



American
Gear Manufacturers
Association

Technical Resources

10FTM15

AGMA Technical Paper

Drive Line Analysis for Tooth Contact Optimization of High Power Spiral Bevel Gears

By J. Rontu and G. Szanti and
E. Mäsä, ATA Gears Ltd.

Drive Line Analysis for Tooth Contact Optimization of High Power Spiral Bevel Gears

Jesse Rontu and Gabor Szanti and Eero Mäsä, ATA Gears Ltd.

[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

It is a common practice in high power gear design to apply relieves to tooth flanks. They are meant to prevent stress concentration near the tooth edges. Gears with crownings have point contact without load. When load is applied, instantaneous contact turns from point into a Hertzian contact ellipse. The contact area grows and changes location as load increases. To prevent edge contact, gear designer has to choose suitable relieves considering contact indentations as well as relative displacements of gear members.

In the majority of spiral bevel gears spherical crownings are used. The contact pattern is set to the center of active tooth flank and the extent of crownings is determined by experience. Feedback from service, as well as from full torque bench tests of complete gear drives have shown that this conventional design practice leads to loaded contact patterns, which are rarely optimal in location and extent. Too large relieves lead to small contact area and increased stresses and noise; whereas too small relieves result in a too sensitive tooth contact.

Today it is possible to use calculative methods to predict the relative displacements of gears under operating load and conditions. Displacements and deformations originating from shafts, bearings and housing are considered. Shafts are modeled based on beam theory. Bearings are modeled as 5-DOF supports with non-linear stiffness in all directions. Housing deformations are determined by FEM-analysis and taken into account as translations and rotations of bearing outer rings. The effect of temperature differences, bearing preload and clearances are also incorporated.

With the help of loaded tooth contact analysis (LTCA), it is possible to compensate for these displacements and determine a special initial contact position that will lead to well centered full torque contact utilizing a reasonably large portion of the available tooth flank area. At the same time, crownings can be scaled to the minimum necessary amount. This systematic approach leads to minimum tooth stressing, lower noise excitation as well as increased reliability and/or power density as compared to conventional contact design method.

During recent years ATA Gears Ltd. has gained comprehensive know-how and experience in such analyses and advanced contact pattern optimization. The methodology and calculation models have been verified in numerous customer projects and case studies.

Copyright © 2010

American Gear Manufacturers Association
500 Montgomery Street, Suite 350
Alexandria, Virginia, 22314

October 2010

ISBN: 978-1-55589-990-5

Drive Line Analysis for Tooth Contact Optimization of High Power Spiral Bevel Gears

Jesse Rontu and Gabor Szanti and Eero Mäsä, ATA Gears Ltd.

Introduction

In a majority of spiral bevel gears produced, the tooth contact is initially placed at the center of tooth flank during manufacturing. Sufficient crownings are applied to prevent the contact from reaching tooth edges under load. However, the use of large crowning also has a down side of increasing contact stresses, since the area in contact at a particular moment is reduced. With constantly growing demands for higher power density and lower noise generation, there are pressures for decreasing crownings. In more general sense, there is often a great need to optimize tooth flank topography for a certain application. This requires accurate knowledge about behavior of tooth contact under load. Regardless of the optimization goal, the change of relative position of bevel gears under load is an important factor.

When loads and temperature differences are applied to a gear drive, the relative position of pinion and wheel changes due to deformations and displacements related to bearings, shafts and housing. This causes changes in the tooth contact, significance of which is dependent on magnitude and mutual relations of the displacements as well as characteristics of the tooth geometry. One of the main concerns is the spreading and movement of contact pattern, which ideally should be located at the center of tooth flank under load and cover as much of the flank area as possible. If the behavior of contact pattern is known, pre-compensation can be applied in finish machining of tooth flanks to ensure good running properties under load. Usually this means that the tooth flank topography is modified so that the initial contact pattern (without load) is moved from the center of the flank by a certain amount.

Traditionally the knowledge of tooth contact behavior has been attained through practical experiences. This requires time consuming and expensive prototype testing. Alternative approach

is the one based on computer simulation, by which significant cost saving are possible. During recent years, tooth contact optimization based on *Loaded Tooth Contact Analysis* (later LTCA) has been applied with good success in numerous customer projects involving marine, industrial and automotive bevel gear applications. In LTCA the mesh of spiral bevel gears is simulated using 3D tooth geometry, taking into account the actual relative position of gears under load. This paper describes computational process used to determine how the relative position of bevel gears changes when load and temperature differences are applied on a gear drive. The process is combination of different calculation methods and is hereafter referred to as Drive Line Analysis. In addition to the methods usually used in Drive Line Analysis, some alternative approaches are also mentioned to provide a more general overview of applicable methods.

