

New Developments in Design, Manufacturing and Applications of Cylkro–(Face) Gears

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American Gear Manufacturers Association



TECHNICAL PAPER

New developments in Design, Manufacturing and Applications of Cylkro–(Face) Gears

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[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 geometry of both Cylkro–gear and pinion are calculated and optimized concerning contact ratio, lines of action and contact. By means of F.E.M. analysis and use of a load distribution program, the DIN 3990 (ISO/DIS 6336) calculation methods for bending strength and pitting resistance were adapted to include Cylkro–gear calculations. The method developed is tested on a back test bench.

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**NEW DEVELOPMENTS IN DESIGN,
MANUFACTURING AND APPLICATIONS
OF CYLKRO®--(FACE) GEARS**

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1. INTRODUCTION

During the past 3 years, calculation methods and high quality manufacturing processes for Cylkro-gear (face gear) transmissions were developed at Crown Gear's. This paper is a review of the research done.

The geometry of both Cylkro-gear and pinion are calculated and optimized concerning contact ratio's, lines of action and contact. By means of F.E.M. analysis and use of a Load Distribution Program, the DIN 3990 (ISO/DIS 6336) calculation methods for bending stress and pitting resistance were adapted to include Cylkro-gear calculations too. The method developed is tested on a back to back test bench.

After investigation of different methods for manufacturing, the hobbing, skive-hobbing, honing and grinding processes were implemented. The geometry of the hob and the hobbing process were improved using a Chip Analysis Program. Computer support enabled measuring of Cylkro-gears and feed back of results to new machine settings.

Certain features of the Cylkro-gear transmission, such as axial freedom of the pinion and shaft angles from 0° to 110°, create numerous new applications and possibilities.

2. GEOMETRY CYLKRO-TRANSMISSION

2.1. Introduction

The Cylkro-transmission is an angular gear transmission (generally a right-angle drive) consisting of an involute cylindrical pinion and a Cylkro-gear (face gear). The geometry of the pinion, the position of the pinion with regard to the axis of rotation of the Cylkro-gear and the transmission ratio determine the geometry of the Cylkro-teeth. Y.-J.D. Chen [1] derived the mathematical description for the gear meshing and the resulting face gear tooth surface for offset and non-orthogonal face gear drives. D. Overdijk [2] derived the face gear geometry for any combinati-

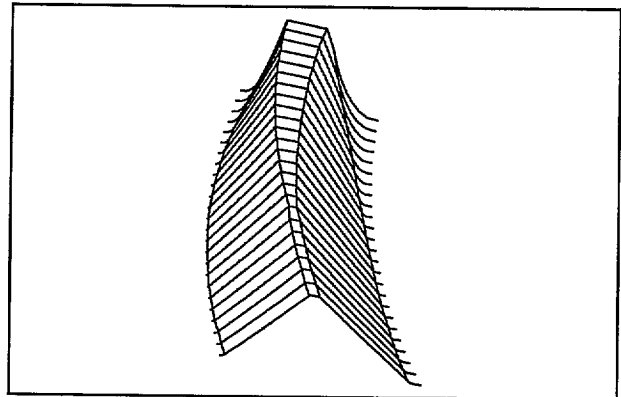


Fig. 1: Helical Cylkro-tooth

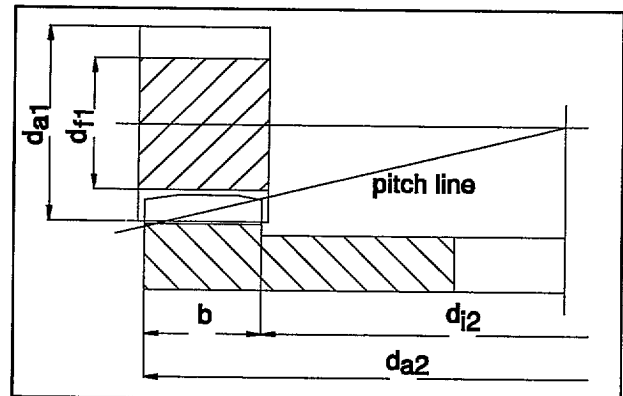


Fig. 2: 90° Cylkro-transmission: tip diameter pinion: d_{a1} ; root diameter pinion: d_{f1} ; inner Cylkro-diameter: d_{f2} ; outer Cylkro-diameter: d_{a2} ; face width: b

on of helix angle of the pinion, offset distance and shaft angle. These formulas have been applied to develop a computer program. Figure 1 shows the Cylkro-tooth that meshes with an offset pinion with helical teeth (hypoid transmission). Generally a spur pinion, on-center and with a shaft angle of 90° is applied. In this article the latter gearing is discussed [Figure 2].

