



AGMA Technical Paper

Application and Improvement of Face Load Factor Determination Based on AGMA 927

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Application and Improvement of Face Load Factor Determination Based on AGMA 927

(Accurate and Fast Algorithm for Load Distribution Calculation, for Gear Pair and Planetary Systems, Including Duty Cycle Analysis)

Dr. Ulrich Kissling, KISSsoft AG

[The statements and opinions contained herein are those of the author and should not be construed as an official action or opinion of the American Gear Manufacturers Association.]

Abstract

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. In the international standard for cylindrical gear rating, the ISO 6336-1 [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. Also AGMA 2001 (or AGMA 2101) [2] proposes a formula for $K_{H\beta}$, different from ISO 6336, but again not always appropriate. Therefore a note in AGMA stipulates, that "it may be desirable to use an analytical approach to determine the load distribution factor".

In the last edition of ISO 6336 (2006), a new annex E was added: "Analytical determination of load distribution". This annex is entirely based on AGMA 927-A01 [3]. It is a well-documented procedure to get a direct and precise number for the face load factor.

Today an increasing number of gear designers are using tooth contact analysis (TCA) methods [4] to get precise information over the load distribution on the full gear flank. Contact analysis is very time consuming and does not permit to get a value for $K_{H\beta}$, as defined by the ISO or AGMA standard. A contact analysis result combines different factors of ISO 6336 as $K_{H\beta}$, $K_{H\alpha}$, Z_{ϵ} , Z_{β} , Z_B , Z_D and buttressing effects, etc., thus to 'extract' $K_{H\beta}$ from a TCA is not possible.

The use of the algorithm, as proposed by AGMA 927, is a good solution to get proper values for $K_{H\beta}$; it is simpler and therefore much quicker than a contact analysis calculation. The paper explains how this algorithm can be applied for classic gear pair rating procedure, for ratings with complex duty cycles and even for planetary systems with interdependent meshings between sun, all planets and ring.

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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. In the international standard for cylindrical gear rating, the ISO 6336-1 [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. Also AGMA 2001 (or AGMA 2101) [2] proposes a formula for $K_{H\beta}$, different from ISO 6336, but again not always appropriate. Therefore a note in AGMA stipulates, that "it may be desirable to use an analytical approach to determine the load distribution factor".

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How it started: A problem during the drilling of the world longest tunnel in the Swiss Alps

Since 1999 the world's longest tunnel (57 km or 36 miles) is under construction in the Swiss Alps. In 2002 a problem was found in one of the tunnel boring machines during an inspection. The main drive of the machine consists of a large ring gear, driven by 8-12 pinions. The outer ring of some of the bearings on the pinion shaft did rotate in the housing and therefore the bearing seat was worn. Underground in the tunnel the bores were repaired as well as possible, the final check showed that the coaxiality had a deviation up to 0.2 mm (0.008 in). We were requested to propose the best possible flank line modification to compensate the coaxiality error. For logistical reasons all the pinions had to be replaced; all pinions should get the same modification. Therefore our job was to propose a modification, which could compensate best possible a coaxiality error between -0.2 and +0.2 mm and to prove, that with these pinions, the remaining 1500 operation hours until the tunnel break-through could be performed without failure.

This engineering problem contained some new, interesting aspects. In the shaft calculation of KISSsoft [5] we had since long-time a feature to calculate the gap between the face of the gear and a stiff wall. This was a helpful feature to find easily the optimum flank line modification. But the given problem needed some improvement of the software, because for the life time calculation according ISO 6336 the determination of the face load factor $K_{H\beta}$ was needed; and therefore the load distribution over the face width had to be calculated considering the stiffness of the mating gear.

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 ($c_{\gamma\beta}$ [1]), the line load distribution is determined. This approach is well documented in ISO 6336-1. The standard simplifies the real situation through assumption of a linear load distribution (Figure 1).

