A Modular Approach to Computing Spiral Bevel Gears and Curvic Couplings

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In general, bevel gears and curvic couplings are completely different elements. Bevel gears rotate on nonintersecting axes with a ratio based on the number of teeth. Curvic couplings work like a clutch (Fig. 1).

Computing these different elements in the same manner is based on the idea that curvic couplings are actually a special kind of plunge-cut spiral bevel ring gear with a pitch angle of 90°, a spiral angle close to 0° and a constant tooth depth. This principle allows curvic couplings to be computed like spiral bevel gears.

The Motion Concept

Nearly all gearing systems are based on the idea of making teeth by generating the tooth with a tool that has a straight profile. This allows smooth motion for both the tool and the workpiece. A straight rack is used to make cylindrical gears. The workpiece constantly rotates while the rack moves linearly at a constant speed.

Fig. 1—Spiral bevel gears and Curvic Couplings.

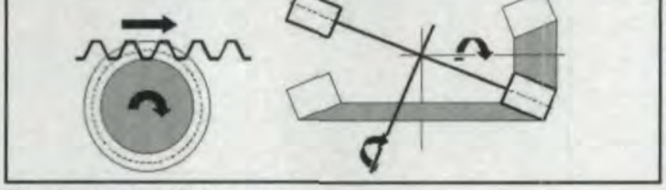

Fig. 2—Rack and plane gear.

$$\alpha = \alpha_1 \ldots \alpha_m \ldots \alpha_2$$

$$\beta = a_\beta + b_\beta(\alpha - \alpha_m) + c_\beta(\alpha - \alpha_m)^2 + \ldots + g_\beta(\alpha - \alpha_m)^6$$

$$\chi = a_\chi + b_\chi(\alpha - \alpha_m) + c_\chi(\alpha - \alpha_m)^2 + \ldots + g_\chi(\alpha - \alpha_m)^6$$

$$\varepsilon = a_\varepsilon + b_\varepsilon(\alpha - \alpha_m) + c_\varepsilon(\alpha - \alpha_m)^2 + \ldots + g_\varepsilon(\alpha - \alpha_m)^6$$

$$\tau = a_\tau + b_\tau(\alpha - \alpha_m) + c_\tau(\alpha - \alpha_m)^2 + \ldots + g_\tau(\alpha - \alpha_m)^6$$

$$\gamma = a_\gamma + b_\gamma(\alpha - \alpha_m) + c_\gamma(\alpha - \alpha_m)^2 + \ldots + g_\gamma(\alpha - \alpha_m)^6$$

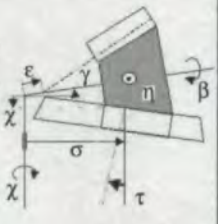

Fig. 3—Taylor approach for motion concept.

Bevel gears are made using a plane gear instead of a rack. The plane gear has a straight tooth profile, and the workpiece and the tool rotate constantly according to their number of teeth (Fig. 2).

The existing spiral bevel gearing systems mainly differ in the tooth lengthwise form. Typical Klingelnberg or Oerlikon gears have an epicycloidal lengthwise shape and constant depth. Typical Gleason gears have tapered depth and an arc in the lengthwise direction. The Palloid system creates gears with an involute profile in the lengthwise direction and constant depth. The tooth lengthwise form is defined by the lengthwise shape of the tool's teeth. In the case of a Gleason gear type, the tool is a cutter with blades arranged in a circle. In the case of a Klingelnberg or Oerlikon design, the tool is a cutter with several blade groups interfering to adjacent gaps while cutting. The involute lengthwise form of a Palloid design is achieved using a tapered hob. Basically, the principle of generating with a straight tooth profile tool is similar.

For achieving modifications to the flank topography, additional motions are used along with those of the basic principle. The amount of these additional modifications is small in comparison to the basic motion. In order to allow for all possible motions between workpiece and tool, an effective nonrestricting mathematical approach for the kinematics of the tool the and workpiece is needed.

The basic description of the motion is developed using a Taylor series. The independent variable is the rotation of the plane gear called a cradle. All other motions depend on the cradle angle α, as shown in Figure 3. A Taylor series is easy to handle. The effect of higher order coefficients has a clear effect on flank topography. For optimizing a gear set, these coefficients are a very powerful tool affecting flank shape. Using several coefficients, the tendency toward oscillation becomes a severe disadvantage for a Taylor series. By finding a good compromise between real free-form motions and handy coefficients correlating to modifications on the flank shape, a new approach is introduced.

