Calculation of Optimum Tooth Flank Corrections for Helical Gears

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Abstract:

Stricter requirements are being imposed on heavy duty gears in terms of running behavior and load carrying capacity. Furthermore, the expanding techniques in the field of gear manufacturing are demanding the development of suitable designing methods for tooth flank corrections. In order to calculate these corrections, the spatial stress and deformation state in the mesh has to be determined. This article reports on the further development of a finite element calculation method into an optimizing and designing system for flank corrections on spur and helical gears.

Main Influences on the Bearing Capacity and Running Behavior of Gears

The load carrying behavior of gears is strongly influenced by local stress concentrations in the tooth root and by Hertzian pressure peaks in the tooth flanks produced by geometric deviations associated with manufacturing, assembly and deformation processes. The dynamic effects within the mesh are essentially determined by the engagement shock, the parametric excitation and also by the deviant tooth geometry.[1]

The engagement shock results from a displaced starting point of engagement due to rotational deviations or pitch errors within the gear system. This transferred start of engagement is located outside the plane of action. Here deviations occur in the value and direction of the normal velocity components of the contacting tooth flanks; thus, vectorial difference
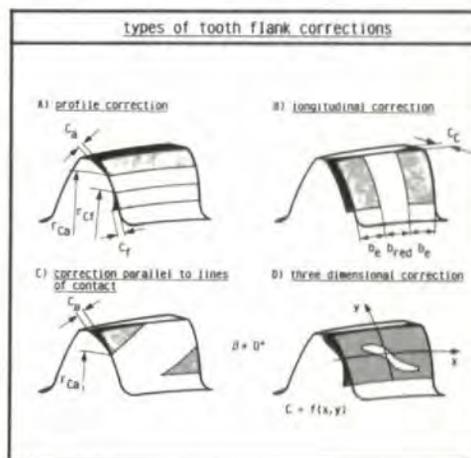

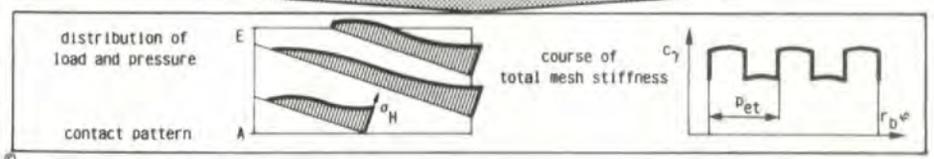Fig. 1— (upper left) Objectives of tooth flank corrections:
- reduction of pressure peaks
- decrease of displacement sensivity
- diminishing of engagement shocks
- lessening of parametric excitation

Fig. 2— (below) Basic principles of contact analysis of loaded gears.

