



AGMA Technical Paper

Striving for High Load Capacity and Low Noise Excitation in Gear Design

By Dr. K. Stahl, Dr. M. Otto and
M. Zimmer, Gear Research
Centre (FZG)

Striving for High Load Capacity and Low Noise Excitation in Gear Design

Dr. K. Stahl, Dr. M. Otto and M. Zimmer, Gear Research Centre (FZG)

[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

In the design process of gearboxes, common requirements are high load capacity and low noise excitation. Reaching both goals is laborious and normally requires a trade-off. Detailed analyses of contact conditions and deformations are necessary. These should take place in an early design stage to realize a mostly straightforward design approach and prevent late design changes.

Focused on cylindrical gears, the paper covers an approach starting at the first draft of a gearbox. Defining the macrogeometry of the teeth regarding load capacity calculation according to standards leads to a reasonable gear design. On that basis, the micro geometry of the teeth is specified and load distribution as well as noise excitation is calculated. The design parameters are interdependent so provisions have to be made to adjust each step on the remaining ones. Effects resulting from changing profile contact ratio under load and contact patterns not covering the whole flank have to be regarded. The beneficial effect of a modified microgeometry is dependent on the ability to precisely account for contact conditions and meshing clearances.

To find an optimal solution for the competing goals of capacity and excitation, detailed calculation methods are required. To be able to apply latest research results, these are implemented in highly specialized software. The task described above is handled by using the software that was developed at the Gear Research Center (FZG) with funding by the German Research Association for Gears and Transmissions (FVA). The underlying calculation methods and analyzed phenomena are covered.

Copyright © 2013

American Gear Manufacturers Association
1001 N. Fairfax Street, Suite 500
Alexandria, Virginia 22314

September 2013

ISBN: 978-1-61481-068-1

Striving for High Load Capacity and Low Noise Excitation in Gear Design

Dr. K. Stahl, Dr. M. Otto and M. Zimmer, Gear Research Centre (FZG)

General method

In the early stage of gearbox design, elements have to be dimensioned by simple methods that don't require many input data, because only few values are already determined and fixed. Using these rough design methods is efficient and provides a fast early design process. In this stage many dimensions of the gears and the toothing are determined that have significant influence on load capacity and noise excitation. Though a detailed analysis is not in focus yet, the final goal of optimization has to be kept in mind. The last step of optimization usually is an adequate design of flank modifications. The possibility to improve the gear mesh behavior by flank modification is closely dependent on the selected main geometry of the toothing.

A different approach, e.g., starting with a detailed FE-Simulation of the whole gearbox, is not efficient. Complexity of the model and number of necessary parameters are so big so that it severely limits the flexibility that is needed in the early stage of the design process.

A fast approach to flank microgeometry for low noise excitation is described by Houser and Harianto [9]. A variation of microgeometries is analyzed by an analytical method and an interactive selection allows the identification of a desired result. The modifications are designed to also account for the influence of manufacturing deviations. For increasing normal module, that influence gets less significant, since other parameters (clearances, deformations) amount to most of the resulting deviations in contact.

A method with focus on load distribution is discussed in [7] by Thoma et al., the tooth contact analysis includes the influences of bearings and shafts. The load distribution resulting from the calculation is used to evaluate load distribution factors according to ISO 6336-1.

Pears et al. [15] describes an evaluation method for the contact stiffness in the mesh that is used for an analysis of transmission error. The influences of further elements like shafts and bearings are included.

The process described in this paper starts with rough design methods and successively proceeds to more detailed methods. Necessary calculations are carried out by several specialized software programs. The programs from early design stages provide data for latter, more complex steps and programs. By this approach flexibility stays high and necessary design changes can be realized even late in the design process. The resulting data are passed on to the more complex analyses without obstacles between the different programs. Data management is controlled by a unified User Interface for all programs (FVA Workbench [4], see Figure 1).