For user driven modifications, we keep the Taylor coefficients; for automatically driven modifications, we superimpose cubic splines. The motion concept is a Taylor series plus real free-form additions based on cubic splines S(x).

The NeutralData approach (Fig. 4), a mathematical description of the relative motion between workpiece and tool, allows free-form capabilities to be combined with the well known Taylor coefficients. The classical way to do this is based on a conventional hypoid machine with the generating motion expressed by a

Taylor series up to order 4 for any of the axes. Basic settings introduced by Gleason use this approach. Since any relative motion of workpiece and tool can be described, this approach supports all existing spiral bevel gearing systems.

The Modular Approach

In designing a software tool for computing spiral bevel gears and curvic couplings, the first step is to divide the task into several independent modules (Fig. 5). If a single algorithm has to match all topics in the computations, it will be impossible to make it user friendly and effective. The presented approach divides the task into several steps to be done one after the other.

Designing a gear set always starts with the basic data like outside diameter, number of teeth, offset, etc. The first step is to design the blank geometry and the initial settings for the motion of gear and tool. This basic geometry is then used to compute the load capacity according to standards like DIN or AGMA. The next step is to optimize the running behavior by modifying the tooth flank topography. The characteristic to be improved is the Ease-off, representing the minimum contact distance for meshing pinion and gear without motion error. While designing the Ease-off, the blank geometry is not touched. The effects of modifying the flank shape influence the stress behavior. This is not taken into account by DIN or AGMA standards and needs to be computed by either a Finite Element or Boundary Element approach performed after optimizing the flanks.

Dimension Computation

Designing a spiral bevel gear set begins with defining the size of the gear set and its ratio. To make sure that the basic geometry will meet the load carrying capacity, standards like DIN or AGMA are used. Based on the load and the basic geometry standards, compute the safety coefficients. This approach makes sure that the size of the gear set is close to what it needs to be for the desired load carrying capacity. These standards are very reliable since they are calibrated by many test runs. The standards for calculating the load carrying capacity of curvic couplings have to consider that the teeth don't mesh.

The initial data on a gear set is either the basic geometry or the power to be transmitted. Starting with the basic geometry, the load carrying capacity can be analyzed by calculating the standards. Beginning with the load, the standards will come up with the size of the gear set. It is up to the design engineer to find a compromise between minimizing the gear size and maximizing the load carrying capacity. The result of the dimension computation is the blank geometry including all details of the tooth shape.

Ease-Off Design

Designing an Ease-off is the most important task in computing spiral bevel gears. The Ease-off is a graphical representation of the crownings on the pinion and the ring gear. Mathematically, the Ease-off is computed by calculating the minimum distance while meshing the teeth according to the number of teeth on each component. Beside surface effects on the flank, the Ease-off contains all the information on the running behavior of the gear set. As a result, the contact pattern and the motion curve can be computed. Computing the Ease-off for curvic couplings is quite simple since the teeth don't mesh.

$$\alpha = \alpha_1 \ldots \alpha_m \ldots \alpha_2$$
$$\beta = a_\beta + b_\beta (\alpha - \alpha_m) + c_\beta (\alpha - \alpha_m)^2 + \ldots + g_\beta (\alpha - \alpha_m)^6 + S_\beta(\alpha)$$
$$\chi = a_\chi + b_\chi (\alpha - \alpha_m) + c_\chi (\alpha - \alpha_m)^2 + \ldots + g_\chi (\alpha - \alpha_m)^6 + S_\beta(\alpha)$$
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$$\gamma = a_\gamma + b_\gamma (\alpha - \alpha_m) + c_\gamma (\alpha - \alpha_m)^2 + \ldots + g_\gamma (\alpha - \alpha_m)^6 + S_\gamma(\alpha)$$

Fig. 4—NeutralData approach.

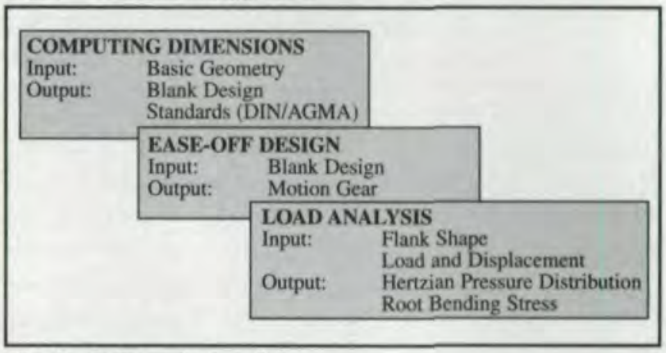

Fig. 5—Independent modules.