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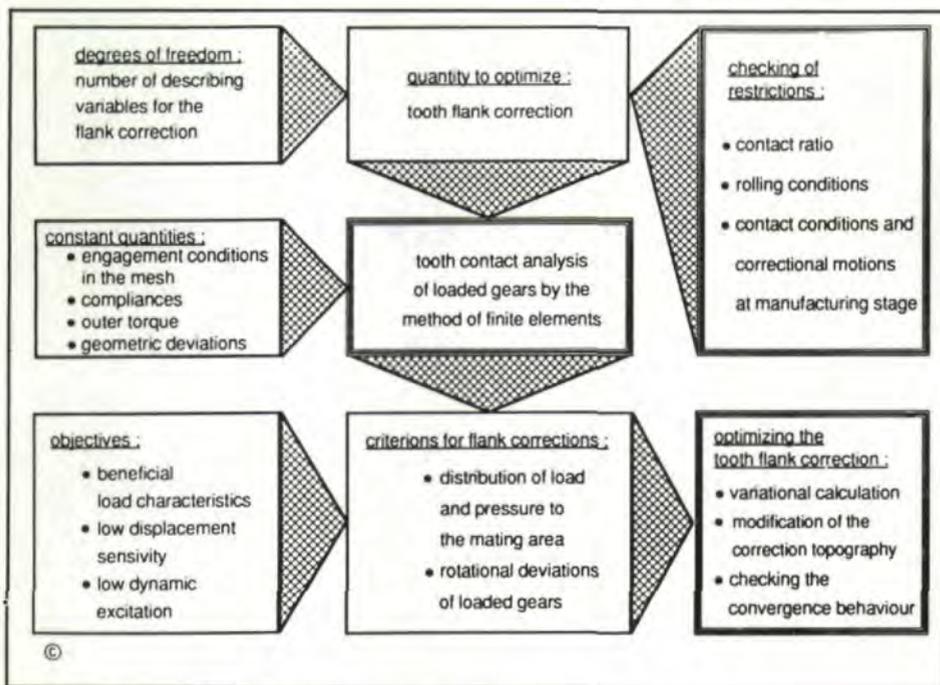
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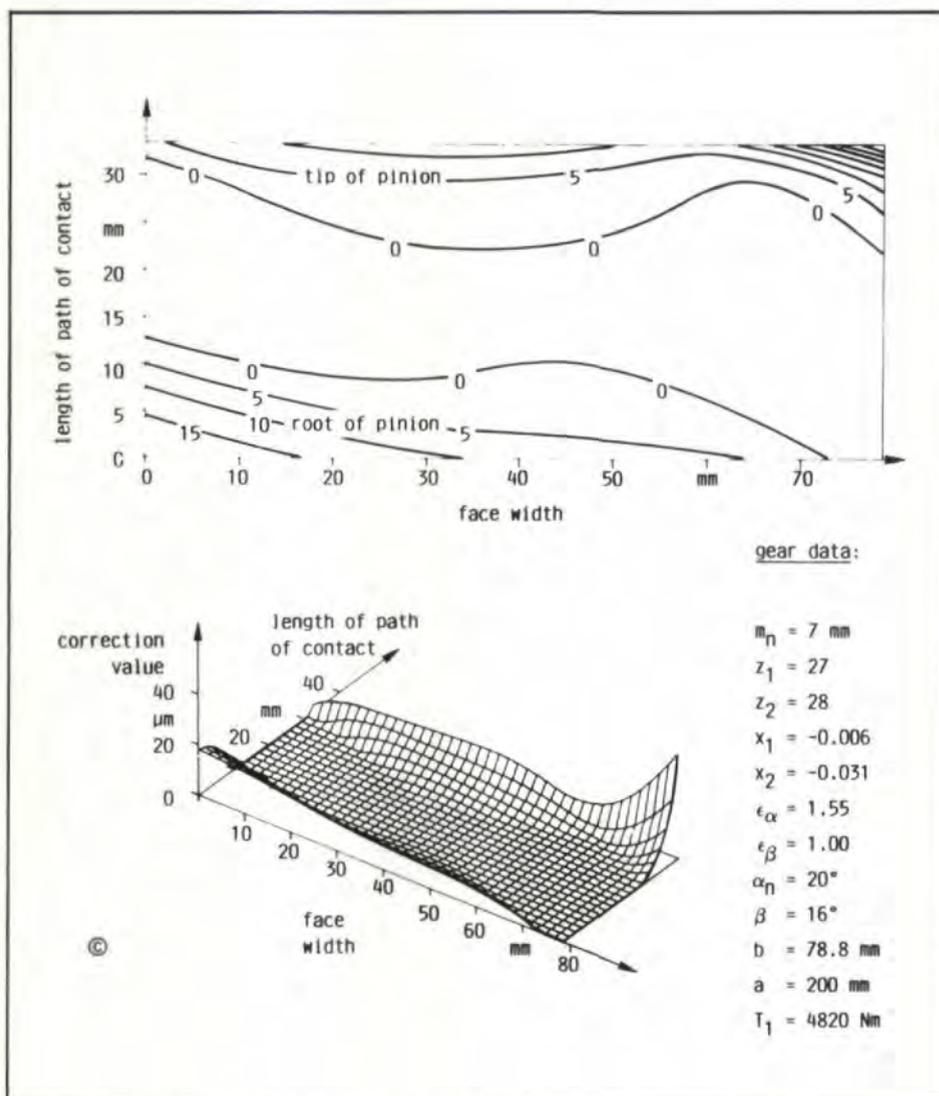
Fig. 3 — Optimization of tooth flank corrections.

