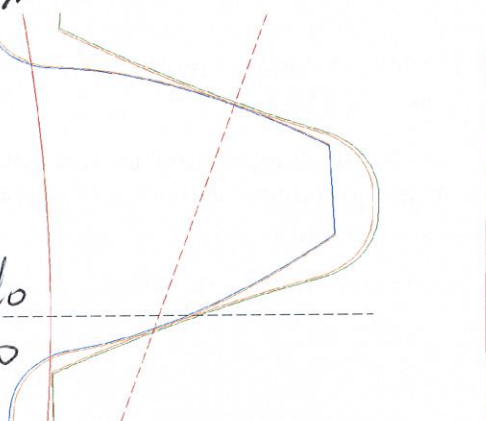
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Gioco di dentatura

Si ottiene gioco tra i fianchi aumentando la distanza tra gli assi o riducendo lo spessore del dente.

Metodo classico

NON STANDARD



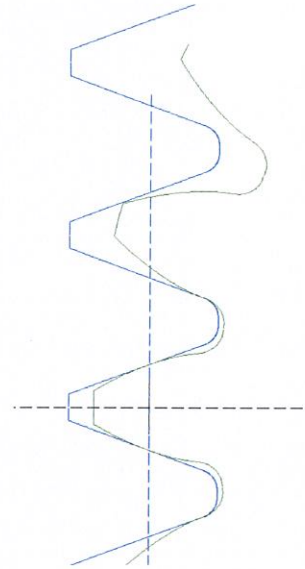
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Gioco di dentatura

Lo spessore dente viene ridotto avvicinando l'utensile all'ingranaggio.

Questa posizione si chiama spostamento di fabbricazione x_E .

Parallelamente anche il diametro di piede viene cambiato.



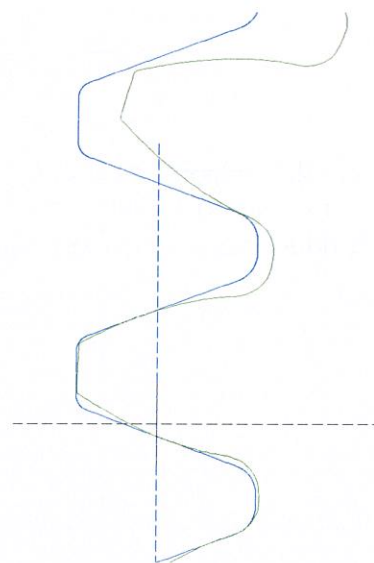
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Utensile topping

Un utensile topping taglia anche il diametro di testa.

Tramite l'avvicinarsi dell' utensile al ingranaggio anche il diametro di testa viene cambiato.

Diametro di testa e di piede sono in correlazione tramite l'utensile.



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Tolleranza spessore dente DIN 3967

Scelta di una classe di tolleranza secondo DIN 3967
(per ingranaggi con modulo > 0.5 mm):

- | | |
|--|--|
| • Casted rims | a29, a30 |
| • Rims (normal backlash) | a28 |
| • Rims (tight backlash) | bc26 |
| • Turbo gears (high temperatures) | ab25 |
| • Polymer machines | c25, cd25 |
| • Locomotive gear trains | cd25 |
| • Standard mechanical engineering, Heavy machinery, not reversing | b26 |
| • Standard mechanical engineering, Heavy machinery, reversing | c25, c24, cd25, cd24, d25, d24, e25, e24 |
| • Automobiles | d26 |
| • Agricultural vehicles | e27, e28 |
| • Machine tools | f24, f25 |
| • Printing machines | f24, g24 |
| • Measuring gearboxes | g22 |

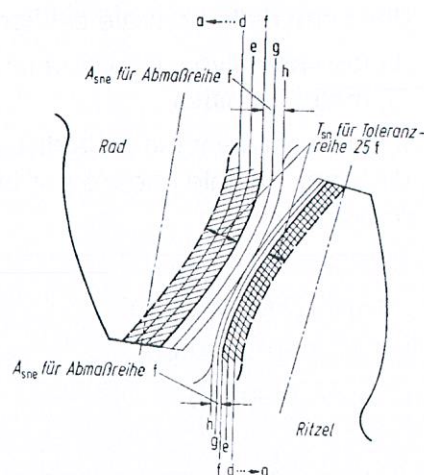
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Tolleranza spessore dente DIN 3967

Le lettere della sigla di tolleranza danno l'informazione per la tolleranza superiore.

,a' è per il gioco più grande, ,h' per gioco zero.

La cifra tra 21 e 30 indica la larghezza del intervallo di tolleranza.



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Tolleranza spessore dente 58405

Sceita di una classe di tolleranza secondo DIN 58405
(per piccoli ingranaggi);
Proposte secondo DIN 58405, parte 2:

| Material | Process | Tolerance of center distance | Tolerance of tooth thickness |
|--------------------------|------------------|------------------------------|------------------------------|
| Steel hardened | grinding | 5J | 5f |
| Steel heat treated | very fine milled | 6J | 6f |
| Light metal | fine milled | 7J | 7f |
| Light metal | fine milled | 8J | 8f |
| Steel/Laminated material | very fine milled | 6J | 6e |
| Steel/Laminated material | very fine milled | 7J | 7d/7c |
| Light metal | fine milled | 8J | 8d/8c |
| Polymers | milled | 9J | 9e/9d |
| Polymers | injection molded | 10J | 10e |

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Gioco tra i fianchi

Il gioco normale j_n viene misurato nella direzione della normale sul fianco.

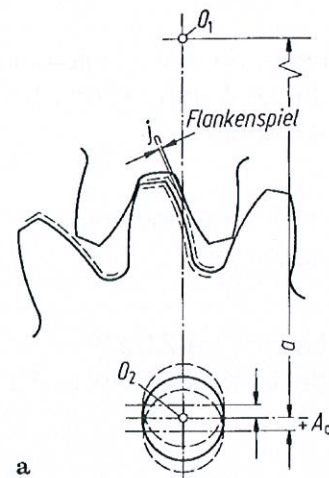
Il gioco primitivo j_t si misura invece sul diametro primitivo.

Il gioco radiale j_r indica la distanza in direzione assiale che c'è per far toccare le ruote nei fianchi

$$j_t = [(A_{sn1} + A_{sn2}) + 2 * A_a * \tan(\alpha_n)] / \cos(\beta)$$

$$j_n = j_t * \cos(\alpha_n) * \cos(\beta)$$

$$j_r = j_t / (2 * \tan(\alpha_{wt}))$$



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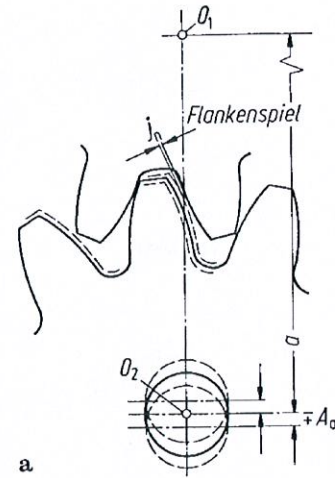
Tolleranze → giochi denti fianchi
 con tolleranze DIN 8595 → oppure
 scarto di spessore deducibile da
Waldarber.

Gioco di dentura al collaudo

Il gioco di dentatura al collaudo è il gioco misurato quando gli ingranaggi sono montati nel riduttore. Questo gioco è normalmente più piccolo del gioco teorico, perchè il non-parallelismo delle assi ed errori di fabbricazione riducono il gioco.

$$\Delta j_{\Sigma\beta} = -f_{\Sigma\beta} \cdot \frac{b}{L_G}$$

$$\Delta j_F = -\sqrt{\left(\frac{F_\beta}{\cos \alpha_t}\right)^2 + \left(\frac{F_f}{\cos \alpha_t}\right)^2} + f_p^2$$



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Trovi tolleranze x quota interasse nella sezione
 tolleranze → opzioni di visualizzazione
 se con tolleranze min o max o media

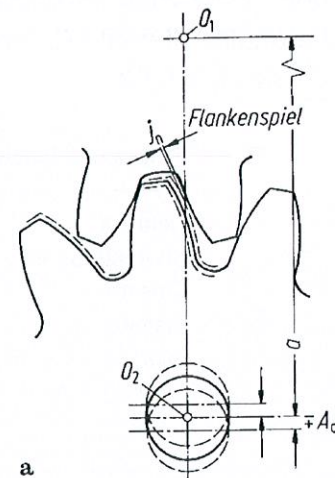
Il gioco al funzionamento

Il gioco di dentatura al funzionamento considera in più del gioco al collaudo:

- La temperatura (dilatazione ruote e carcassa)
- e, nel caso di ingr. sintetici, il rigonfiamento tramite assorbimento d'acqua.

$$\Delta j_g = a \cdot (\Delta \vartheta_G \cdot \alpha_G - \Delta \vartheta_R \cdot \alpha_R) \cdot 2 \cdot \frac{\tan \alpha_n}{\cos \beta}$$

$$\Delta j_Q = \left(\frac{1}{3} w \cdot a\right) \cdot 2 \cdot \frac{\tan \alpha_n}{\cos \beta}$$



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modifica n° denti controllo Wildhaber
in tollerante → n° denti misura

III. Tolleranze e misure di controllo



Nella terza parte vengono presentate le tolleranze di fabbricazione e il di controllo dello spessore dente.

Prof. r. peimento solumetallo → nessuno
X nitruv → NO RETTIFICA
X cement → ~~NO~~ RETTIFICA

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Qualità degli ingranaggi

La qualità degli ingranaggi secondo ISO 1328 è simile alla qualità secondo DIN 3961 o BS 436/ 2.

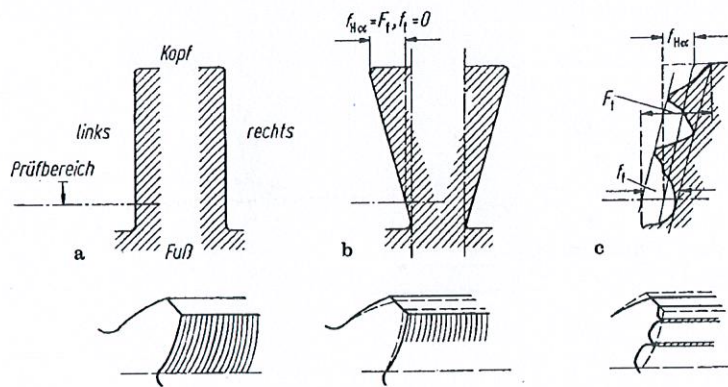
La qualità di dentatura raggiungibile dipende dal processo di fabbricazione:

| Processo di fabbricazione | Qualità |
|----------------------------|----------|
| Rettifica | 2...7 |
| Sbarbatura (Shaving) | 5...7 |
| Fresare | (5)6...9 |
| Piallare | (5)6...9 |
| Iniezione, Sinterizzazione | 8...12 |
| Fucinatura di precisione | 8...10 |

Qualità secondo AGMA 390.03 = 17-Qualità secondo ISO

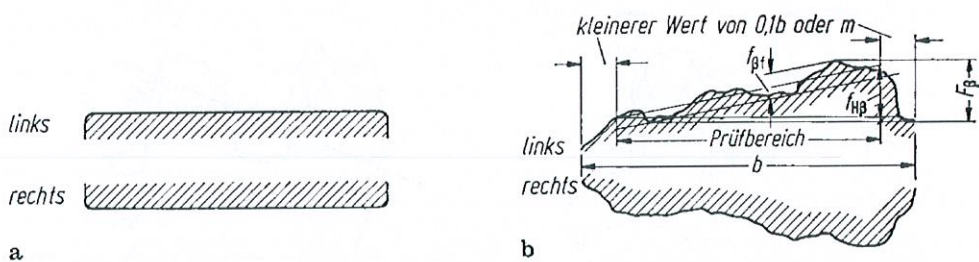
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Deviazione del profilo



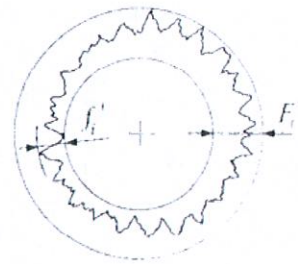
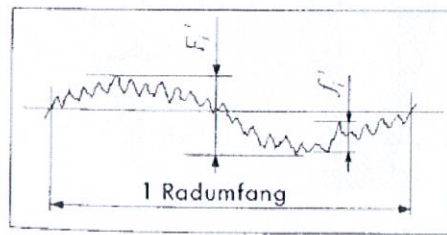
- F_f Deviazione totale profilo
- f_f Deviazione forma profilo
- $f_{H\alpha}$ Deviazione angolo profilo

Deviazione linea fianco



- F_β Deviazione linea fianco totale
- $f_{\beta f}$ Deviazione di forma della linea fianco
- $f_{H\beta}$ Deviazione dell'angolo linea fianco

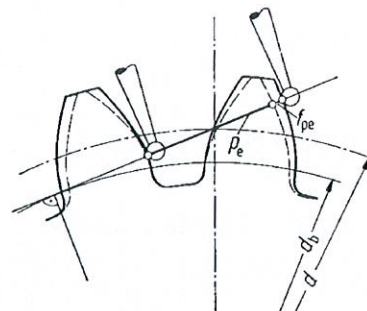
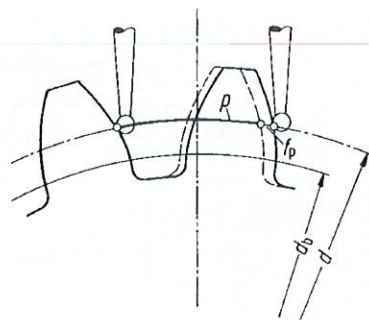
Errori di rotolamento