Relative position of bevel gears

In nominal position the pitch cone apexes of bevel pinion and wheel (if not a hypoid gear pair) coincide. Deviation from this position (location + orientation) can be fully defined by four displacement values, which are hereafter referred as *relative displacements*. As shown in Figure 1, they consist of deviation of shaft angle (S), offset (E), pinion axial location (P) and wheel axial location (G).

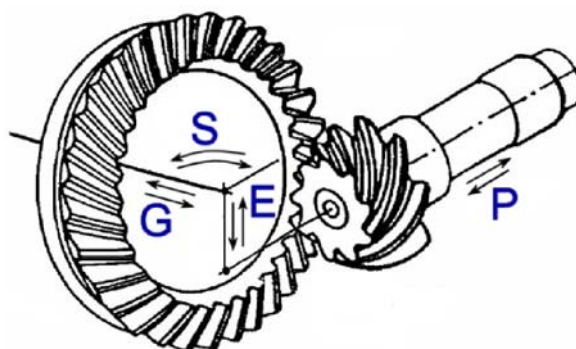


Figure 1. Relative displacements of bevel gears

Drive line analysis

To fully understand tooth contact behavior in a certain application, the chain of events from assembly (tooth contact adjustment) to operating conditions (loads and temperature differences applied) has to be traced. To accurately determine the displacements of bevel gears, a detailed analysis of the whole drive line consisting of shafts, bearings and housing, is required. The “core” of analysis is comprised of separate calculation models for pinion shaft and wheel shaft, which are hereafter referred to as *shaft-calculation models*. These models are used to simulate deformations and displacements of shafts and bearings. Several commercial software with ranging levels of capabilities are available for this purpose. Beam theory is practically always used to calculate shaft deflection, but there are significant differences in the way bearings are modeled. At the simplest level, bearings are considered as radially stiff “hinges”, which does not represent reality very well. In the more advanced software, such as used in Drive Line Analysis described in this paper, bearing internal geometry and stiffness nonlinearity are considered in order to accurately model the real behavior.

Bearing stiffness

Gears are usually supported to gear unit housing with rolling bearings. Bearing stiffness varies significantly depending on type of rolling element (Figure 2), affecting the displacement behavior of the shaft-bearing system. Another significant factor is the internal alignment capability (Figure 3). In shaft calculation models used in Drive Line Analysis, bearings are modeled as supports with 5 degrees of freedom: two radial-, two tilt- and one axial direction (Figure 11). The “missing 6th DOF” is the bearing rotation, which is naturally of no interest. Nonlinear bearing stiffness in every direction is modeled starting from deformations of individual contacts between rolling elements and raceways (Figure 4), also taking into account the internal clearance and operational contact angle. With this modeling method it is possible to accurately predict the distribution of loads and subsequently the displacements. Although nonlinear bearing stiffness leads to iterative calculation, the calculation times

are minimal due to the analytical theories applied. More detailed description of the modeling theory can be found in DIN ISO 281 Beiblatt 4 [1].

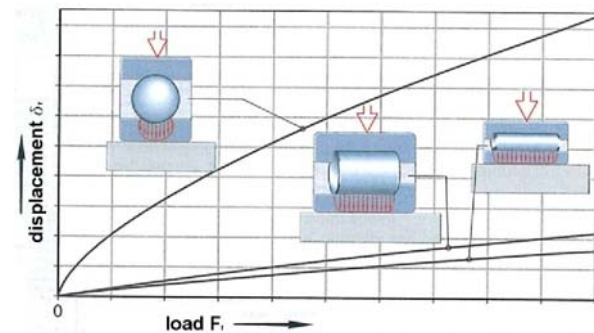


Figure 2. Radial stiffness of different bearing types [2]