brings about an unwanted impact speed and shock activation.[2]

The parametric excitation is a consequence of the changing number of mating teeth and the contact line movement over the tooth flank during engagement. As a result, a periodic modulation of the mesh stiffness follows and induces rotational deviations and activates unwanted vibrations, even when the outer load for the gear is constant.[3]

Geometry of Flank Corrections for Helical Gears

The flank geometry of helical gears can be modified in various ways with respect to manufacturing requirements and the differential improvements on the bearing capacity and dynamic behavior. Fig. 1 gives an overview of the principle correction forms.

Flank corrections in the direction of tooth depth as shown in Fig. 1a are simply carried out by involute tip or root reliefs. In this case, the part of the flank to be taken back consists of a corrected involute profile, which is defined by a modified base circle diameter and by the intersection point with the uncorrected involute curve. This is achieved by tools having a basic rack system with altered profile angles or by using a modified working pitch diameter at generation. Another form of profile correction which has a smooth transition to the original, uncorrected involute curve is manufactured by tools with crowned basic rack profiles. Profile corrections are mostly used to decrease the engagement shock and the in-

gear data:

$m_n = 7$ mm
$z_1 = 27$
$z_2 = 28$
$x_1 = -0.006$
$x_2 = -0.031$
$\epsilon_\alpha = 1.55$
$\epsilon_\beta = 1.00$
$\alpha_n = 20°$
$\beta = 16°$
$b = 78.8$ mm
$a = 200$ mm
$T_1 = 4820$ Nm

Fig. 4 — Optimized three-dimensional correction with basic gear data.

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PROF. DR. — ING. MANFRED WECK holds the university chair of Machine Tools at the Technical University Aachen, West Germany. He is also director of the Laboratory for Machine Tools and Industrial Management (WZL) and manager of the machine tool department of the Fraunhofer Institute for Production Technology (IPT) in Aachen. Prof. Weck's professional interests are in the areas of machine testing and evaluation, machine design and calculation, development of automatic control and monitoring systems, automation of production and handling and gear cutting machines and gears.

DIPL. — ING. GEORG MAUER received his degree in mechanical engineering at the Technical University of Aachen and is currently working as a scientific assistant in the gear research group of WZL.

volved strain and noise.

Longitudinal flank corrections in the direction of the face width (Fig. 1b) are used to attain a low displacement sensitivity and to avoid strain peaks occuring due to displaced positions of the gear axis (twisting, tilt) or helix deviations of the flanks.

For helical gears, a correction running parallel to the lines of contact is advantageous. (See Fig. 1c.) The intersection line of the corrected and uncorrected area is identical to a line of contact between the mating teeth. The maximum correction value is found at the end or starting point of the engagement. The corrections are carried out by helical involute areas which are mathematically defined by a modified base circle and an additional incremental change made in the helix angle. This form of flank correction has the advantage of producing a smaller loss in the contact ratio, so that the geometric correction may have a greater rolling length and more beneficial normal vector conditions.