- F_i' Deviazione rotolamento fianco singolo
- f_i' Salto rotolamento fianco singolo
- F_i'' Errore composto su due fianchi
- f_i'' Salto composto su due fianchi

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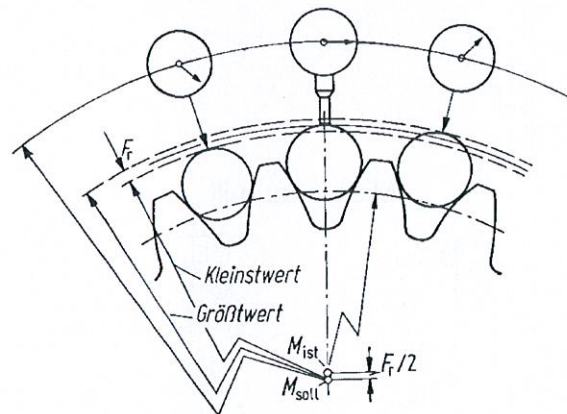
Errori di passo



- f_p Deviazione singolo del passo
- f_{pe} Deviazione del passo sulla linea d'ingranamento

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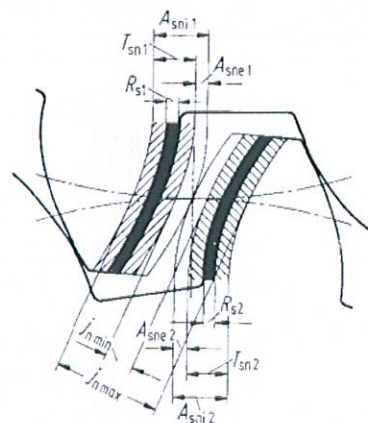
Deviazione concentricità



F_r Deviazione concentricità

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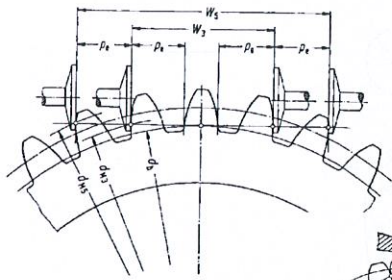
Intervallo dell'errore spessore dente



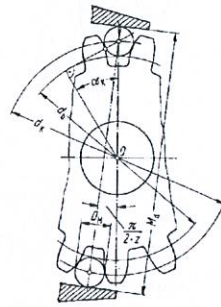
R_s Intervallo dell'errore sullo spessore dente

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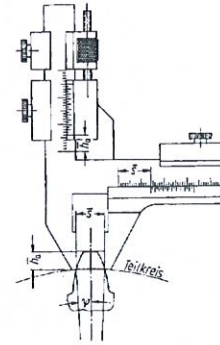
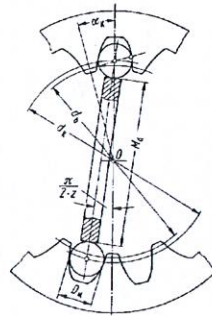
Controllo dello spessore dente



Misura della quota
Wildhaber
o scartamento)



Misura su rulli

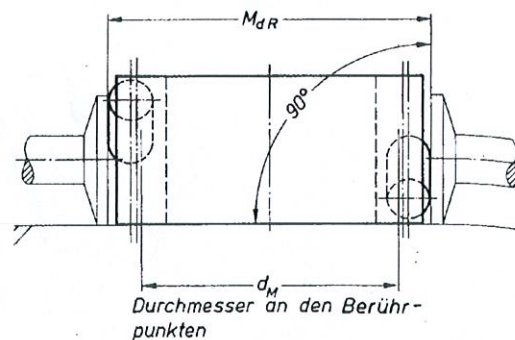
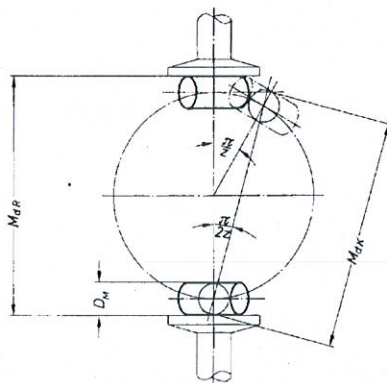


Misura dello
spessore



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Controllo del spessore dente



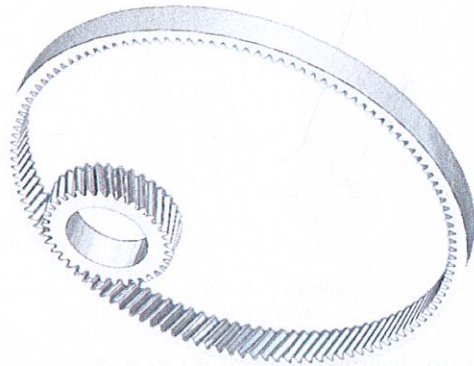
Per ingranaggi elicoidali con un no. denti dispari c'è una differenza tra la misura su rulli e su sfere.



www.KISSsoft.AG

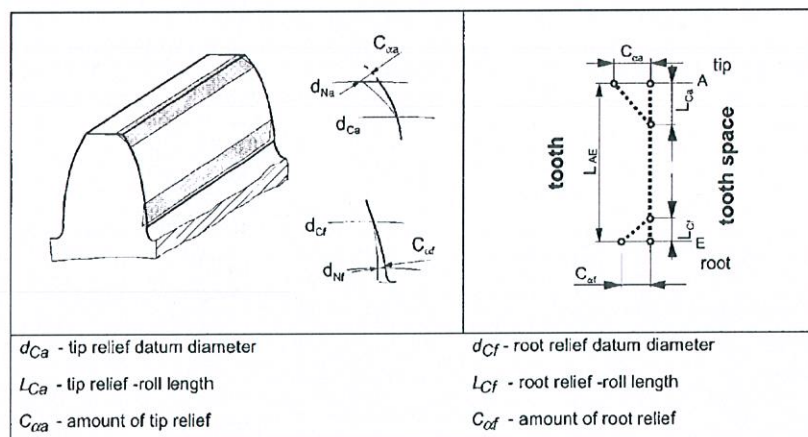
IV. Correzioni della dentatura

Nella quarta parte vengono presentate possibili correzioni della dentatura.



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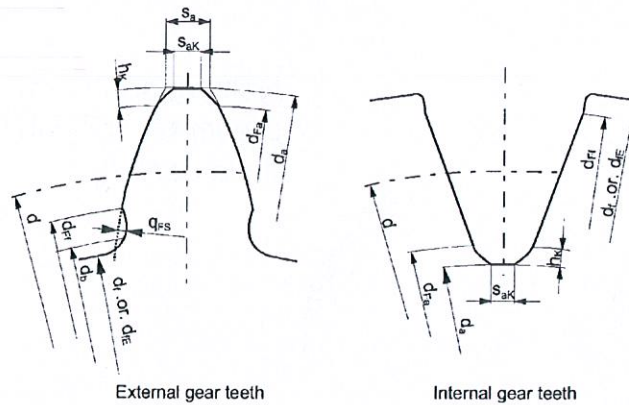
Correzione del profilo



Per evitare il 'urto di contatto' all'inizio o alla fine dell'ingranamento, si può fare una correzione del profilo alla testa o al piede del dente.

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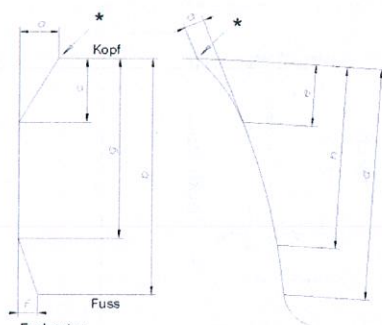
Correzione del profilo



La correzione (del profilo) di testa si può definire sul profilo di riferimento con una curvatura al piede, quello al piede con una protuberanza. Si può fare la correzione alla fresatura o alla rettifica.

Correzione del profilo

*CALCOLO
↓
CORREZIONI*



(Fig. Z-003/1)

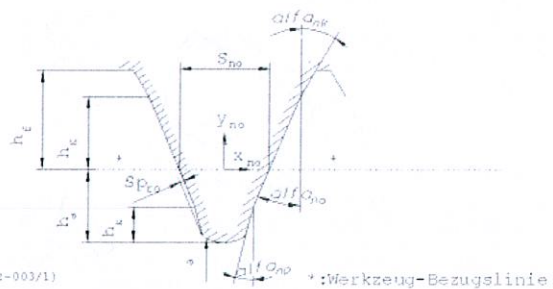


Diagramma del profilo

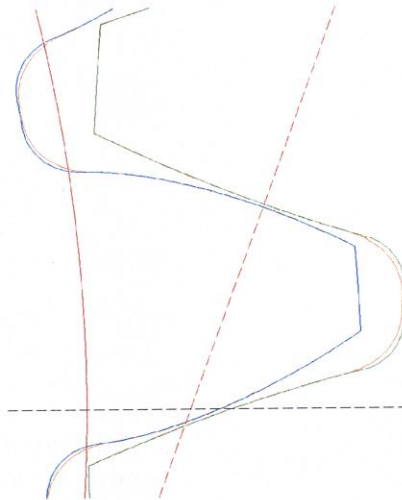
Profilo di riferimento dell'utensile con:

- Protuberanza
- Semi-topping (curvatura al piede)



Procedimento di Generazione spostata

Per ottenere altre forme di piede, il diametro di rotolamento di generazione può essere spostato. Qui il modulo è stato cambiato da 6 a 5.837 ed l'angolo di pressione da 20° a 15° (il diametro di base rimane uguale). Altezza testa e piede, spostamento del profilo, vanno cambiati in modo che anche lo spessore del dente rimanga identico.

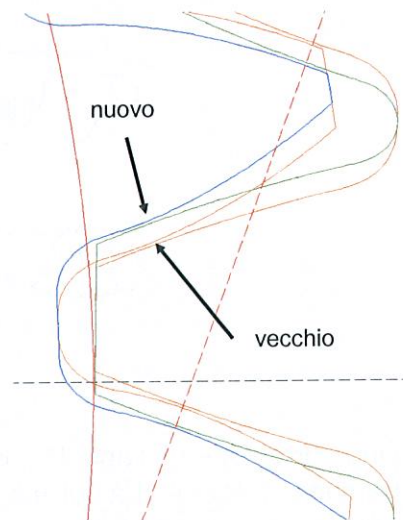


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Variazione spessore dente

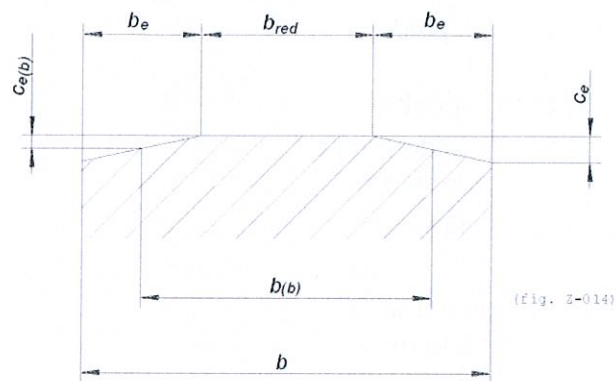
Nel caso di combinazione di materiali con resistenza molto differente, per esempio acciaio con sintetico, lo spessore del dente può essere variato.

Lo spostamento di profilo x viene cambiato, lasciando diametro di testa e di piede invariato, in modo che la ruota in acciaio diventa più sottile.



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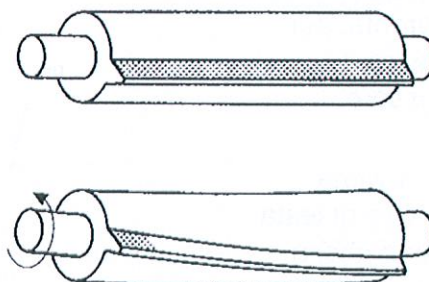
Bombatura / Spoglia d'estremità



Per ottenere una portata omogenea lungo la larghezza della dentatura, si può introdurre per esempio una bombatura. Così vengono compensati sia le deformazioni sia gli errori di fabbricazione.

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Correzione con l'angolo d'elica



Una correzione dell'angolo d'elica può compensare la deformazione tramite la torsione. La correzione può essere fatta in modo ideale soltanto per un momento torcente esatto.

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Special Problems of Gears

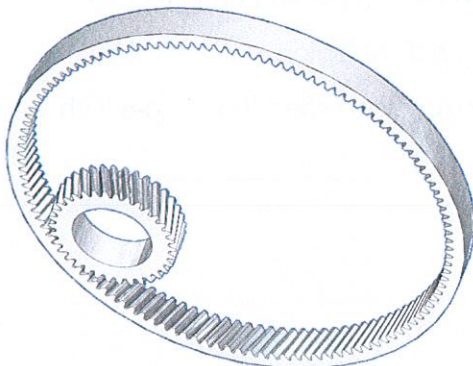
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Rosengartenstrasse 4
8608 Bubikon
Switzerland

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Fax: +41 55 254 20 51
info@KISSsoft.AG
www.KISSsoft.AG

SHARING KNOWLEDGE

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Required safety factor



How can we define the
required safety factor?