2.2. Lines of action

A cross-section of the Cylkro-transmission can be simplified by regarded as the meshing of a pinion with a "rack" [Figure 3] (the profile of the Cylkro-tooth in a cross-section deviates from a straight line and is therefore not a rack). With a velocity V_{b1} on the base diameter d_{b1} of the pinion, the "rack" with pressure angle α_2 has a horizontal velocity V_{hor} :

$$V_{hor} = \frac{\omega_1 \times d_{b1}}{2 \times \cos(\alpha_2)} \quad [1]$$

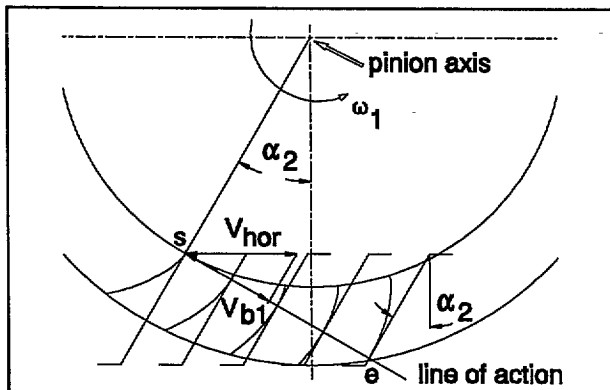


Fig. 3: Line of action: pressure angle rack: α_2 ; horizontal velocity: V_{hor} ; base diameter velocity: V_{b1} ; start point action: s; end point action: e

The horizontal velocity of the Cylkro-gear at the outer diameter (d_{a2}) is higher than at the inner diameter (d_{i2}), so the pressure angle of the "rack" at the outer diameter (α_{a2}) is greater than the pressure angle of the "rack" at the inner diameter (α_{i2}) [Figure 4].

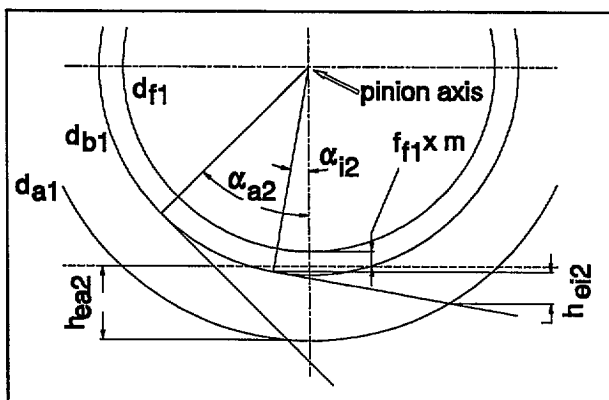


Fig. 4: Lines of action and active Cylkro-tooth depth: inner Cylkro-diameter: α_{i2} , h_{a12} ; outer Cylkro-diameter: α_{a2} , h_{a2} ; maximum Cylkro-tooth depth: line -----

Each diameter of the Cylkro-gear d_2 has a corresponding pressure angle of the Cylkro-toothflank α_2 whereby:

$$d_2 = \frac{m \times z_2 \times \cos(\alpha_n)}{\cos(\alpha_2)} \quad [2]$$

m Module mm
 z_2 Number of teeth of the Cylkro-gear -
 α_n Normal pressure angle of the pinion degrees

The zone of action is a twisted plane built up by straight lines with a pressure angle α_{i2} at the inner diameter and a pressure angle α_{a2} at the outer diameter [Figure 5]. With formula 2 the inner and outer diameters can be calculated with the chosen pressure angles α_{i2} and α_{a2} . These diameters also determine the face width.

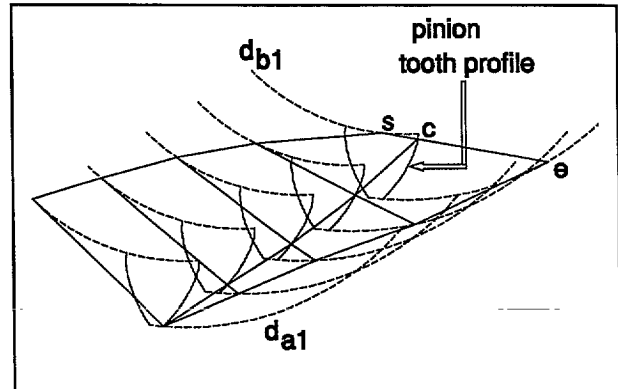


Fig. 5: Zone of action: start line of action: s; line of contact: c; end line of action: e

2.3. Active tooth depth

In figure 4 the active tooth depth h_{a2} of the Cylkro-gear is shown. The depth h_{a2} varies along the facewidth with the angle of the line of action. At the inner diameter the active tooth depth (h_{a12}) of the Cylkro-gear is relatively small but meshes with the full involute profile of the pinion (outer diameter: h_{a2}).