Determination of the gap in the tooth contact

In the MAAG book [6], the deduction of the gap through superposition of bending and torsion deformation is explained (Figure 2). As additional simplification it is assumed, that the mating gear is infinite stiff.

Without flank line modification, in the example shown in Figure 2, the load would be bigger on the torque input side. If a modification as shown in Figure 2 is applied on the pinion flank line, then a uniform load distribution would result. This is true, if the meshing gear is effectively very stiff – or if also on the mating gear a flank line modification is applied. In the formulas for $K_{H\beta}$ of ISO 6336-1 (chapter 7) it is assumed, that the pinion shaft is much slender than the gear shaft, thus the deformation of the gear shaft is much less and can be neglected. For gear pairs with a reduction $i > 2$, in many cases this is a realistic assumption.

In Figure 3 the wording “deviations in the contact plane” is explained. The deformation in every section of the shaft must be determined in the operating pitch point (W). A displacement of the point W due to bending or torsion parallel to the tooth flank will change a little bit the sliding velocity between the flanks, but otherwise has no effect at all. To get the necessary data for the determination of the gap, the component of deformation in point W (x, z coordinate) normal to the flank, f_{bn} and f_{bt} , are requested. With this data the gap between the meshing flanks is directly located.

Manufacturing errors, housing deformations and bearing stiffness result normally as linear deviation over the face width. These values can be considered through radial displacement of a bearing versus another and through considering the bearing stiffness when calculating the shaft deflection.

This procedure was implemented in our shaft calculation software [5] in 1997. Figure 4 displays the user interface. The software recognises automatically all the gears on the shaft, and deduces the meshing point W coordinates and the normal N to the flank.

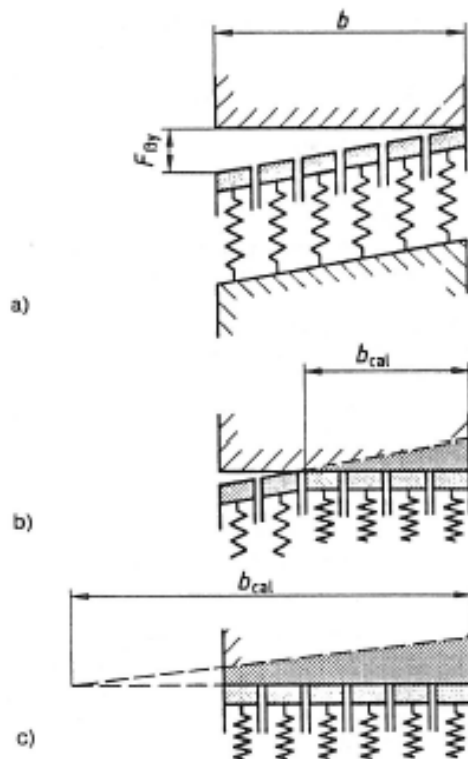


Figure 1. Display of the gap and the corresponding load distribution following ISO 6336-1 [1]