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Using new calculation method

- A calculation method like ISO6336 defines how to calculate safety factors but gives no value for necessary factors
- Proposal for safety factors for “normal” gears:

| | root | flank | scoring |
|------------------|------|-------|---------|
| module > 2 | 1.4 | 1.0 | 1.8 |
| module 1.0 - 2.0 | 1.2 | 0.9 | 1.8 |
| module 0.5 - 1.0 | 0.9 | 0.8 | - |
| module < 0.5 | 0.5 | 0.5 | - |

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Using new calculation method

- Select general factors including material values
- Make calculations with known gearboxes including critical ones
- Use the results to define necessary safety factors
- Use this factors to size new gearboxes and refine the factors with increasing experience

Calculation method + Experience (know how) = Optimal results

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Using new calculation method

| Given \ Resulting | Life time Nominal load | Safety factors Nominal load | Safety factors Lifetime |
|---|---|---|--|
| Using nominal load and application factor | Safety factors | Lifetime | Max. load |
| | Standard method in main window of KISSsoft | Lifetime calculation without load spectrum | Sizing of torque without load spectrum |
| Using load spectrum | Safety factors | Lifetime | Max. load |
| | Safeties with load spectrum | Lifetime calculation with load spectrum | Sizing of torque with load spectrum |

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Gear failure



What can we do, if gears are
damaged?

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Pitting problems

- Decrease hertzian stress (bigger radius)
 - Increase pressure angle
 - Smaller module, more teeth
- Increase permissible stress
 - Material with higher σ_{Hlim}
 - Higher quality
 - Tooth flank modification

Root strength problem

- Decrease root stress
 - Bigger module
 - Bigger radius at root (tip rounding of tool)
 - Bigger helix angle
- Increase permissible stress
 - Material with higher σ_{Flim}
 - Higher quality
 - Tooth flank modification

GRIPPAGGIO

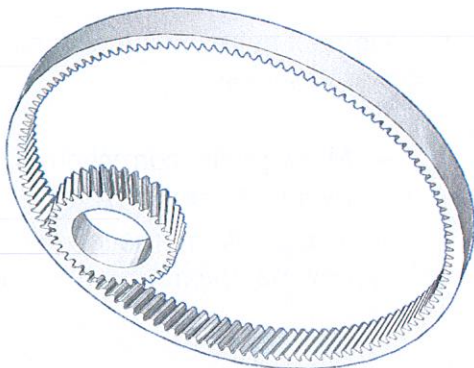
Scuffing problem

- Decrease pressure at tip
 - Profile modification
- Decrease sliding speed
 - Optimised profile shift (addendum modification)
- Better lubrication
 - Oil with better test results against scuffing
 - Oil with higher viscosity
 - Lower oil temperature

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Noise and vibration

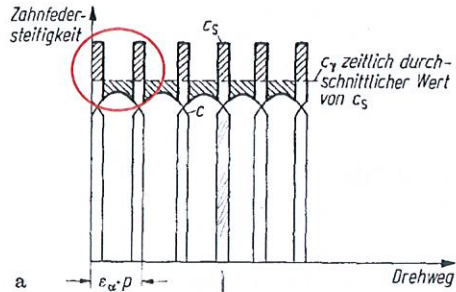


What are the sources of noise and vibrations and what can we do to avoid it?

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Decrease stiffness variation



Periodo con 2 denti contatto

- Variation in tooth stiffness generates vibrations

- Try to achieve a contact ratio as whole number
- The best would be a whole number for transverse and overlap contact ratio
- Else try to get the total contact ratio as whole number

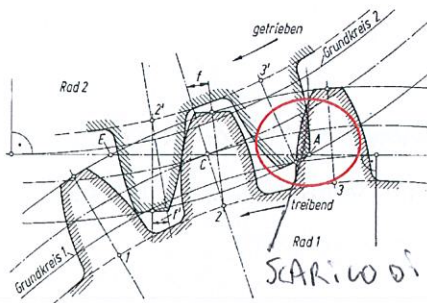
Zahnfedersteifigkeit: Tooth pair stiffness, Drehweg: Rotation, zeitlich durchschnittlicher Wert von c_s : time average value of c_s

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URTO di CONTATTO

Avoid impact at start of mesh



- An impact at beginning of contact generates noise

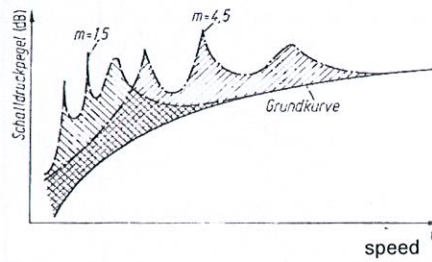
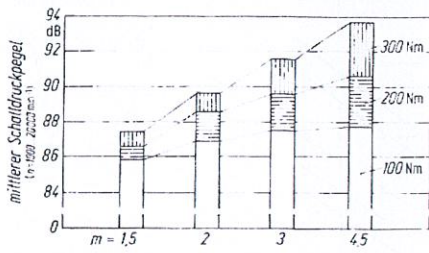
- Make profile correction to avoid the impact
- The profile correction can only be optimal for one load

Rad: Gear, Grundkreis: Base diameter, Getrieben: Driven, Treibend: Drive

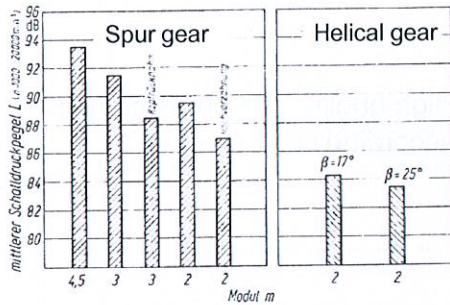
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Noise vs. Module and helix angle

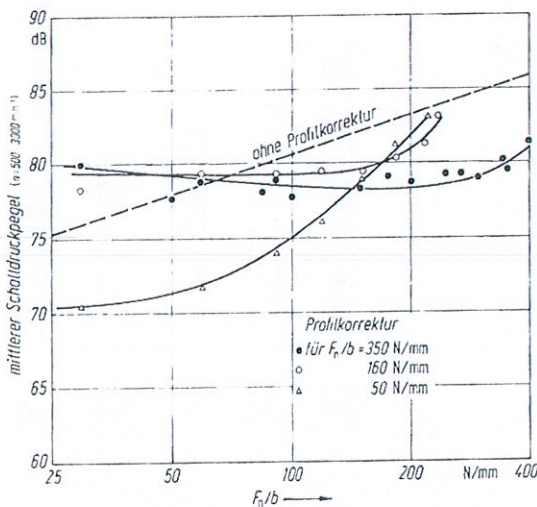


Noise will increase with the module ($\beta=0$, $a=91.5\text{mm}$) according Niemann



Helical gears will also improve noise level according Niemann.

Noise vs. Profile modification

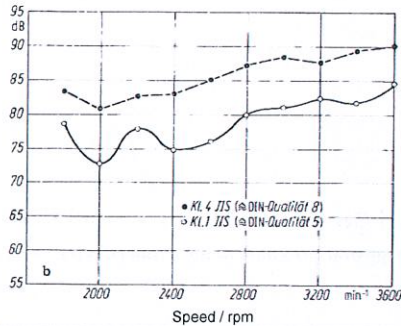
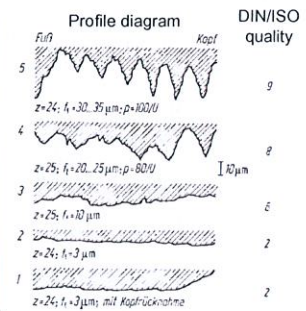
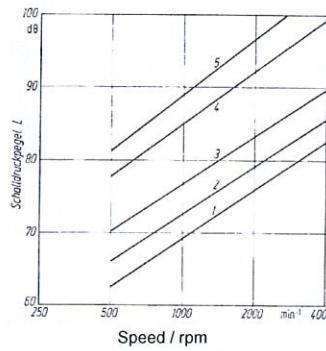
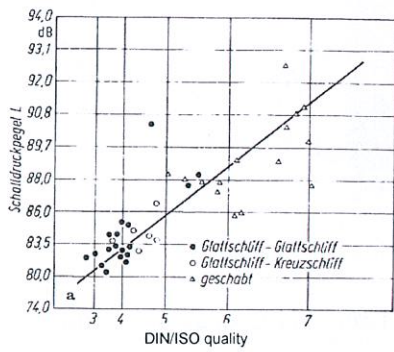


Profile modification is reducing noise according Niemann.

$z_1=z_2=25$, $\beta=0$

mittlerer Schalldruckpegel: medium sound pressure level, ohne: without, Profilkorrektur: Profile modification

Noise vs. Gear quality

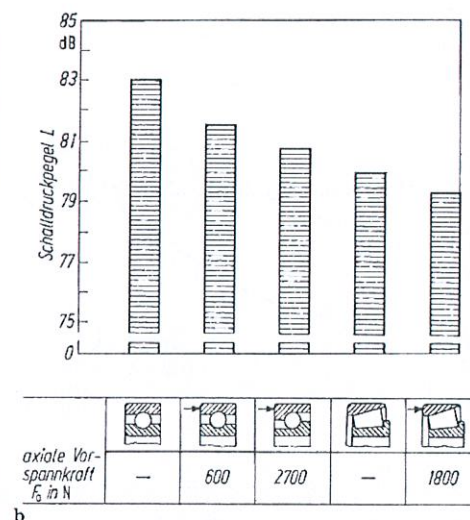
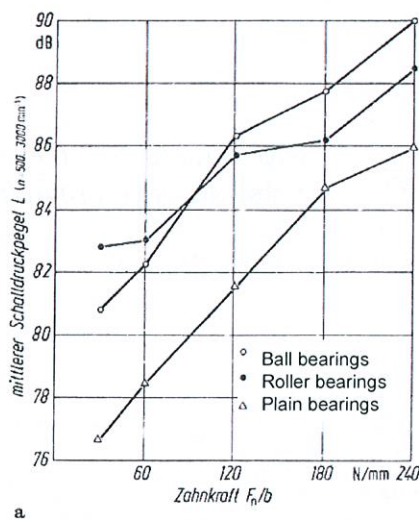


High quality is decreasing noise according Niemann.

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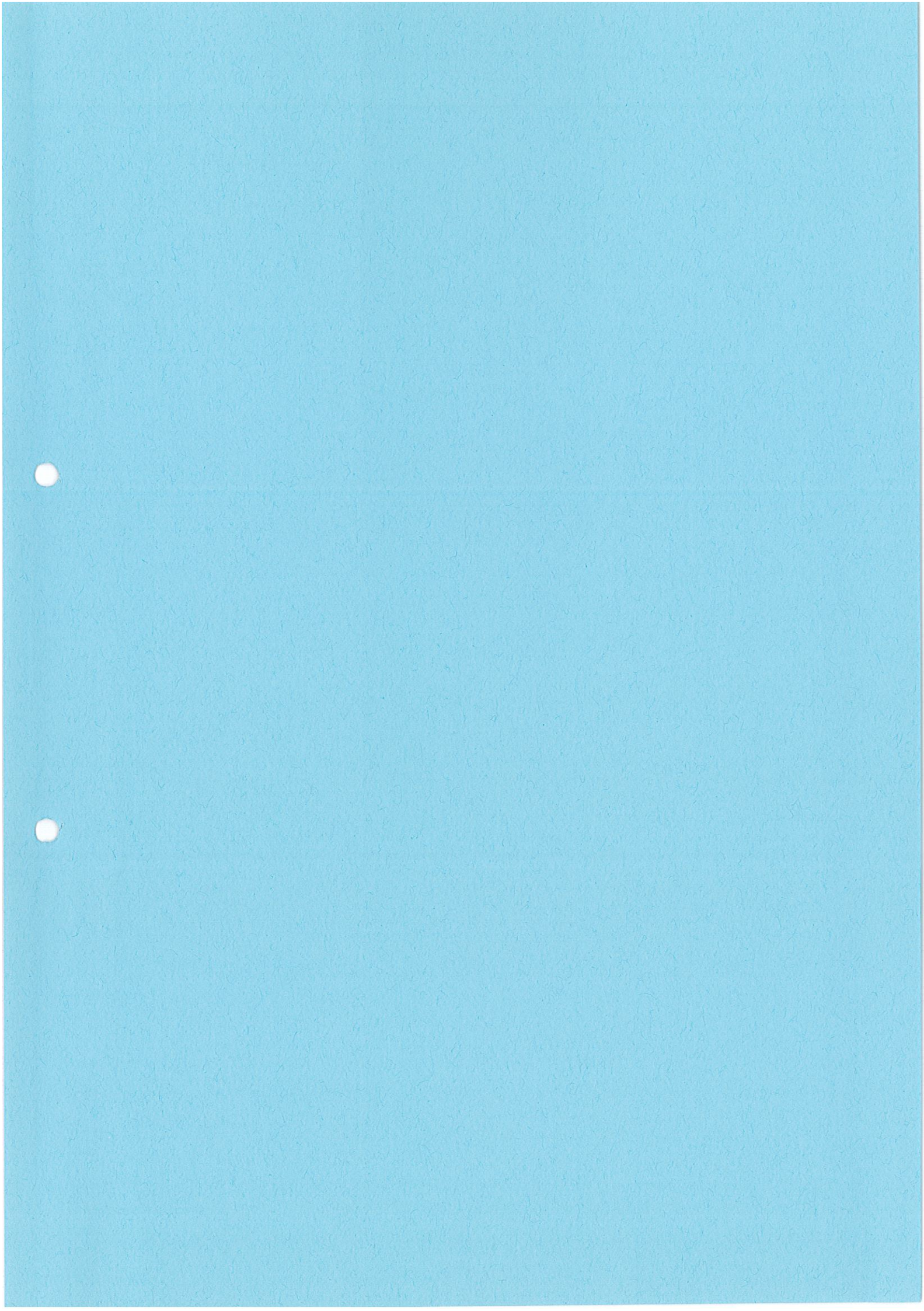
Noise vs. Bearings



The type of bearings and axial pretension has an influence on noise according Niemann

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Short introduction

Layout of flank line modifications using the $K_{H\beta}$ calculation method according Annex E of ISO 6336-1

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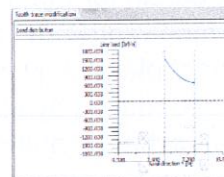
Ulrich Kissling

DMK 2015

Introduction

The face load factor $K_{H\beta}$, which in rating equations represents the load distribution over the common face width in meshing gears, is one of the most important items for a gear strength calculation.

$$K_{H\beta} = \frac{w_{\max}}{w_m} = \frac{\text{maximum load per unit face width}}{\text{average load per unit face width}}$$



In the international standard for cylindrical gear rating, the ISO6336-1, using method C, some formulas are proposed to get a value for this factor. But as the formulas are simplified, the result is often not very realistic.