2.4. Fillet of the Cylkro-gear

The fillet of the Cylkro-gear depends on the geometry of the cutting tool and varies along the facewidth. At the inner diameter the fillet is relatively large as compared to the outer diameter. Figure 6 shows three cross-sections of the Cylkro-tooth in which the changes in pressure angle, the size of the active flank and the fillet can be seen.

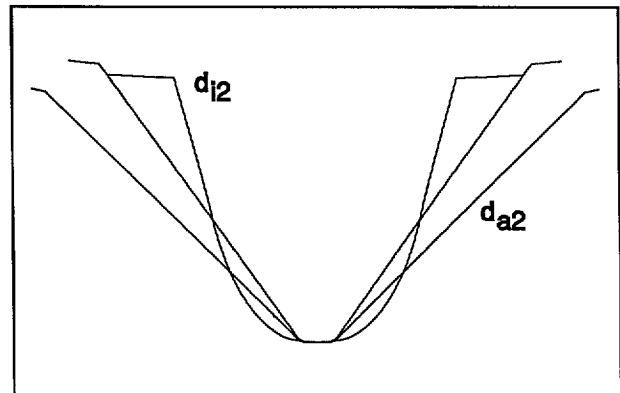


Fig. 6: Cross-sections of the Cylkro-tooth

2.5. Active tooth flanks

Theoretically the active tooth flanks are limited in the direction of the tooth depth by α_{i1} (pressure angle of the pinion at start of fillet) and α_{a1} (pressure angle of the pinion at tip diameter) and along the face width by α_{i2} and α_{a2} . In the following paragraphs the practical limits are described.

2.5.1. Minimum pressure angle Cylkro-gear: α_{i2}

Theoretically the pressure angle α_{i2} may be 0° , in which case the active tooth flank of the Cylkro-gear is reduced to a point. In practice a larger pressure angle is chosen as with smaller pressure angles undercutting of the active tooth flank may occur. Undercutting removes all or part of the active tooth flank, resulting in a shorter line of action. Figure 7 shows the effect of undercutting at the minimum pressure angle of the Cylkro-gear α_{i2} as a function of the number of teeth of the pinion (z_1) and the gear ratio.

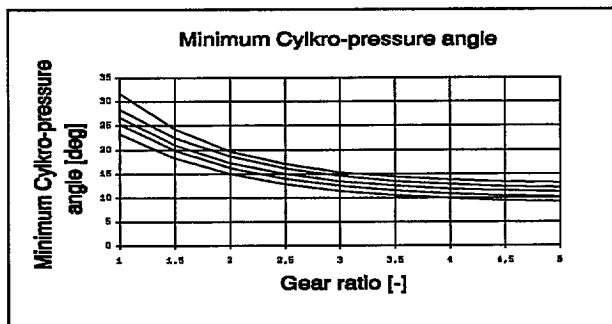


Fig. 7: Minimum Cylkro-pressure angle: from the top downward the lines are corresponding with z_1 (number of teeth of the pinion) = 10, 12, 15, 18 and 24 teeth

2.5.2. Minimum pressure angle pinion: α_{i1}

The pinion may start meshing from the minimum pressure angle of the pinion α_{i1} . The maximum tooth depth of the Cylkro-gear is determined by the required bottom clearance between the tip surface of the Cylkro-tooth and the root surface of the pinion ($f_{r1} \times m$) [Figure 4]. At the smaller pressure angles of the Cylkro-gear, it is possible that the allowable Cylkro-tooth depth will not be achieved with the pressure angle of the pinion α_{i1} . To prevent improper meshing between the dedendum of the teeth of the pinion and the addendum of the teeth of the Cylkro-gear, the tooth depth of the Cylkro-gear has to be reduced at the inner diameter.

At larger pressure angles of the Cylkro-gear, α_{i1} does not mesh with the Cylkro-teeth because of the required tip clearance of the Cylkro-gear and meshing starts at a larger pressure angle of the pinion [Figure 4].