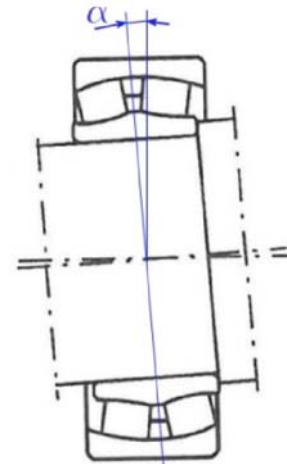


Figure 3. Bearing internal alignment capability

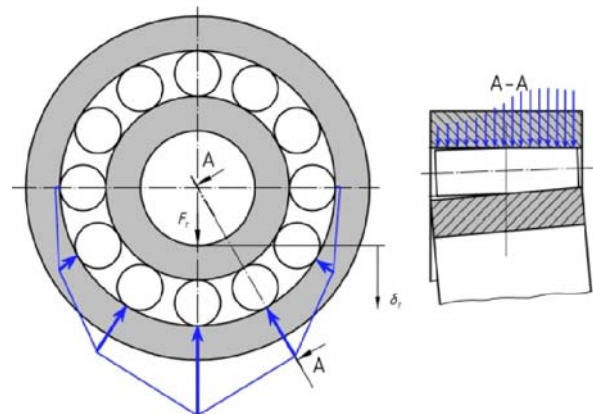


Figure 4. Deformation of individual element contact

Loads

Tooth forces are considered as point loads acting on the center of the tooth flank at mean pitch diameter d_m (Figure 5). Tooth force components F_t , F_r and F_a are calculated based on mean spiral angle β_m , normal pressure angle α_n and pitch cone angle δ . Especially for the wheel, the axial location of the acting point of the tooth forces does not always represent the axial location where the forces are actually conveyed to the shaft. An example of such situation is presented in Figure 6. In shaft-calculation models this is taken into account by transferring the tooth forces axially by a distance of d_f and correspondingly adding two additional bending moments $M_x = F_r d_f$ and $M_y = F_t d_f$. The same principle is used to correctly model bearing reactions in cases with bearings with nonzero pressure angle (e.g., taper roller bearings).

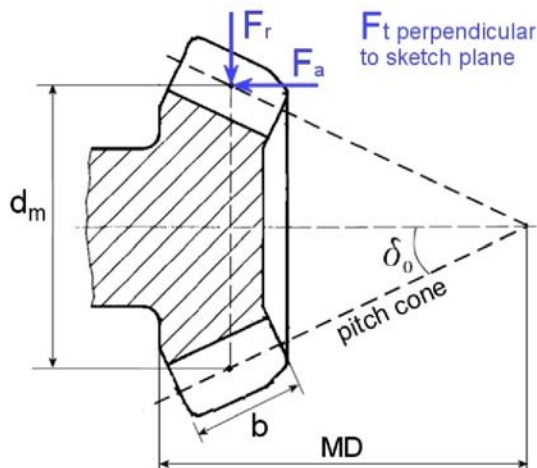


Figure 5. Application point of tooth forces

In addition to tooth forces there are usually external forces that also need to be included, such as propeller thrust force in marine thrusters. They are applied to their appropriate location on the shaft using the same principles as with the tooth forces. The weight of components is seldom important from deformations' point of view, but might instead be significant for other reasons discussed later in this paper.

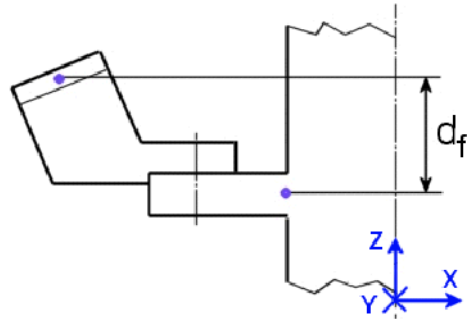


Figure 6. Application point used in shaft calculation

Bearing clearances and preload

Depending on the arrangement, bearing clearance and pretension can have significant influence on gear displacements. In shaft calculation models, values in **operating conditions** are used, which often differ significantly from assembly values due to temperature differences. Clearances can be divided into *internal* and *external* clearances (Figure 7).

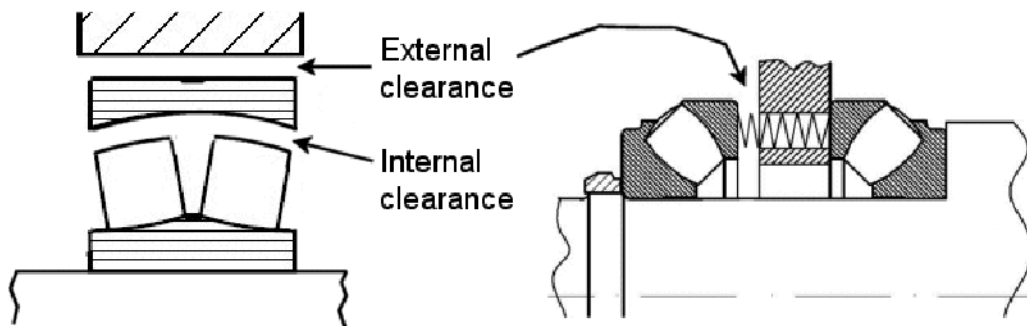


Figure 7. Examples of internal and external bearing clearances