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Ulrich Kissling

DMK 2015

www.KISSsoft.AG

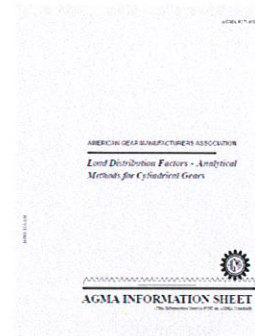
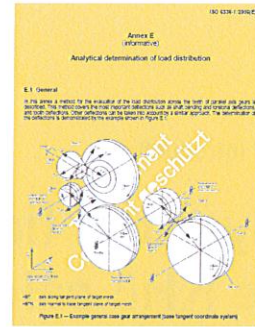
Introduction

In the actual edition (2006 version) of the ISO6336 standard for the load capacity calculation of spur and helical gears a new annex E was added:

"Analytical determination of load distribution".

This annex is entirely based on the AGMA 927-A01 standard.

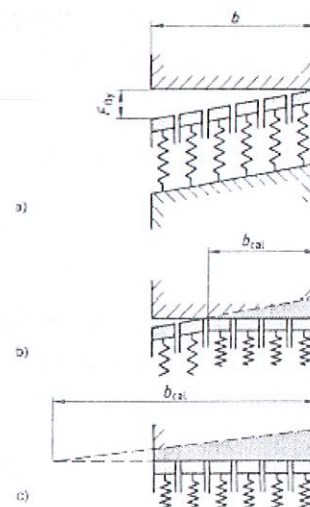
It is a well documented procedure to get a direct and precise number for the face load factor.



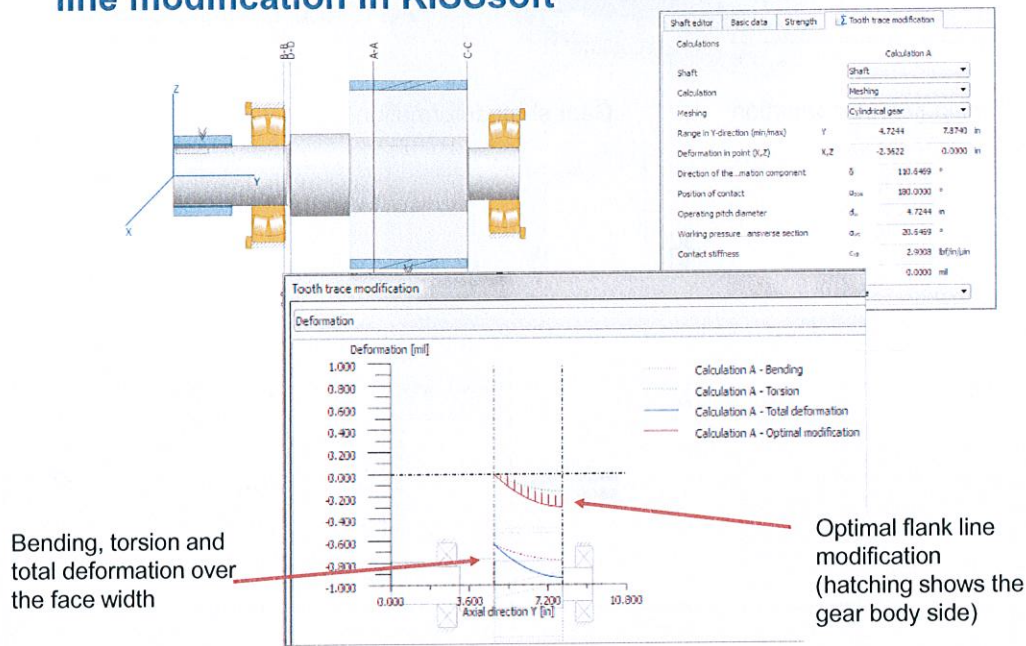
Theory: Determination of the load distribution over the face width

The cause for the uneven load distribution over the face width are flank line deviations in the contact plane of two gears. Deviations are caused mainly by elastic deformations of the shaft, stiffness and clearance of bearings and housing, manufacturing tolerances and thermal deformations.

The determination of the load distribution is – as documented in the gear theory – performed in two steps. At first the gap in the tooth contact is calculated. Then, using the tooth mesh stiffness ($c_{\gamma\beta}$), the line load distribution is determined. This approach is well documented in ISO6336-1.



Display of the gap and proposition for an optimum flank line modification in KISSsoft



Slide 5
09.02.2016

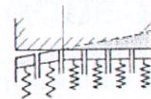
Ulrich Kissling

DMK 2015

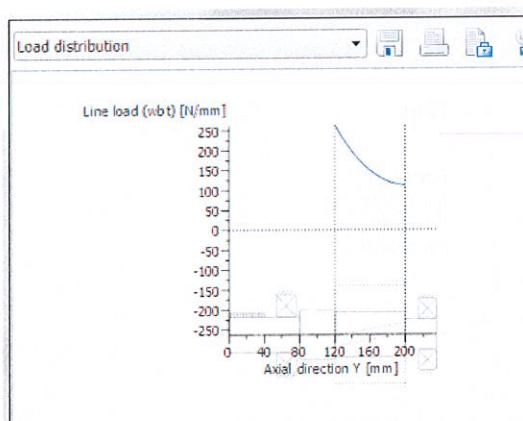
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Load distribution in the tooth contact and face load factor $K_{H\beta}$

$$F_t = \sum_i F_{ti} (\delta_i * c_{g\beta})$$



$$K_{H\beta} = \frac{w_{\max}}{w_m} = \frac{\text{maximum load per unit face width}}{\text{average load per unit face width}}$$



| Results | | |
|------------------------------------|-------------|----------------|
| Medium deformation - Calculation A | | |
| wmax | wm | wmax/wm (=KHβ) |
| 262.95 N/mm | 162.70 N/mm | 1.6161 |

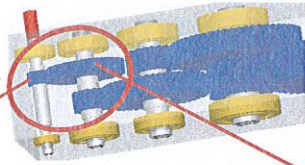
Load distribution and numbers for the maximum and mean line load and $K_{H\beta}$

Slide 6
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Ulrich Kissling

DMK 2015

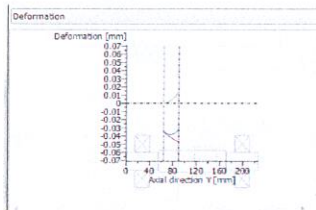
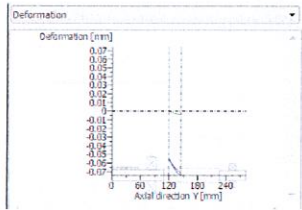
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This is the ISO6336-1, Annex E, proceeding.

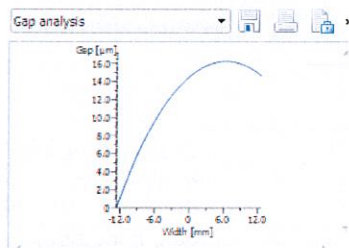
Pinion shaft deformation

Gear shaft deformation

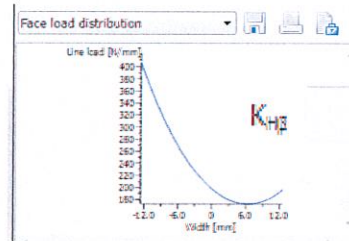


+

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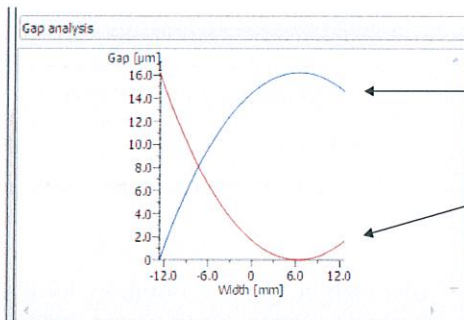
⇒



Layout and optimization of flank line modifications

Flank line modifications for **nominal torque** (no duty cycles)

The optimum flank line modification is just the inversion of the gap!



Gap

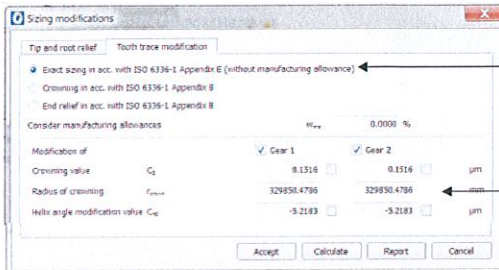
Proposition
for optimum
flank line
modification

fma = 0 µm / Rfb = 0 µm:
wmax = 407.1366 N/mm wm = 233.2786 N/mm KHbeta = 1.7453

Layout and optimization of flank line modifications

Flank line modifications for **nominal torque** (no duty cycles)

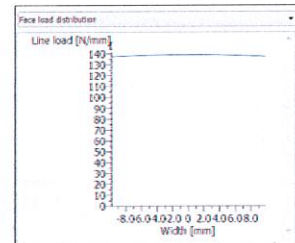
So providing a sizing function for the optimum flank line modification is easy...



Sizing function

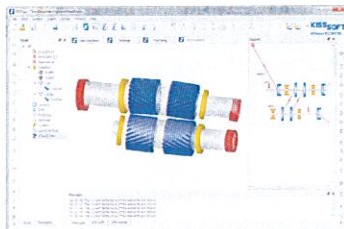
Proposition for optimum flank line modification.

Result

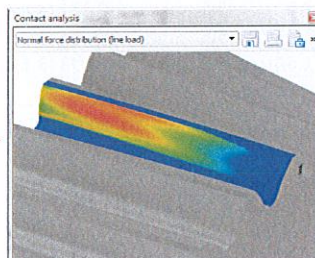


Comparison with TCA (Tooth Contact Analysis)

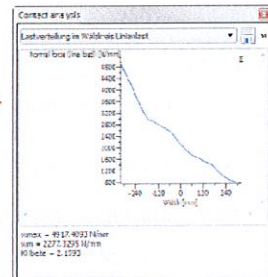
To be able to compare Annex E results with TCA, we added a new graphic, extracting from TCA results the line load on the operating pitch diameter.



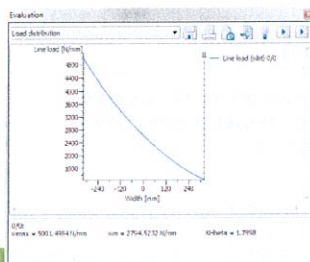
From this double helical gear box



We get this line load distribution (left side of double helical)



Line load on the operating pitch diameter

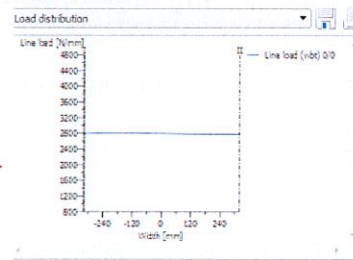


< Here the result
< with Annex E !

Comparison with TCA (Tooth Contact Analysis)

Here we determined optimum flank line modification of the left side of the double helical using Annex E, and compared with TCA.