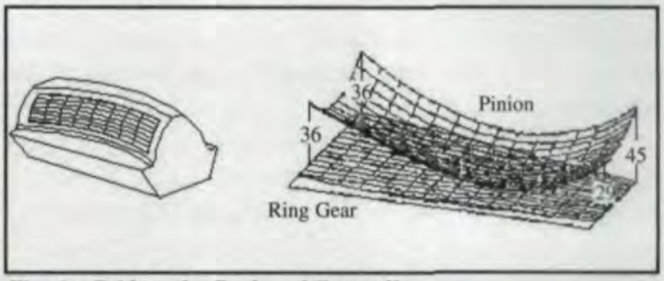

Fig. 6—Grid on the flank and Ease-off.

An Ease-off is computed based on a grid covering the pinion and ring gear teeth as shown in Figure 6. For every point on the ring gear's grid, the minimum contact distance to the meshing pinion's tooth is calculated and displayed graphically. To optimize the Ease-off, a numerical representation of the graph shown in Figure 6 is needed. The principle is to extract independent components having a clear influence to the contact pattern and to the motion curve.

Figure 7 shows the five basic components of an Ease-off. Profile crowning limits the contact pattern in its height and affects the angle between the contact path and the pitch line. Increasing the profile crowning makes the gear set less sensitive to deflections but increases the motion error. Lengthwise crowning changes the contact pattern length. With less lengthwise crowning, the gear set will run more smoothly. However, sensitivity to deflection will increase and the load carrying capacity will be spoiled.

Changing the pressure angle difference will move the contact in a vertical direction. Changing the spiral angle difference will move the pattern in a horizontal direction. The longitudinal twist changes the angle of the contact path, the so-called bias angle.

When designing the Ease-off, enter the numerical values of the crowning, angle difference and twist. The algorithm then changes the relative motion between workpiece and tool to achieve the desired Ease-off. Since there are many parameters in NeutralData affecting the motion, the program will ask the operator which parameter to use. Changing, for example, the spiral

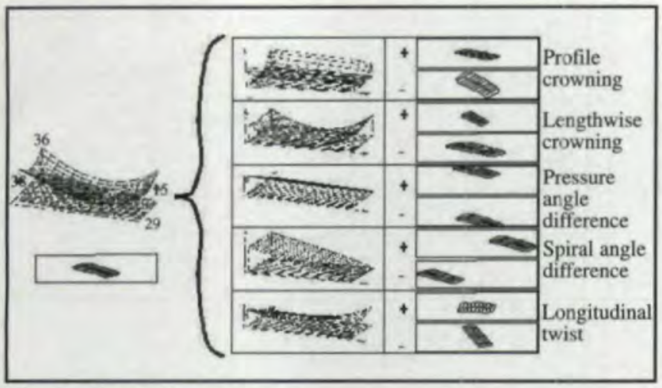

Fig. 7—Ease-off components.

angle difference can be done by modifying the Radial Distance φ or by modifying the machine root angle γ (including a subsequent modification of the Sliding Base χ) or by changing the tool's inclination defined by the Tilt τ and the Swivel σ. For some applications there are restrictions in the root angle, which is not allowed to be changed even by a very small amount. If the user does not have a chance to enable or disable parameters for optimization, the program is not usable. The selection of the parameters to be used is a compromise between an operator driven approach and a fully automatic approach that does not permit user intervention.

The Ease-off optimization changes the motion between workpiece and tool without affecting the blank geometry. Nothing related to dimension computation is changed by this procedure.

Load Analysis

The Load Analysis is the most time consuming part of the software package. A rough assumption on the load carrying capacity is performed during load analysis. At this moment, only the size of the gear and its basic geometry are used for computing the safety coefficients that tell the operator whether or not the gear size is good.

Computing the stress behavior with the theoretical shape of the flank is a very precise tool. The characteristics of the Ease-off are taken into account, and experience with a lot of gear sets shows the potential benefits of this approach.

Based on the deflection, the torque and the flank shape, a calculation is made to compute the deflection of the teeth under load. One of the results is the Hertzian pressure distribution. Depending on the load, the growth of the contact pattern could rise with peaks in the pressure when the contact extends to the edge of the teeth. The maximum pressure will be very high in these areas. If this result occurs, the profile crowning needs to be increased. This will increase the average level of the pressure, but since the contact will not extend to the edge, the maximum pressure can be decreased. This effect can be demonstrated by optimizing the design of a passenger car gear set.

In Figure 9, the Ease-off, contact pattern and motion curve are shown. The Ease-off has a strong lengthwise crowning and zero profile crowning. The contact path is very steep while the transmission error is about 31 µrad.