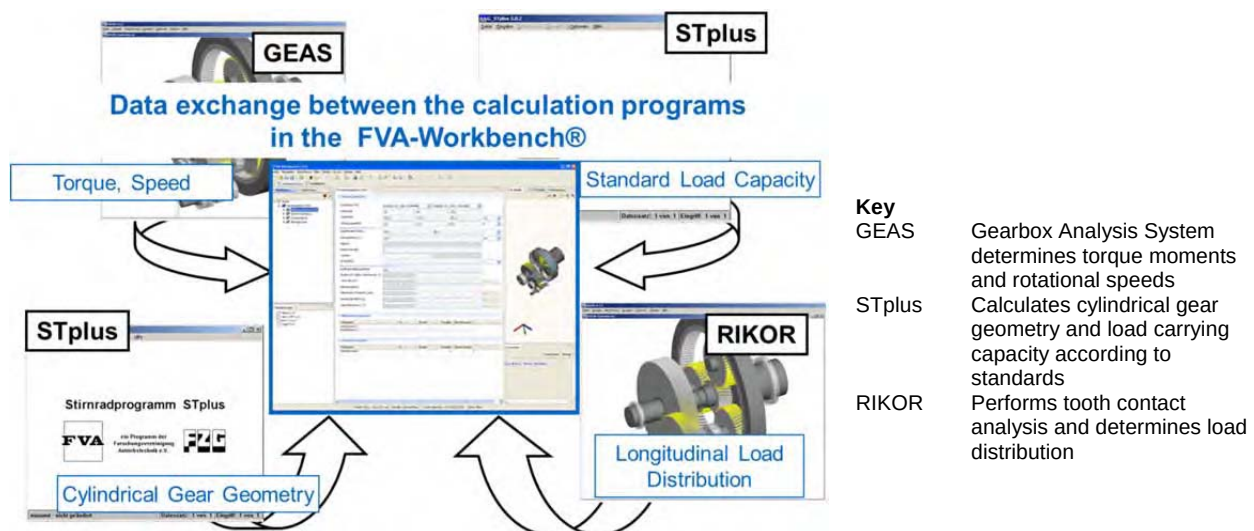


Figure 1. Interaction of programs in design process

Succession of design steps:

- Define gearbox structure;
- Distribute the ratios of the different stages to reach the gearbox ratio (GAP [1]);
- Design main geometry of the gears (STplus [8]) according to load capacity requirements ;
- Dimension shafts and bearings, perform detailed calculation of shaft deformation and bearing lifetime (RIKOR/WELLAG [16],[18]), include housing elasticity and further influences ;
- Determine load distribution by tooth contact analysis (RIKOR [16]);
- Design flank modifications to reach high load capacity, check for noise excitation;
- Optimize flank modifications for low noise excitation (DZP [5]).

Finally, the goals high load capacity and low noise excitations are reached by designing flank modifications. The goal high load capacity leads to a detailed definition of the flank modifications to ensure even load distribution, usually at high loads. Noise excitation behavior may require a different microgeometry, because the noise relevant operating range is related to lower torque moments. A trade-off between the two goals is required to determine the final flank modifications.

Gear main geometry

As an example, a gearbox structure with two helical gear stages and one planetary gear stage will be used (see Figure 2). The focus in this paper will be the high speed stage. For this given structure, the ratios of the stages have to be determined. The goal here is to reach a design that is compact and light. The ratio is an important factor for the mass of the gears. The torque moment transmitted in the mesh is the main influence for the dimension of the teeth. With given power, low speed shafts transmit a higher torque moment than high speed shafts. This leads to bigger teeth and wider gears on low speed shafts and ultimately a higher mass of the respective stages. Therefore a planetary stage with the advantage of internal power split is used for the low speed stage and helical stages are selected for the higher speeds. Determining the ratios of the stages is an optimizing task that is performed by the program GAP according to suggestions by Winter [14] or Linke [13].