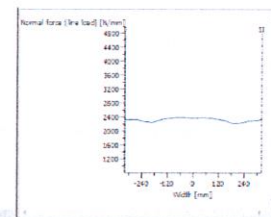
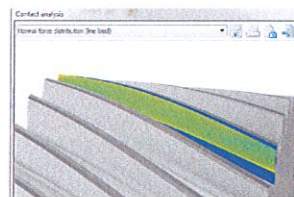
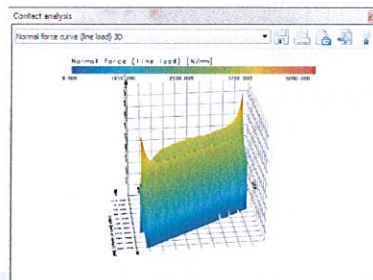
| Gear | Flank | Type of modification | Value [μm] | Factor 1 |
|--------|-------|---|-------------------------|----------|
| Gear 1 | both | Tip relief, linear with transition radius | 112.0000 | 0.1170 |
| Gear 1 | both | Helix angle modification, parallel | -284.0000 | |
| Gear 1 | both | Crowning | 23.0000 | |
| Gear 2 | both | Tip relief, linear with transition radius | 112.0000 | 0.1170 |
| Gear 2 | both | Helix angle modification, parallel | -58.0000 | |
| Gear 2 | both | Crowning | 21.0000 | |



Flank line modifications found with Annex E

Line load with Annex E

Line load found with TCA:



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Flank line modifications for applications with duty cycles

| Element (Load case) no. | Frequency | Load on a pinion (kNm) | Speed (1/min) | Radial mesh misalignment f_{E6} [mm] | Tangential mesh misalignment f_{E8} [mm] |
|--------------------------|-----------|------------------------|---------------|--|--|
| 1 | 0.980097 | 33.5 | 5.5 | +/-0.143 (5.6 mil) | +/-0.183 (7.2 mil) |
| 2 | 0.019602 | 67.0 | 5.5 | +/-0.121 (4.8 mil) | +/-0.411 (16.2 mil) |
| 3 | 0.000294 | 111.6 | 5.5 | +/-0.084 (3.3 mil) | +/-0.686 (27.0 mil) |
| 4 | 0.000007 | 111.6 | 5.5 | +/-0.078 (3.1 mil) | +/-0.754 (29.7 mil) |
| Total requested lifetime | | | 32'000 hours | | |

Duty cycle with axis misalignment of a ship steering module

| Load case | $K_{H\beta}$ (AGMA927) | Bending safety factor SF | Pitting safety factor SH |
|-----------|------------------------|--------------------------|--------------------------|
| 1 | 2.22 | 2.96 | 1.38 |
| 2 | 2.23 | 2.73 | 1.27 |
| 3 | 2.28 | 2.78 | 0.97 |
| 4 | 2.40 | 2.80 | 0.94 |

$K_{H\beta}$, SF, SH calculated individually for each load case with actual data and flank line modification

| Load case | Damage (%) |
|-----------|------------|
| 1 | 0.00 |
| 2 | 71.93 |
| 3 | 27.19 |
| 4 | 0.88 |

Damage (%) of the different load cases, when analyzing the overall lifetime using Palmgren-Miner rule

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Finding the best crowning for a ship steering drive with extreme duty cycle

Modification A

| Gear | Type of modification | Value (min) [µm] | Fact | Fact | Value (max) [µm] | Factor |
|--------|----------------------|------------------|------|------|------------------|--------|
| Gear 1 | Crowning | 10.0000 | | | 400.0000 | |

| ID | Wt. [%] | H _{max} [N] | K _{sp} | Value [µm] |
|------|---------|----------------------|-----------------|------------|
| 11-- | 100.000 | 0.000 | 2.778 | 0.000 |
| 11-- | 100.000 | 0.000 | 2.744 | 10.000 |
| 11-- | 100.000 | 0.000 | 2.739 | 20.000 |
| 11-- | 100.000 | 0.000 | 2.674 | 30.000 |
| 4-- | 100.000 | 36300.000 | 2.640 | 40.000 |
| 5-- | 100.000 | 18200.000 | 2.465 | 50.000 |
| 6-- | 100.000 | 18900.000 | 2.570 | 60.000 |
| 7-- | 100.000 | 21400.000 | 2.535 | 70.000 |
| 8-- | 100.000 | 21200.000 | 2.501 | 80.000 |
| 9-- | 100.000 | 25400.000 | 2.466 | 90.000 |
| 10-- | 100.000 | 27800.000 | 2.431 | 100.000 |
| 11-- | 100.000 | 30200.000 | 2.397 | 110.000 |
| 12-- | 100.000 | 32900.000 | 2.362 | 120.000 |
| 13-- | 100.000 | 36000.000 | 2.327 | 130.000 |
| 14-- | 100.000 | 39500.000 | 2.291 | 140.000 |
| 15-- | 100.000 | 43300.000 | 2.257 | 150.000 |
| 16-- | 100.000 | 47500.000 | 2.222 | 160.000 |
| 17-- | 100.000 | 52000.000 | 2.187 | 170.000 |
| 18-- | 100.000 | 57000.000 | 2.153 | 180.000 |
| 19-- | 100.000 | 62400.000 | 2.118 | 190.000 |
| 20-- | 100.000 | 70200.000 | 2.083 | 200.000 |
| 21-- | 100.000 | 77500.000 | 2.048 | 210.000 |
| 22-- | 100.000 | 85200.000 | 2.012 | 220.000 |
| 23-- | 100.000 | 92300.000 | 1.976 | 230.000 |
| 24-- | 100.000 | 99700.000 | 1.940 | 240.000 |
| 25-- | 100.000 | 106000.000 | 1.904 | 250.000 |
| 26-- | 100.000 | 114000.000 | 1.867 | 260.000 |
| 27-- | 100.000 | 119000.000 | 1.830 | 270.000 |
| 28-- | 100.000 | 125000.000 | 1.793 | 280.000 |
| 29-- | 100.000 | 130000.000 | 1.756 | 290.000 |
| 30-- | 100.000 | 135000.000 | 1.719 | 300.000 |
| 31-- | 100.000 | 140000.000 | 1.682 | 310.000 |
| 32-- | 100.000 | 145000.000 | 1.645 | 320.000 |
| 33-- | 100.000 | 150000.000 | 1.608 | 330.000 |
| 34-- | 100.000 | 155000.000 | 1.571 | 340.000 |
| 35-- | 100.000 | 160000.000 | 1.534 | 350.000 |
| 36-- | 100.000 | 165000.000 | 1.497 | 360.000 |
| 37-- | 100.000 | 170000.000 | 1.460 | 370.000 |
| 38-- | 100.000 | 175000.000 | 1.423 | 380.000 |
| 39-- | 100.000 | 180000.000 | 1.386 | 390.000 |
| 40-- | 100.000 | 185000.000 | 1.349 | 400.000 |

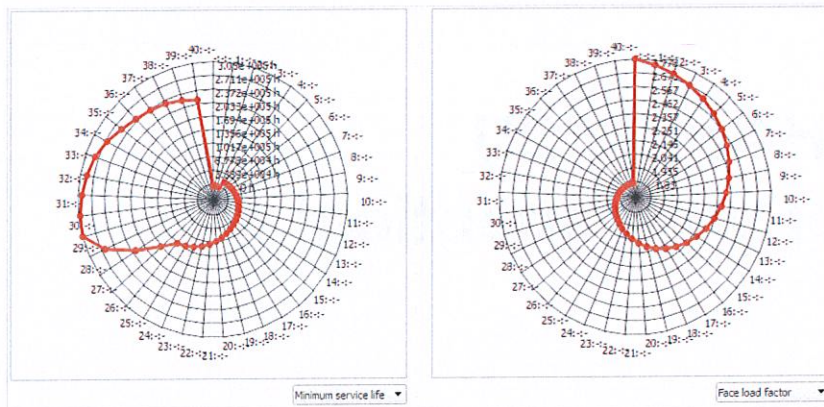
Table of the results:
 The column 'Value' displays crowning value C_b in µm
 ID: Indicates the modification variant (-:- for No modification at all; 1:- for the first variant [$C_b=10$]; 2:- for the second variant [$C_b=20$])

Optimization Parameters (only crowning with $C_b=10 \dots 400$; step of 10 µm)

Result: Lifetime of actual solution 29'000 hours; with variant 29:- 305'000 hours

Finding the best crowning for a ship steering drive with extreme duty cycle

Graphical display of the results



Graphical representation of the optimization results:

Life time with duty cycle
 For variant -:- to 40:-
 Scale from 0 to 305'000 hours life time

$K_{H\beta}$ of load case 4, based on Annex E
 For variant -:- to 40:-
 Scale from 1.85 to 2.78

Conclusion

Annex E in ISO6336, "Analytical determination of load distribution" is entirely based on the AGMA 927-A01 standard. It is a very useful method to get a realistic value for the face load factor $K_{H\beta}$, and much faster than using contact analysis.

Basically the algorithm is a one-dimensional contact analysis, providing good information about the load distribution over the face width.

The procedure is very helpful, when for a gear pair the optimum flank line modifications have to be found.

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KISSSOFT
Calculation programs for machine design

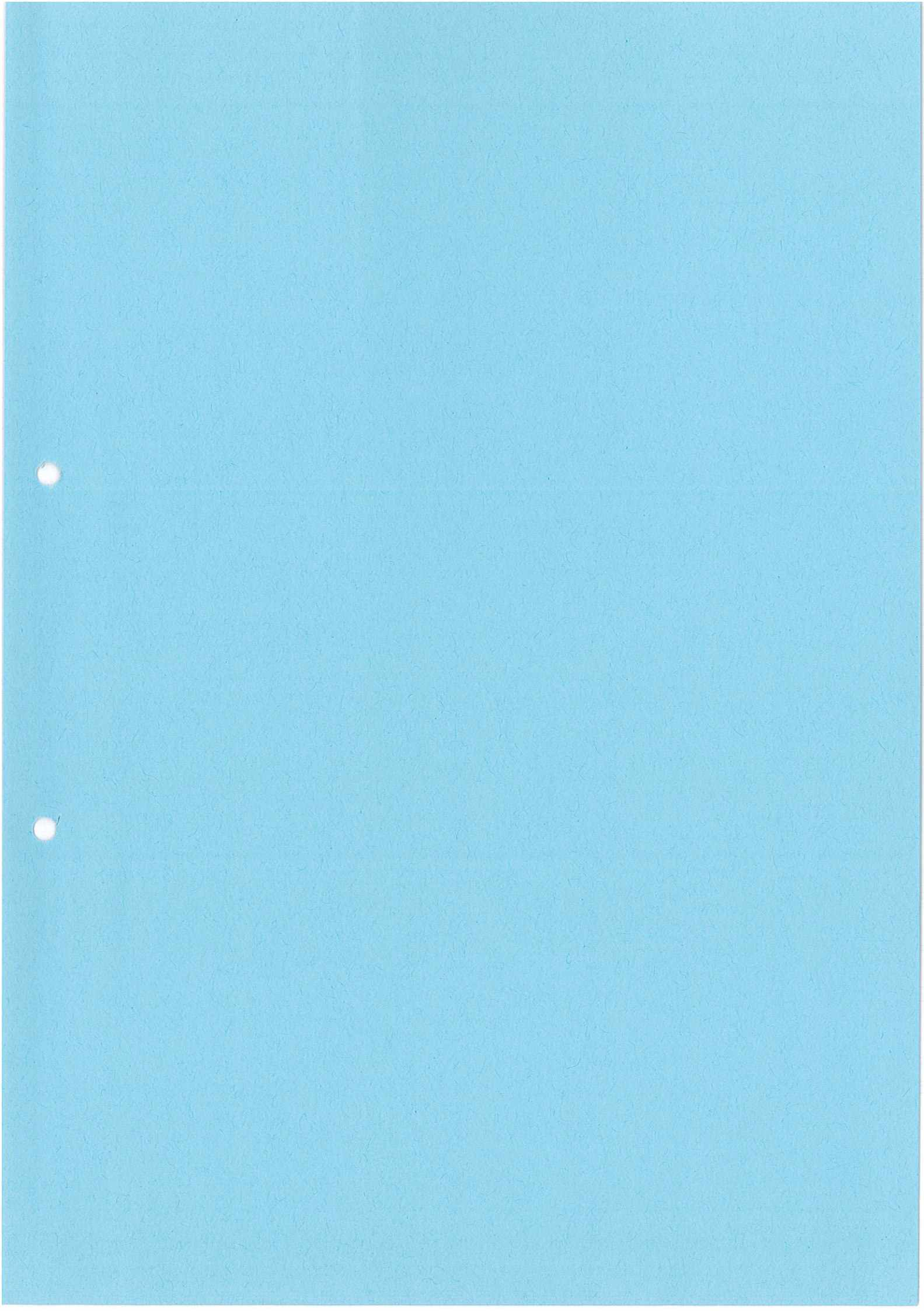
THANK YOU
for your attention!