2.5.3. Maximum pressure angle Cylkro-gear: α_{a2}

Overcutting of the active tooth flanks constitutes a practical limit for the maximum pressure angle of the Cylkro-gear α_{a2} . On account of the combination of undercutting and the minimum required tooth thickness at the tip, the tooth depth of the Cylkro-gear will be reduced at the outer diameter. As a guide a maximum pressure angle $\alpha_{a2} \approx 45^\circ$ can be used when the pinion has a large addendum modification.

2.5.4. Resulting active tooth flanks

Figure 8 shows the resulting active tooth flanks: **Pinion:** Generally the active tooth flank is determined by α_{i1} , α_{a1} , α_{i2} and α_{a2} . Local by:

- Line 1 : Undercutting Cylkro-tooth
- Line 2 : Required bottom clearance ($f_{r1} \times m$)
- Line 3 : Overcutting Cylkro-tooth

Cylkro-gear: Generally by: α_{i2} , α_{a2} and required clearances between pinion and Cylkro-teeth ($f_{r1} \times m$, $f_{a1} \times m$). Local by:

- Line 4 : Avoiding contact between dedendum of pinion and addendum of Cylkro-tooth
- Line 5 : Overcutting
- Line 6 : Fillet (or undercutting)

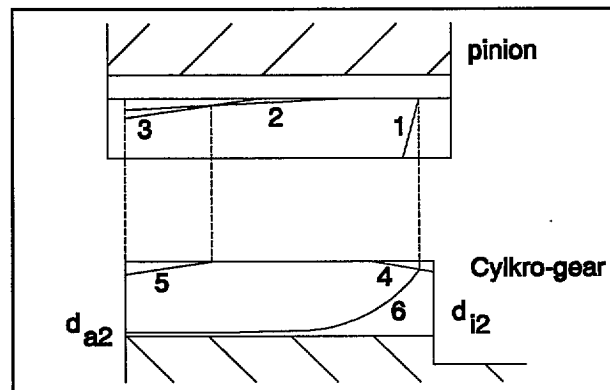


Fig. 8: Active flanks of pinion and Cylkro-tooth

2.6. Lines of contact

Figure 9 shows the contact situation for one position of the pinion at the inner and outer diameter. The point of contact of the Cylkro-flank at the inner diameter is on a smaller radius of the pinion than at the outer diameter. Therefore, the lines of contact are inclined, even when we have a spur pinion. Theoretically every combination of pinion and matching Cylkro-gear has line contact at every instant.

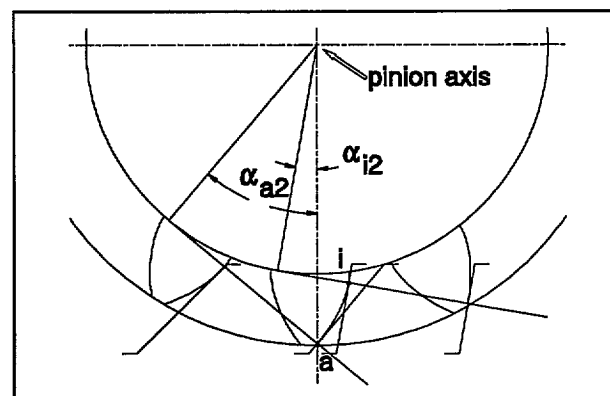


Fig. 9: Line of contact: point of contact at inner Cylkro-diameter: i; point of contact at outer Cylkro-diameter: a

In Figure 10 the lines of contact are shown for different angular positions of the pinion. With a driving pinion, the meshing starts at the tip of the Cylkro-teeth at the outer diameter. The meshing conditions can be derived from this figure.

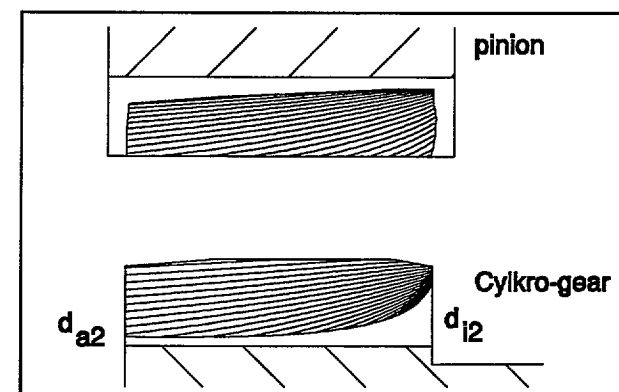


Fig. 10: Lines of contact at pinion and Cylkro-flank