The correction forms described above are to a certain extent limited in optimizing the running behavior and load carrying capacity, because the flank corrections do not meet the gear geometry and mating conditions adequately. These conditions can be met by using three dimensional (often called topologic[4]) flank corrections (See Fig. 1d.), which are characterized by variable correction forms in both directions of tooth depth and face width.[5] There are, however, various marginal conditions which have to be taken into consideration, these being the continuity of differentials, minimum contact ratio, rolling conditions and manufacturing processes.[1]

Optimizing Method Based on Finite Element Calculations

To obtain the required starting position to optimize flank corrections, one has to return to the basic principles of contact analysis of loaded gears illustrated in Fig. 2. The continuous, spatial load deformation problem is replaced by a system of discrete contact points, defined by a matrix with deformation indicies $[\alpha]$, a load vector $\{F\}$ and a vector containing contact distances $\{s\}$, which describes the geometric flank deviations or corrections. The equation system is then set up for different rolling positions and is solved by considering the marginal conditions concerning the total load and the rigid body
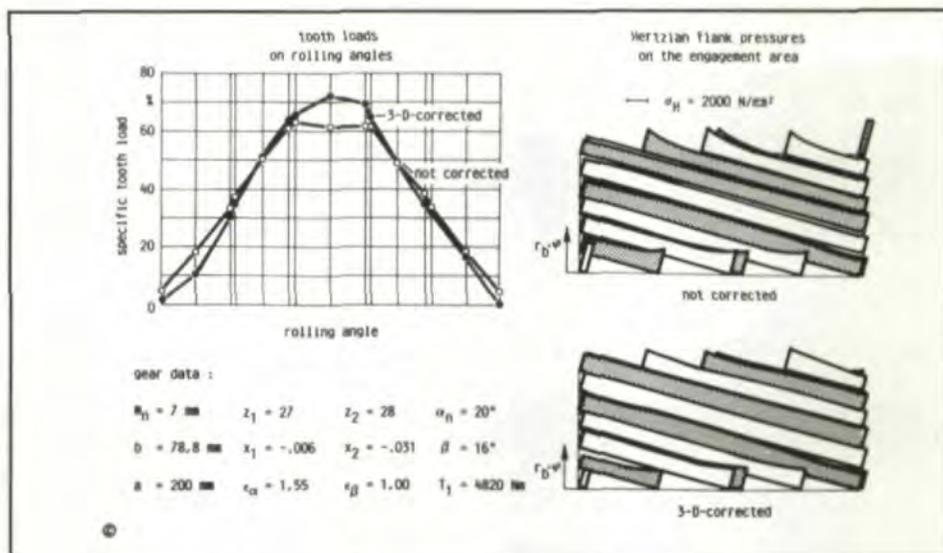

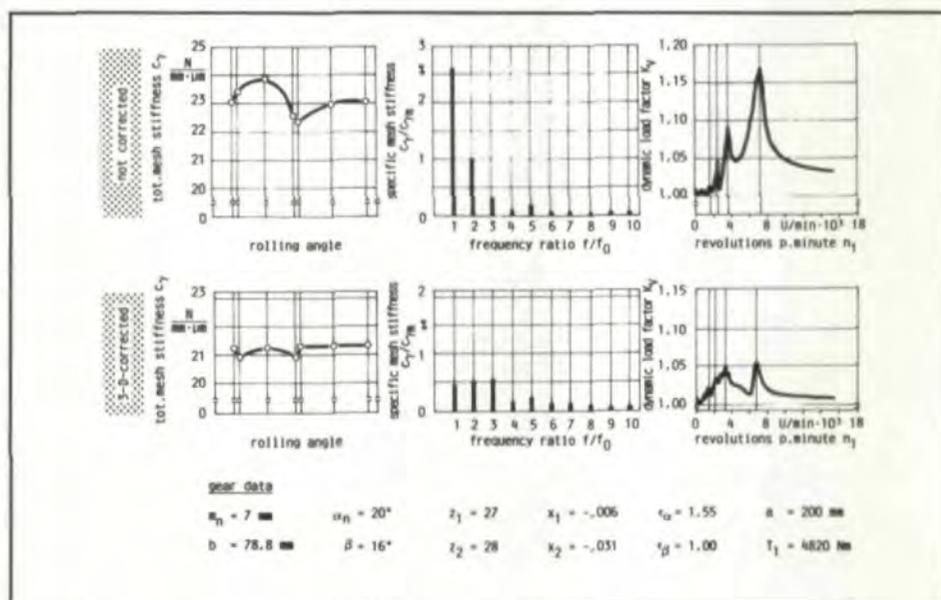Fig. 5 – Calculated tooth loads for a full cycle of meshing contact.