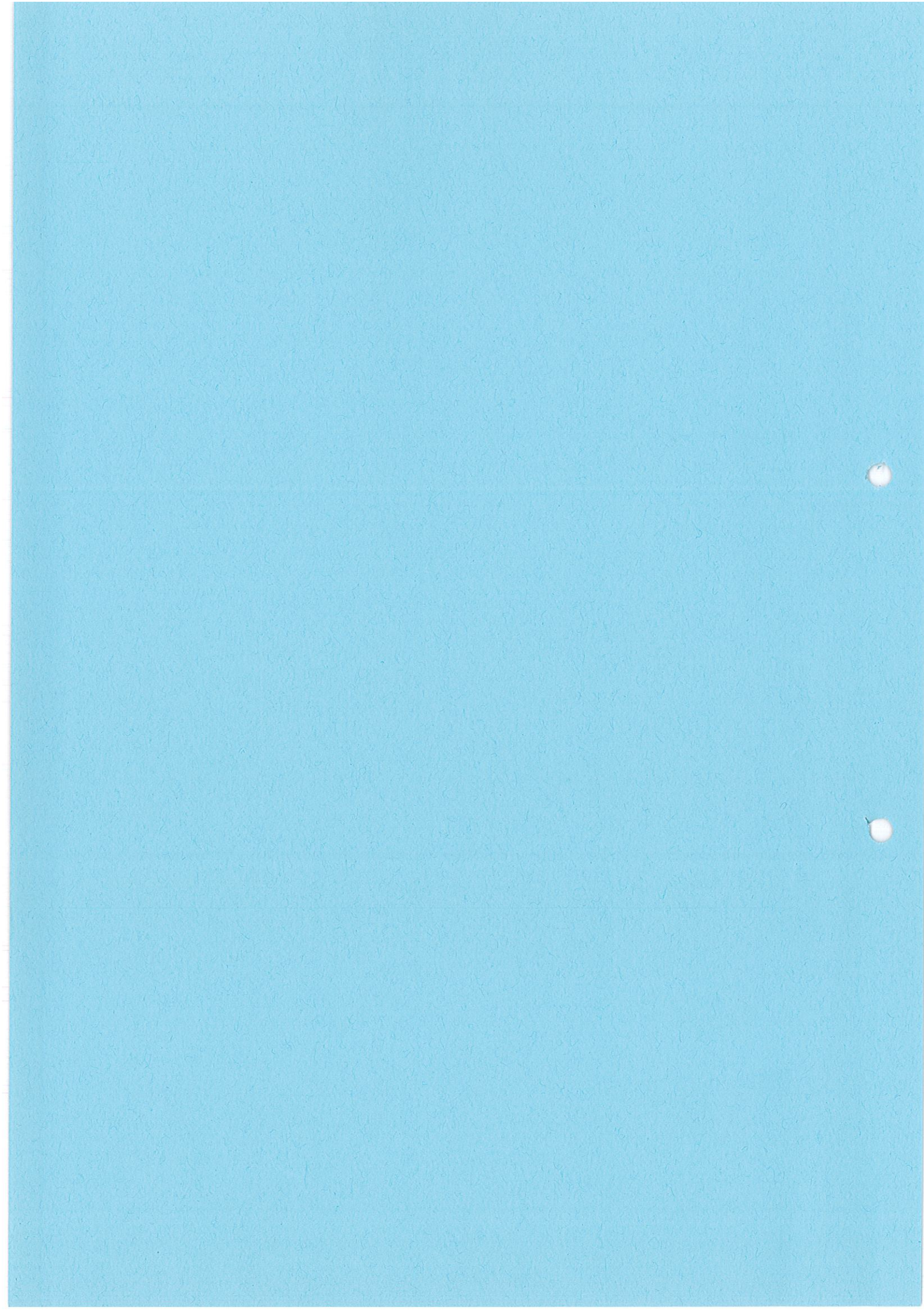
SHARING KNOWLEDGE

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Progettazione grossolana

KISSsoft
Calculation programs for machine design

→ calcolo → ti ritorna varie option

che puoi ordinare secondo molte varianti
tra le quali il rapporto coppia/peso !!

Sizing the gear micro geometry – Step by Step

Rapporti senza
multiplici in comune?

Tasto ~~dx~~
x opzioni !!

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SHARING KNOWLEDGE

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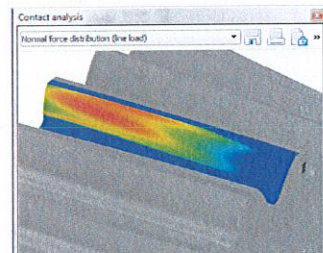
Layout of the Micro Geometry

The last phase in sizing a gear pair is to specify the flank line and profile modifications (also known as the "micro geometry").

To do so, the primary objective for which optimization has to be achieved: noise, service life, scuffing, micropitting or efficiency must be selected.

The calculation method for proving the effects achieved by micro geometry, the contact analysis under load ("Loaded Tooth Contact Analysis", or LTCA), is complex and time-consuming.

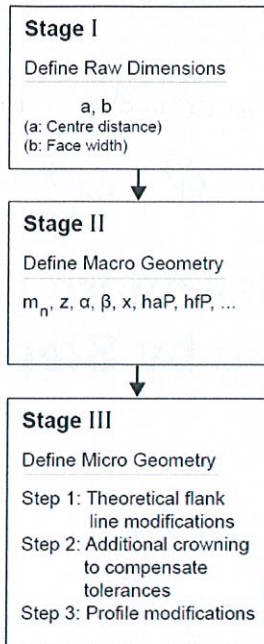
Unfortunately, the interpretation of LTCA results is not easy. All modifications applied on mating gears are interacting, so the decision of which modification to add or to change is difficult.



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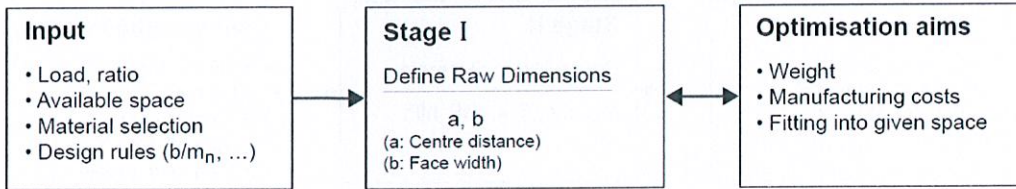
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Layout process of a gear set



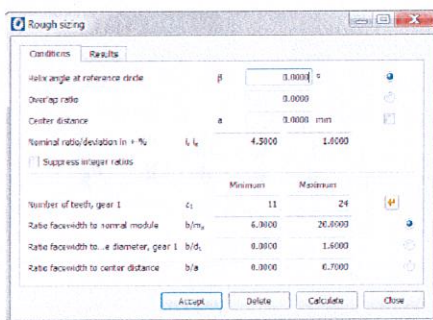
Stage I

Layout process of a gear set: Stage I



Lay out of: Raw dimensions

Rough sizing



Rough sizing

Conditions Results

| a [mm] | b ₁ [mm] | m _n [mm] | T _{1, max} [Nm] | W [kg] |
|---------|---------------------|---------------------|--------------------------|--------|
| 220.000 | 24.447 | 5.000 | 1625.222 | 19.500 |
| 262.000 | 58.985 | 4.000 | 1625.222 | 19.743 |
| 228.449 | 34.770 | 3.828 | 1628.848 | 20.284 |
| 239.151 | 73.880 | 3.654 | 1625.428 | 20.468 |
| 314.000 | 41.585 | 3.000 | 1625.219 | 20.492 |
| 301.000 | 46.117 | 4.500 | 1632.708 | 20.755 |
| 289.000 | 58.999 | 4.500 | 1625.344 | 20.761 |
| 301.000 | 49.494 | 4.500 | 1630.579 | 20.965 |
| 262.000 | 50.000 | 4.500 | 1625.205 | 21.130 |
| 339.000 | 48.169 | 5.000 | 1622.110 | 21.153 |
| 262.000 | 63.363 | 4.000 | 1629.218 | 21.214 |
| 301.000 | 48.928 | 4.500 | 1630.441 | 21.221 |
| 275.000 | 55.827 | 4.500 | 1623.264 | 21.468 |
| 301.000 | 46.550 | 5.000 | 1625.225 | 21.979 |
| 250.000 | 68.843 | 4.000 | 1626.471 | 22.217 |
| 289.000 | 49.358 | 5.000 | 1625.400 | 22.235 |
| 125.000 | 48.835 | 5.000 | 1625.273 | 22.448 |
| 262.000 | 61.204 | 4.500 | 1632.204 | 22.667 |
| 262.000 | 61.154 | 4.500 | 1632.275 | 22.770 |
| 237.000 | 78.432 | 4.000 | 1623.318 | 23.420 |
| 275.000 | 55.100 | 5.000 | 1623.340 | 23.667 |
| 314.000 | 48.291 | 5.000 | 1623.815 | 23.790 |
| 314.000 | 44.400 | 5.000 | 1622.649 | 23.801 |
| 250.000 | 67.756 | 4.500 | 1628.897 | 23.979 |
| 301.000 | 47.134 | 5.000 | 1623.176 | 24.200 |
| 289.000 | 55.667 | 5.000 | 1631.885 | 24.220 |
| 205.000 | 58.195 | 5.000 | 1623.112 | 24.963 |
| 262.000 | 61.205 | 5.000 | 1632.836 | 24.995 |
| 262.000 | 51.436 | 5.000 | 1625.233 | 25.253 |
| 237.000 | 73.333 | 4.500 | 1628.470 | 25.374 |
| 237.000 | 73.928 | 4.500 | 1628.423 | 25.472 |
| 262.000 | 62.397 | 5.000 | 1630.240 | 25.534 |
| 275.000 | 56.561 | 5.000 | 1631.204 | 26.400 |
| 250.000 | 68.583 | 5.000 | 1627.735 | 26.650 |
| 262.000 | 63.309 | 5.000 | 1628.859 | 26.830 |
| 237.000 | 77.814 | 5.000 | 1626.266 | 26.829 |
| 262.000 | 64.930 | 5.000 | 1626.699 | 26.838 |
| 237.000 | 78.335 | 5.000 | 1625.416 | 26.865 |
| 224.000 | 91.901 | 5.000 | 1627.115 | 26.214 |
| 233.667 | 86.770 | 7.000 | 1623.679 | 41.991 |
| 227.015 | 91.570 | 6.818 | 1625.793 | 41.214 |
| 260.633 | 66.210 | 9.279 | 1623.206 | 46.217 |
| 224.000 | 113.901 | 6.000 | 1603.450 | 44.855 |
| 222.176 | 135.300 | 6.600 | 1623.252 | 57.726 |

Accept Delete Calculate Close

Performances achieved by a raw sizing function

In a typical example here

Weight 19.5 .. 57.7 kg

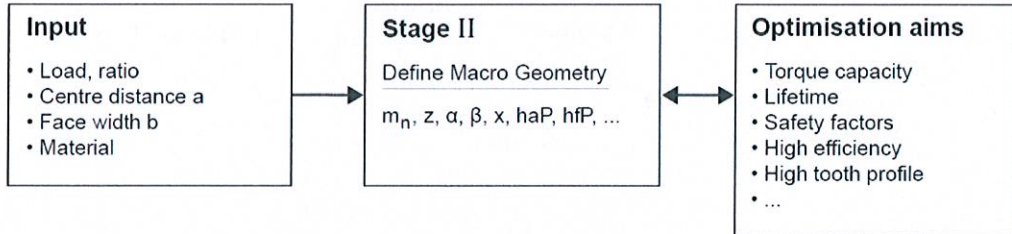
(a: 220..355 mm; b: 35..135 mm)

Cost variation in the range of: 34-100 %

(with the same torque capacity)

Stage II

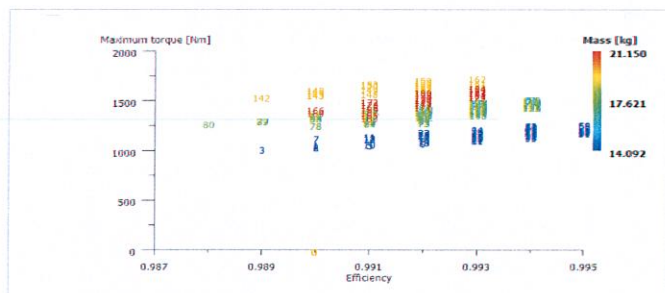
Layout process of a gear set: Stage II



Lay out of: Macro Geometry

Fine Sizing

| Parameter | Value | Unit | Minimum | Maximum | Step |
|------------------------------------|------------|------|-----------|------------|-----------|
| Normal module | 2.0000 | mm | 0.5000 | 5.0000 | 0.5000 |
| Pressure angle at normal addendum | 20.0000 | deg | 15.0000 | 25.0000 | 1.0000 |
| Profile shift at reference circle | 0.0000 | | -1.0000 | 1.0000 | 0.1000 |
| Chamfer thickness | 0.0000 | mm | 0.0000 | 0.0000 | 0.0000 |
| Knock for greater than coefficient | 0.0000 | | 0.0000 | 0.0000 | 0.0000 |
| Planetary gear set | | | | | |
| Planet to planet | 22000.0000 | mm | 8000.0000 | 10000.0000 | 1000.0000 |
| Planet to planet | 0.0000 | mm | 0.0000 | 0.0000 | 0.0000 |
| Planet to planet | 0.0000 | mm | 0.0000 | 0.0000 | 0.0000 |
| Planet to planet | 0.0000 | mm | 0.0000 | 0.0000 | 0.0000 |
| Planet to planet | 0.0000 | mm | 0.0000 | 0.0000 | 0.0000 |
| Planet to planet | 0.0000 | mm | 0.0000 | 0.0000 | 0.0000 |
| Planet to planet | 0.0000 | mm | 0.0000 | 0.0000 | 0.0000 |
| Planet to planet | 0.0000 | mm | 0.0000 | 0.0000 | 0.0000 |
| Planet to planet | 0.0000 | mm | 0.0000 | 0.0000 | 0.0000 |



Performances achieved by a Macro Geometry sizing function

In a typical example here

Torque capacity: 1020-1740 Nm → 58-100%;

Losses: 0.12 - 0.50% → 42-100%

Micropitting Slam: 0.7-2.7 → 26-100% etc.

(with the same centre distance a and face width b)

Esempio oflgear

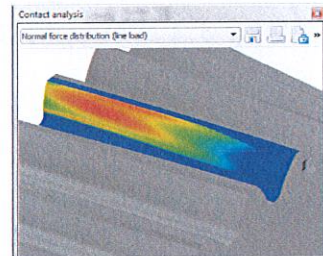
Layout of the Micro Geometry

The last phase in sizing a gear pair is to specify the flank line and profile modifications (also known as the "micro geometry").

To do so, the primary objective for which optimization has to be achieved: noise, service life, scuffing, micropitting or efficiency must be selected.