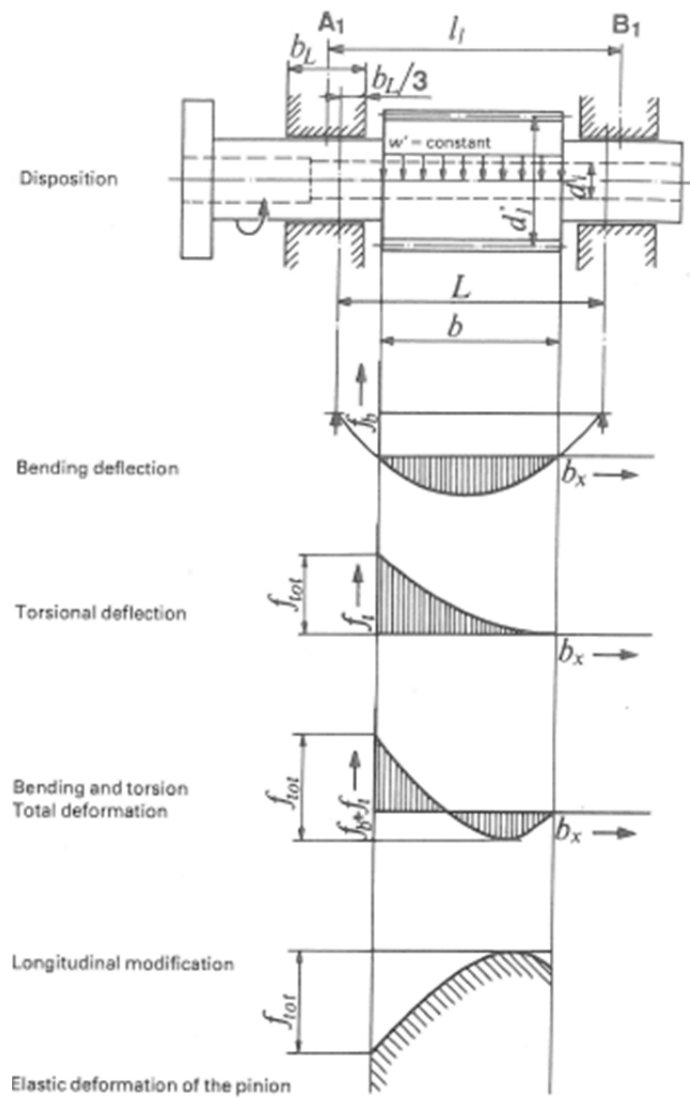


Figure 2. Display of the determination of gap through the deformation components, and deduction of the corresponding flank line modification [6]

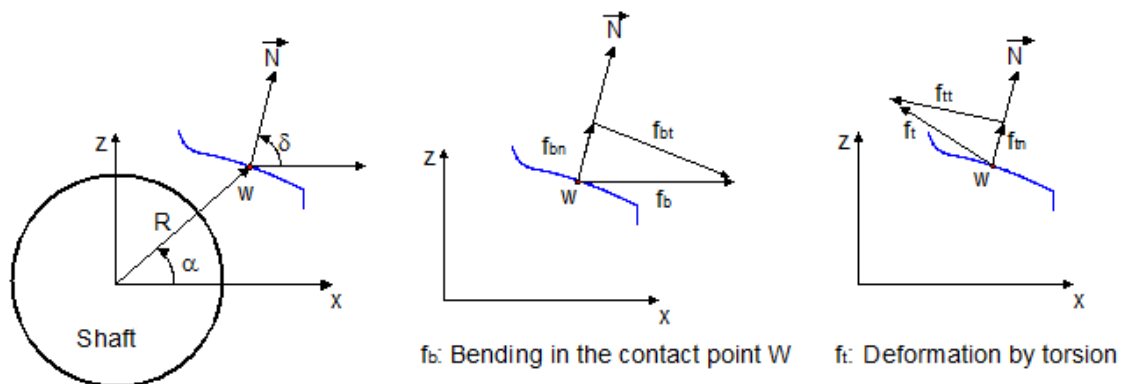


Figure 3. Determination of the gap in the gear mesh (in a shaft section)