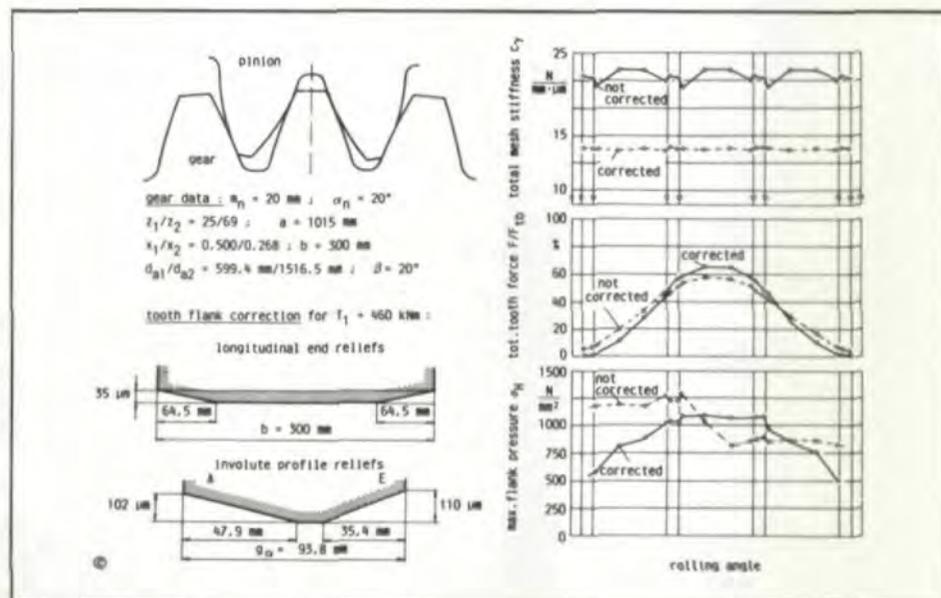Fig. 6 – Periodic courses of the total mesh stiffness.

Fig. 7 – Samples of geometrically simple correction forms.

displacement.[6] The results provide information on the load and pressure distribution on the field of action and the shape of the rotational deviation curves, which is influenced by the alternating mesh stiffness. It becomes clear, therefore, that load distribution and mesh stiffness are direct functions of the correction topography. The pliability indicies [α] are calculated by finite element structures of the mesh and analytical models of shafts and bearings.

This contact analysis is the central starting point to optimize tooth flank corrections. (See Fig. 3.) By varying the correction topography, a beneficial load and pressure distribution on the flanks, a low displacement sensitivity and a lower dynamic activation can be achieved.

The contact and deformation analysis, as described above, presents all the criteria necessary to assess a particular correction with regard to the various objectives. The mathematical variation provides information regarding the trends and dependent variables of the topography and the target objectives, so that the correction can be improved in a stepwise fashion. This numeric process has to be repeated several times after checking that all the restricting marginal conditions are met.

Other research developments at WZL concern the simulation of grinding processes for corrected tooth flanks, whereby correction movements and contact conditions are analyzed. Control data for NC grinding machines are also generated.[7]

Calculated Examples

The following section contains the calculation results of two flank corrections for typical helical gears in industrial use. Fig. 4 shows an optimized three-dimensional correction with the basic gear data. The correction values are drawn up on the plane of action with a maximum relief of approximately $40\mu m$. In the middle of the engagement area an uncorrected region ensuring a contact ratio greater than 1.0 can be seen.[1] The correction topography is observed to run parallel to the original involute flank at the approach contact, which is favorable for lower excitation produced by engagement shocks.