The calculation method for proving the effects achieved by micro geometry, the contact analysis under load ("Loaded Tooth Contact Analysis", or LTCA), is complex and time-consuming.

Unfortunately, the interpretation of LTCA results is not easy. All modifications applied on mating gears are interacting, so the decision of which modification to add or to change is difficult.



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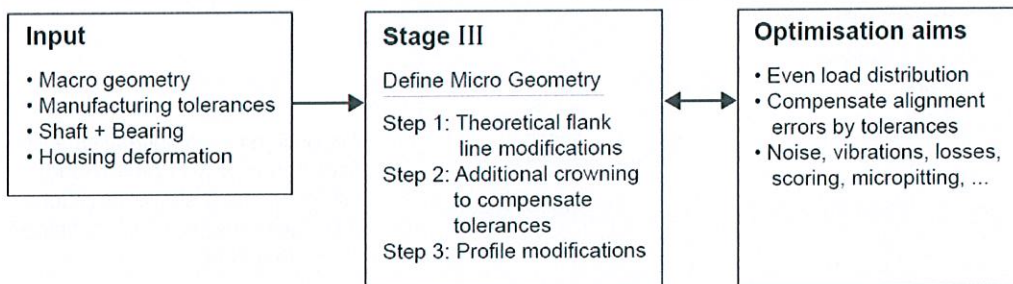
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c'è un menu dove si inseriscono tutte le geometrie (coeff.)

Stage III

Layout process of a gear set: Stage III



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Layout of the Micro Geometry

For a targeted sizing of the micro geometry, a step-by-step approach should be used, first specifying the flank line modification and then the profile modification.

A three-step process is proposed to perform a targeted sizing:

Step 1: Layout of the theoretical flank line modifications

Step 2: Including flank line manufacturing tolerances

Step 3: Layout profile modifications

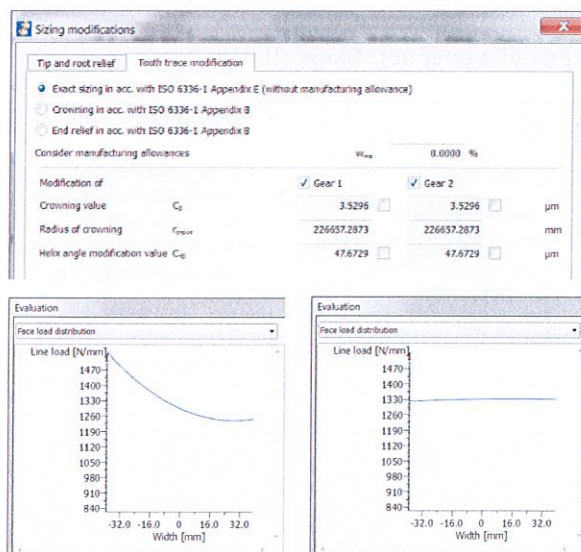
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Step 1: Layout of the theoretical flank line modifications



Load distribution before sizing

Load distribution after sizing

Proposition for an optimal flank line modification to get uniform load distribution for a single stage load (Input gear stage of the two-stage-industrial gearbox)

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Step 2: Including flank line manufacturing tolerances

Main manufacturing tolerances having impact on the load distribution (according ISO6336) are:

- f_{Hb} for the lead variation of the gears ($f_{HbT1} + f_{HbT2}$)
- f_{ma} for the axis misalignment in the contact plane)

According ISO6336-1, Annex E, K_{Hb} has to be calculated five times: Without tolerance, than with $+f_{Hb}$ & $+f_{ma}$, $+f_{Hb}$ & $-f_{ma}$, $-f_{Hb}$ & $+f_{ma}$, $-f_{Hb}$ & $-f_{ma}$. The highest K_{Hb} -value found will be used in the load capacity calculations.

Step 2: Including flank line manufacturing tolerances

When no expertise is available, the following procedure can be applied. In ISO 6336-1, Annex B, for gears having a flank line modification to compensate for deformation, **the crowning amount**

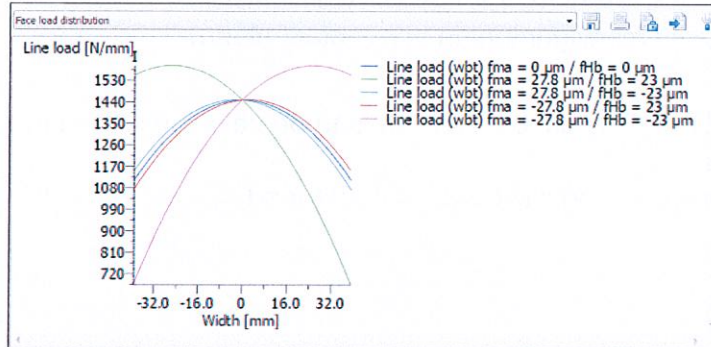
$$C_b = f_{H\beta T} \quad (1)$$

is proposed.

When such an additional modification is applied, clearly the load distribution over the face width as obtained in step 1 is not anymore uniform distributed. Therefore the face load factor K_{Hb} will increase. The goal is to avoid edge contact in all possible combination of deviations.

Step 2: Including flank line manufacturing tolerances

For all five combinations ($0, +f_{Hb}$ & $+f_{ma}$, $+f_{Hb}$ & $-f_{ma}$, $-f_{Hb}$ & $+f_{ma}$, $-f_{Hb}$ & $-f_{ma}$), the line load distribution in the operating pitch diameter has to be calculated and checked for edge contact).

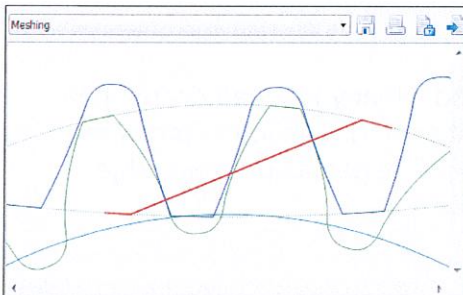


Load distribution with different manufacturing deviation values.

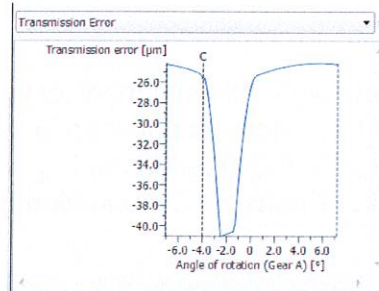
Step 3: Profile modifications

When the flank line modification is defined, the third step is to specify the profile modifications. Important features such as noise, losses, micropitting, scoring and wear can be improved by profile modifications. Therefore the layout criteria must be defined. Then the corresponding strategy is used.

Layout for low-noise:



Contact shock: Gear pair meshing and path of contact calculated with LTCA, showing the prolonged contact at start and end of the mesh.



PPTE: Peak-to-Peak Transmission error

Step 3: Use of a 'modification sizing' tool to find optimum design

Optimization of profile modifications in a case-by-case manner is extremely time-consuming and demanding.



"Analysis of modification variants" tool: Profile crowning variants

Cross-vary value and coefficient 1/2
Base modifications (do not change)

| No. | Gear | Flank | Type of modification | Value [µm] |
|-----|--------|-------|------------------------------------|------------|
| 1 | Gear 1 | both | Crowning | 15.0000 |
| 1 | Gear 1 | both | Helix angle modification, parallel | 57.0000 |
| 1 | Gear 2 | both | Crowning | 14.0000 |
| 1 | Gear 2 | both | Helix angle modification, parallel | 40.0000 |

Tab. I: Contains all modifications which will not be changed

| No. | Gear | Sync | Flank | Type of modification | Number of step | Value (min) [µm] | Value (max) [µm] |
|-----|--------|------|-------|------------------------------|----------------|------------------|------------------|
| 1 | Gear 1 | 1 | both | Profile crowning (barreling) | 7 | 40.0000 | 100.0000 |
| 2 | Gear 2 | 2 | both | Profile crowning (barreling) | 7 | 25.0000 | 85.0000 |

Tab. II: Definition of modifications which will be varied (here: Profile crowning)

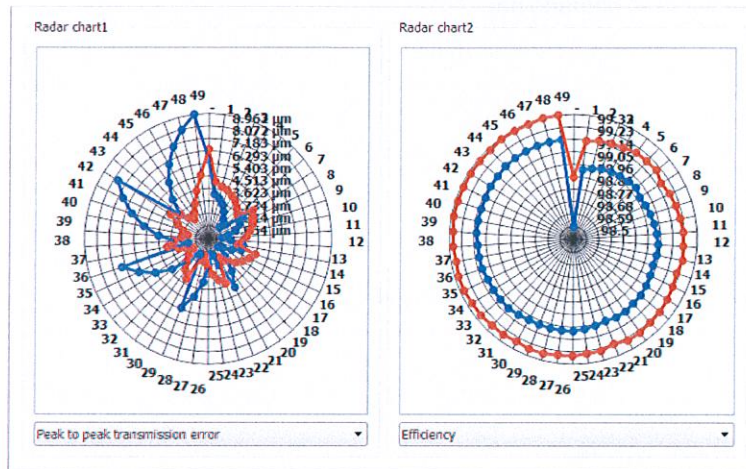
Step 3: Use of a 'modification sizing' tool to find optimum design

Table with numbered variants and selected main results

| Nr. | PPE [µm] | σ_{max} [N/mm ²] | Slam | η | SB | Gear 1: Value [µm] | Gear 1: Factor |
|-----|----------|-------------------------------------|-------|--------|-------|--------------------|----------------|
| 0 | 5.633 | 1791.456 | 1.421 | 98.881 | 3.180 | 0.000 | 0.000 |
| 1 | 2.680 | 1405.849 | 2.050 | 99.150 | 5.362 | 40.000 | 0.200 |
| 2 | 2.680 | 1405.849 | 2.050 | 99.150 | 5.362 | 40.000 | 0.583 |
| 3 | 2.680 | 1405.849 | 2.050 | 99.150 | 5.362 | 40.000 | 0.967 |
| 4 | 2.680 | 1405.849 | 2.050 | 99.150 | 5.362 | 40.000 | 1.350 |
| 5 | 2.680 | 1405.849 | 2.050 | 99.150 | 5.362 | 40.000 | 1.733 |
| 6 | 2.680 | 1405.849 | 2.050 | 99.150 | 5.362 | 40.000 | 2.117 |
| 7 | 2.680 | 1405.849 | 2.050 | 99.150 | 5.362 | 40.000 | 2.500 |
| 8 | 2.017 | 1482.830 | 2.185 | 99.208 | 5.262 | 50.000 | 0.200 |
| 9 | 2.017 | 1482.830 | 2.185 | 99.208 | 5.262 | 50.000 | 0.583 |
| 10 | 2.017 | 1482.830 | 2.185 | 99.208 | 5.262 | 50.000 | 0.967 |
| 11 | 2.017 | 1482.830 | 2.185 | 99.208 | 5.262 | 50.000 | 1.350 |
| 12 | 2.017 | 1482.830 | 2.185 | 99.208 | 5.262 | 50.000 | 1.733 |
| 13 | 2.017 | 1482.830 | 2.185 | 99.208 | 5.262 | 50.000 | 2.117 |
| 14 | 2.017 | 1482.830 | 2.185 | 99.208 | 5.262 | 50.000 | 2.500 |
| 15 | 1.659 | 1553.785 | 2.259 | 99.247 | 5.183 | 60.000 | 0.200 |
| 16 | 1.659 | 1553.785 | 2.259 | 99.247 | 5.183 | 60.000 | 0.583 |
| 17 | 1.659 | 1553.785 | 2.259 | 99.247 | 5.183 | 60.000 | 0.967 |
| 18 | 1.659 | 1553.785 | 2.259 | 99.247 | 5.183 | 60.000 | 1.350 |
| 19 | 1.659 | 1553.785 | 2.259 | 99.247 | 5.183 | 60.000 | 1.733 |
| 20 | 1.659 | 1553.785 | 2.259 | 99.247 | 5.183 | 60.000 | 2.117 |
| 21 | 1.659 | 1553.785 | 2.259 | 99.247 | 5.183 | 60.000 | 2.500 |
| 22 | 2.480 | 1617.688 | 2.316 | 99.277 | 5.122 | 70.000 | 0.200 |
| 23 | 2.480 | 1617.688 | 2.316 | 99.277 | 5.122 | 70.000 | 0.583 |
| 24 | 2.480 | 1617.688 | 2.316 | 99.277 | 5.122 | 70.000 | 0.967 |
| 25 | 2.480 | 1617.688 | 2.316 | 99.277 | 5.122 | 70.000 | 1.350 |
| 26 | 2.480 | 1617.688 | 2.316 | 99.277 | 5.122 | 70.000 | 1.733 |
| 27 | 2.480 | 1617.688 | 2.316 | 99.277 | 5.122 | 70.000 | 2.117 |
| 28 | 2.480 | 1617.688 | 2.316 | 99.277 | 5.122 | 70.000 | 2.500 |

Report length: Short form

Step 3: Use of a 'modification sizing' tool to find optimum design



Two charts with results (PPTe and efficiency) of 25 modification variants
 Red: At 100 percent load; Blue: At 75 percent load

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Step 3: Use of a 'modification sizing' tool to find optimum design

"Analysis of modification variants" tool: Tip relief variants

Cross-vary value and coefficient 1/2
 Base modifications (do not change)

| No. | Gear | Flank | Type of modification | Value [μm] |
|-----|--------|-------|------------------------------------|-------------------------|
| 1 | Gear 1 | both | Crowning | 15.0000 |
| 1 | Gear 1 | both | Helix angle modification, parallel | 57.0000 |
| 1 | Gear 2 | both | Crowning | 14.0000 |
| 1 | Gear 2 | both | Helix angle modification, parallel | 40.0000 |