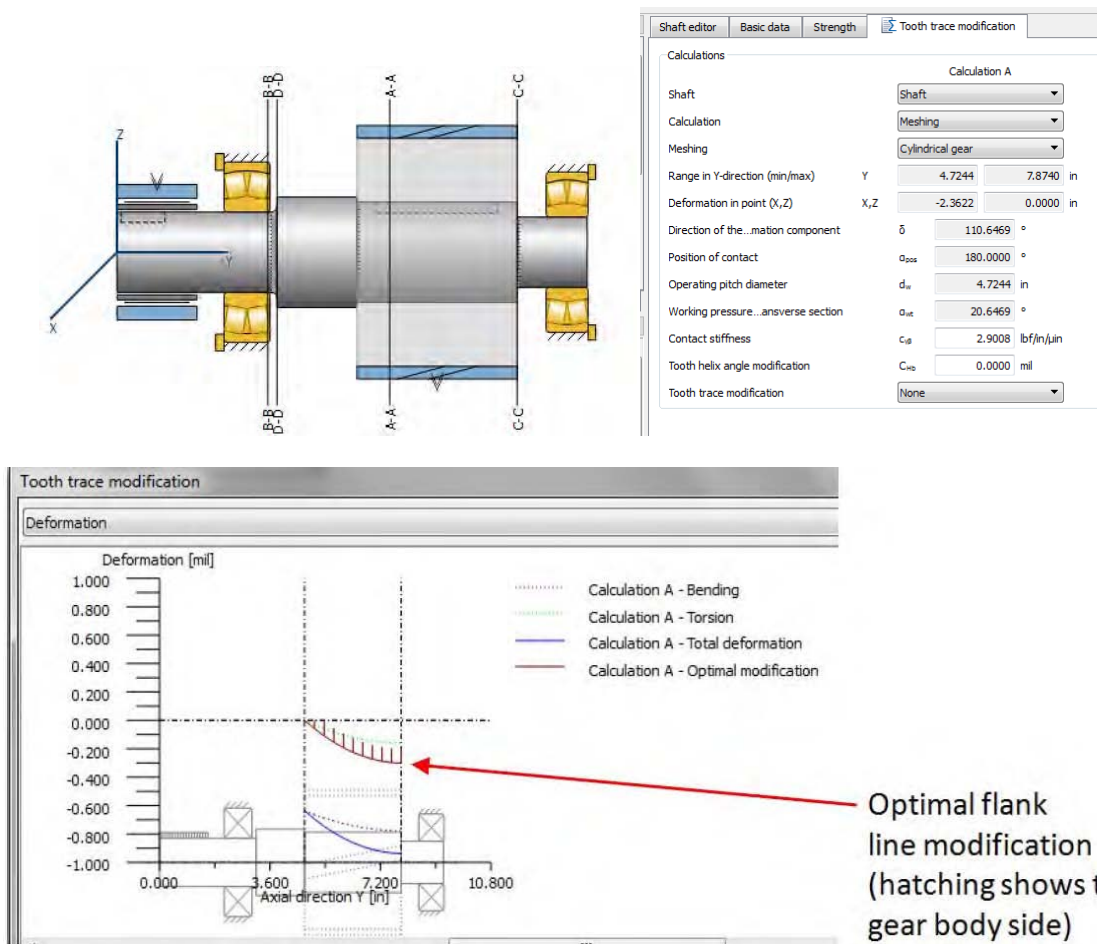


Figure 4. Display of the gap and proposition for an optimum flank line modification in KISSsoft
(Above: Shaft editor and parameters for the determination of the gap in the contact plain. Below: Bending, torsion and total deformation over the face width)

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

The determination of the load distribution (in N/mm or lbf/in) according to ISO 6336 [1] is simple, because the tooth meshing stiffness $c_{y\beta}$ is considered as constant over the face width. The calculation is performed as displayed in Figure 1. The face width is subdivided in some (11...100) sections. To start the iteration, an initial distance δ between the teeth is assumed. Then with $c_{y\beta}$ the partial load F_{ti} per section is calculated. The sum of all F_{ti} has to be identical to the transmitted tangential load F_t :

$$F_t = \sum_i F_{ti} \quad (1)$$

The distance δ is therefore (by iteration) changed until equation 1 is fulfilled. The result is the line load distribution as in Figure 5. The face load factor $K_{H\beta}$ is then the quotient of the maximum line load divided by the mean line load as defined in ISO 6336 [1]:

$$K_{H\beta} = \frac{w_{\max}}{w_m} = \frac{\text{maximum load per unit face width}}{\text{average load per unit face width}} \quad (2)$$

To compensate uneven load distribution (as in Figure 5), adapted flank line modifications should be used. As shown in the theory (MAAG book, figure 1), the optimum flank line modification is identical to the inverted gap curve. An example is shown in Figure 4.