The calculated tooth loads for a full cycle of meshing contact are illustrated in Fig. 5. A lower force level at the start of engagement and a shallower load reception gradient for the corrected gear can be seen clearly. The pressure distribution on the plane of action depicts lower strains for the corrected gear, and the pressure peaks have been successfully reduced where the contact lines end at the root of one of the mating teeth.

In Fig. 6 the periodic courses of the total mesh stiffness are drawn up on one base pitch. The specific range of variation which is characteristic for the intensity of the parametric excitation could be reduced by the three-dimensional flank correction from 3.3% to 0.9%. The simulation of the dynamic behavior of the gear at various rotational speeds shows that the maximum dynamic load factor K_v could be decreased from 1.17 to 1.06. This is achieved by depressing the maximum resonance peaks as well as the first and second order Fourier coefficients of the

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mesh stiffness course.

Clear improvements of the dynamic and bearing behavior can be achieved also by geometrically simple correction forms. (See Fig. 7.) Here the results of the tooth contact analysis of a loaded helical gear for the corrected and the uncorrected case are illustrated. The tooth flank correction consists of longitudinal end reliefs and two involute profile corrections. The relief lengths and values were optimized in the method described previously.

The corrected gear shows several optimized characteristics. These are
* a much smoother mesh stiffness curve with a smaller specific variational range,
* a reduction of the force levels at the beginning and end of engagement,
* a lower load reception gradient,
* a well-balanced and lower pressure course.

The unmodified area on the middle flanks of the corrected gear is sufficient to ensure that a minimum contact ratio of 1.10 is achieved; thereby guaranteeing a smooth operation, free from kinematic and dynamic disturbances.
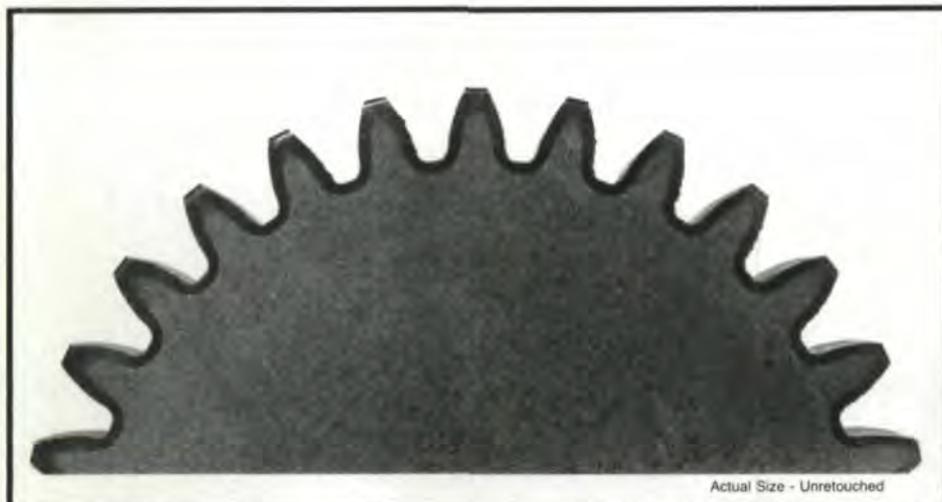
Conclusions

A method is given for optimizing various forms of tooth flank corrections with regard to the load carrying capacity and the running behavior of spur or helical gears. Two examples for corrected gears confirm that it is possible to optimize flank corrections at the designing stage and to make empirical attempts unnecessary.

Even when flanks are corrected by simple geometric modifications which do not require advanced manufacturing methods, the gear characteristics can be improved so that an increased transferable power and a reduction in noise emission is achieved. A more complex correction topography allows for an optimum conformation with the gear geometry and the engagement conditions, while, at the same time, keeping the correction topography within manufacturing restrictions.

It becomes clear that this new calculation makes it possible to influence all essential effects which determine the dynamic and load carrying behavior in a specific way.

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