Tab. I: Contains all modifications which will not be changed

| Conditions I | | Conditions II | | Results | Graphics | Grafik II | | | |
|--------------|--------|------------------|--------|----------------------|-----------------|-------------|-------------|----------------|----------------|
| No. | Gear | Synchronize with | Flank | Type of modification | Number of steps | Value (min) | Value (max) | Factor 1 (min) | Factor 1 (max) |
| 1 | Gear 1 | | 1 both | Tip relief, arc-like | 15 | 10.0000 | 150.0000 | 0.2000 | 2.5000 |
| 2 | Gear 2 | | 1 both | Tip relief, arc-like | 15 | 10.0000 | 150.0000 | 0.2000 | 2.5000 |

Tab. II: Definition of modifications which will be varied (here arc-like tip relief)

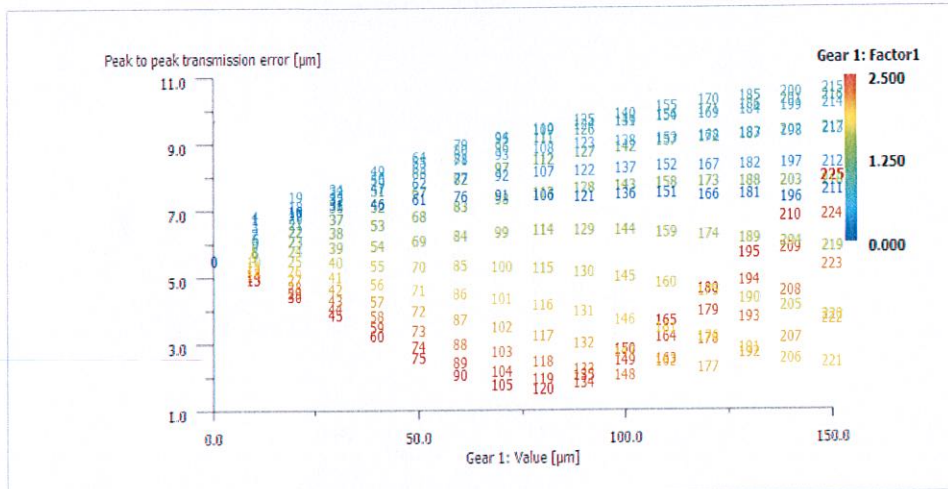
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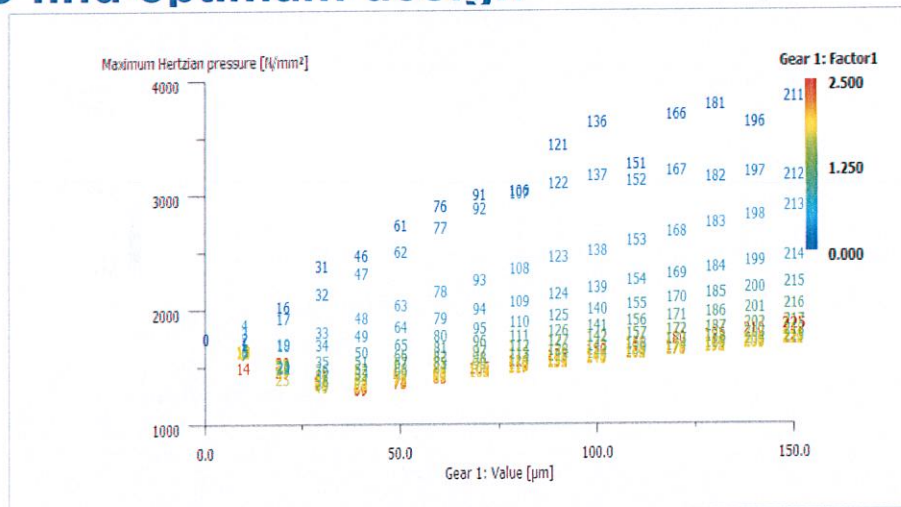
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Step 3: Use of a 'modification sizing' tool to find optimum design



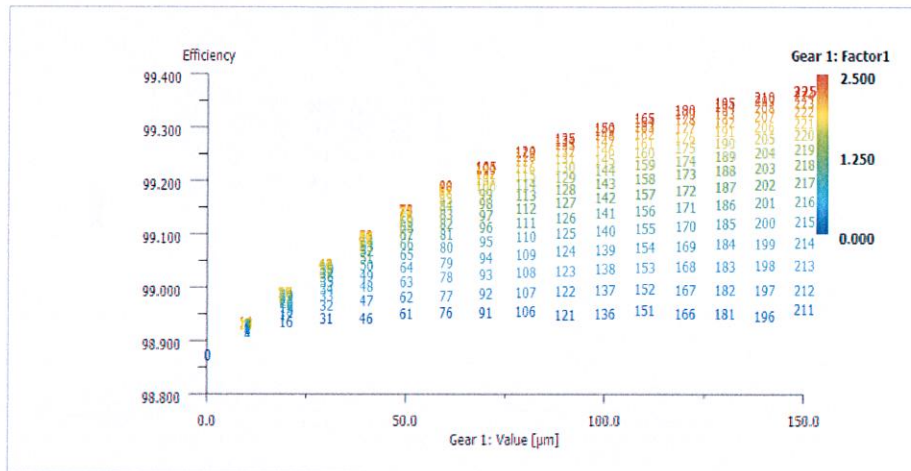
Display of 225 solutions, when the parameters (Tip relief ("Value mm") and Length ("Factor 1") of an arc-like tip relief are varied.
Here: **Peak-to-Peak-Transmission-Error (PPTe)**

Step 3: Use of a 'modification sizing' tool to find optimum design



Display of 225 solutions, when the parameters (Tip relief ("Value mm") and Length ("Factor 1") of an arc-like tip relief are varied.
Here: **Maximum Hertzian Pressure**

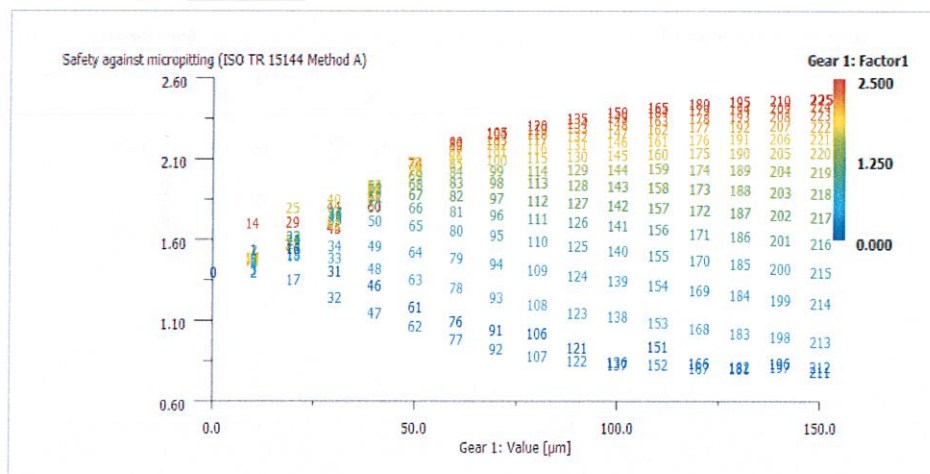
Step 3: Use of a 'modification sizing' tool to find optimum design



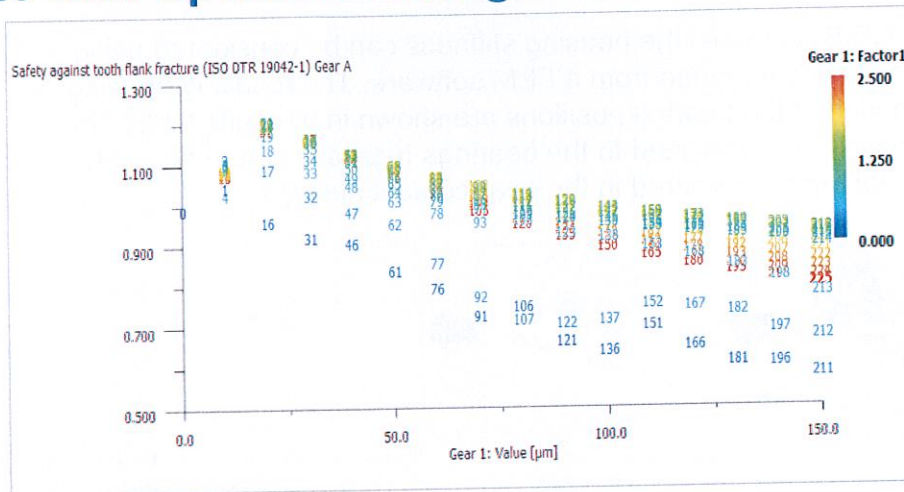
Display of 225 solutions, when the parameters (Tip relief ("Value mm") and Length ("Factor 1") of an arc-like tip relief are varied.

Here: **Efficiency**

Step 3: Use of a 'modification sizing' tool to find optimum design



Step 3: Use of a 'modification sizing' tool to find optimum design



Display of 225 solutions, when the parameters (Tip relief ("Value mm") and Length ("Factor 1") of an arc-like tip relief are varied.

Here: **Safety factor against Tooth Flank Fracture (TFF)**

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Step 3: Finish

End of Step 3:

- Make a first selection of best variants (profile modifications)
- Check the LTCA results of these variants
- Choose the best over all variant
- Recheck load distribution *)

Method is successfully tested:

The time used by the design engineer to find optimum modifications for both stages of a gearbox was 15 minutes.

*) Normally the load distribution as defined in Step 2 is typically not much changed by the added profile modifications.

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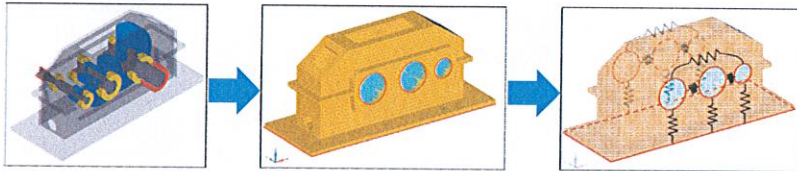
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Considering housing and/or planet carrier stiffness

In any KISSsys model the housing stiffness can be considered using a stiffness matrix imported from a FEM software. The resulting housing deformation at the bearing positions are shown in a results table. The deformations are assigned to the bearings (typically outer ring) in the shaft calculation and considered in the gear contact analysis.

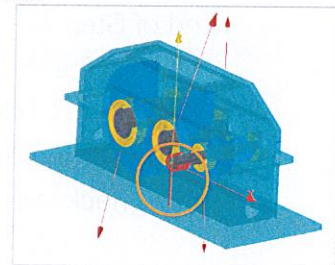


| | b1 | b2 | b1 | b2 | b1 | b2 |
|--------|----------|----------|----------|----------|----------|----------|
| shaft1 | s1 | s1 | s2 | s2 | s3 | s3 |
| ux | -0.21515 | -0.24219 | -0.2166 | -0.24505 | -0.22307 | -0.24986 |
| uy | -0.14555 | 0.052544 | -0.18155 | 0.17393 | -0.12539 | 0.27237 |
| uz | 0.3207 | 0.31393 | 0.11315 | 0.10758 | -0.21719 | -0.21053 |

Bearing outer ring displacements in mm
(x, y: horizontal;
z: vertical)

Example: Use of the 3-step-procedure with a industrial 2-stage gearbox

For a typical industrial two-stage parallel shaft reducer the modifications are optimized using the 3-step method. The process is repeated twice, with and without considering housing stiffness, to get an indication on the influence of the housing.



First the load distributions of the two gear pairs without modifications are calculated. The face load factors are calculated according to Annex E in ISO6336-1, using the axis deformations from the shaft calculation.

| Gear Pair | $K_{H\beta}$ | $K_{H\beta}$ |
|------------------------|-----------------------------|--------------------------|
| | Without housing deformation | With housing deformation |
| HSS (High speed stage) | 1.17 | 1.16 |
| HSS (Low speed stage) | 1.30 | 1.32 |

Face load factors without flank line modifications

Proposed layout procedure Step 1 to 3

- The time used by the design engineer to find optimum modifications for both stages was 15 minutes.
- The optimum flank line modifications as defined in Step 1 are only slightly different when housing stiffness is considered (only 10% change in the helix angle modification value).
- The additional modifications in Step 2 and the profile modifications in Step 3 are identical with and without consideration of housing stiffness.
- The additional crowning added in Step 2 to compensate for manufacturing tolerances is much bigger (5 times) than the difference between modifications in Step 1 used to compensate shaft deflection with and without considering housing stiffness. Therefore, for practice-oriented solutions the influence of the housing stiffness is so small that it is negligible.

Conclusion

Optimization of flank line and profile modifications for a specific application is not an easy task. The three-step methodology has proven highly successful since it was introduced two years ago. The layout of the modifications for an industrial gearbox shows that for a gearbox with parallel shafts including external forces acting on it, the housing deformations have an insignificant influence on the resulting gap in the meshing of the gears.

This method can also be used in applications such as wind power, ship transmission systems, or helicopters in which it is demanding to define the modifications due to the extreme load spectrum or high housing deflections.

THANK YOU
for your attention!

SHARING KNOWLEDGE

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