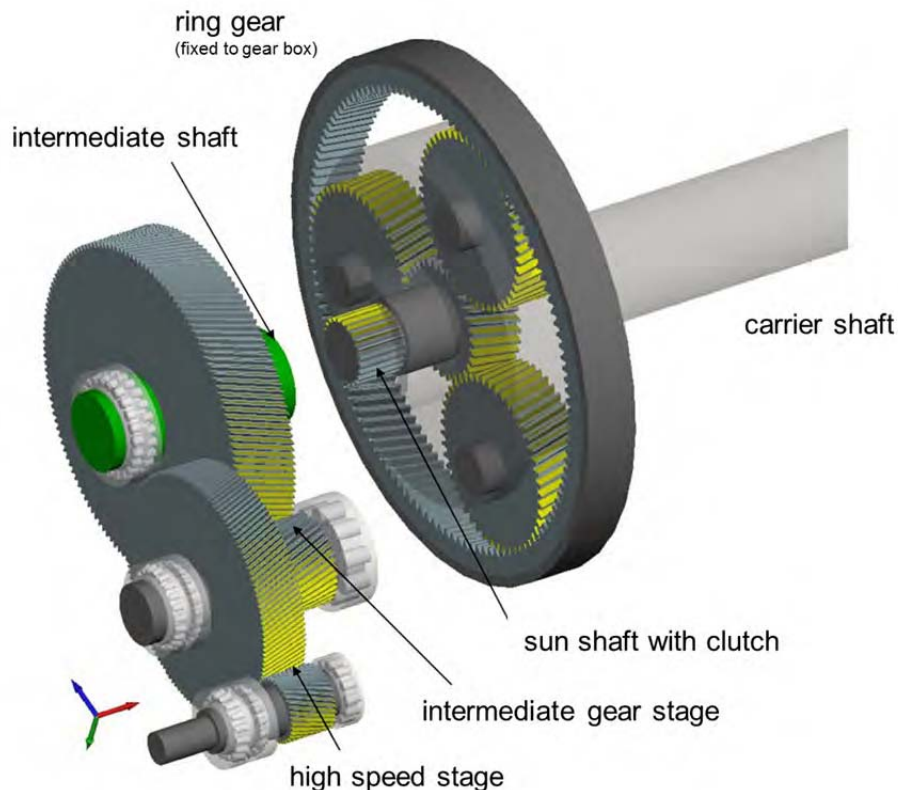


Figure 2. Sketch of example gearbox

It is necessary to rate the load capacity of the gear meshes in the optimization process. This can be done by simple methods like K^* value for flank load capacity and U value for tooth root capacity.

$$K^* = \frac{F_t}{b d_1} \frac{u+1}{u} \quad (1)$$

$$U = \frac{F_t}{b m_t} \quad (2)$$

where

- K^* is flank load capacity;
- F_t is load in transverse plane, N;
- b is face width, mm;
- d_1 is reference diameter, mm;
- u is gear ratio;
- U is tooth root capacity;
- m_t is transverse plane module.

Since the final geometry has to be rated according to ISO 6336 [10], ISO 6336 is used in the basic step as well. For this purpose, many default values are assumed, e.g., for the K-factors of ISO 6336, tool data and further values. This has the advantage that the default values can be determined in more detail during the design process without leaving the framework of ISO 6336.

With the ratios defined, the torque moment and speed of every stage is determined. Table 1 shows the main geometry of the high speed shaft. The detailed tooth geometry can be defined including manufacturing and heat treatment. A geometry calculation with respect to cutting and grinding tool geometry is necessary. The final geometry includes backlash and detailed tooth root contour. On that basis, a full load capacity calculation according to ISO 6336 is performed. This is easily accomplished with the STplus-suite [8] that includes the geometry calculation and the standard calculation as well.

To cover pitting resistance and risk of tooth root breakage, ISO 6336 is adequate. Further limiting factors can be taken into account e.g., by ISO TR 13989 (scuffing) [11], ISO TR 15144 (micropitting) [12] and further standards like API or ABS. Being available in direct connection to the geometry calculation makes the application of these additional methods easy and the process straightforward. Figure 3 shows the tooth form in the mesh and the range of standards that are available for rating the gears in the mesh.

Shafts and bearings

Dimensions of shafts and bearings are covered only by rough estimate at first. A detailed calculation has to follow to ensure sufficient load capacity. To determine the influence on the contact conditions in the mesh, the deformation of the shaft-bearing system has to be calculated.

Table 1. Main gear data

	High speed stage	
	Pinion	Gear
Number of teeth, z	35	138
Gear ratio, u	3.943	
Normal module, m_n , mm	7.5	
Center distance, a , mm	695.0	
Normal pressure angle, α_n , degrees	20.0	
Helix angle, β , degrees	13.0	
Profile shift coefficient, x	0.4020	0.2267
Tip diameter, d_{a1} , mm	292.809	1082.518
Face width, b , mm	235.0	220.0
Transverse contact ratio, ε_α	1.833	
Overlap ratio, ε_β	2.005	
Total contact ratio, ε_γ	3.838	