

Evolution of Worm Gear Standards and their Consequences on Load Capacity Calculation Approach

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Beginning with a brief summary and update of the latest advances in the calculation methods for worm gears, the author then presents the detailed approach to worm gear geometry found in the revised ISO TR 10828. With that information, and by presenting examples, these new methods are explained, as are their possibilities for addressing the geometrical particularities of worm gears and their impact upon the behavior and load capacity of a gearset under working conditions based on ISO TR 14521 — Methods B and C. The author also highlights the new possibilities offered on that basis for the further evolution of load capacity calculation of a worm gearset based on load and contact pressure distribution.

Introduction

The geometry of worm gears has been developed over many years by internal convention specific to gear manufacturers, and by various authors who have contributed reference books on the topic (Refs. 1–6), but there has not always been consensus among them regarding definitions/terminology. There was a need for homogeneity and the first step in that direction was standard DIN 3975—published in 1960. To further contribute to a better approach in the prediction of behavior of worm gears in first service, standardized methods have been published concurrently with geometrical definitions included in the same document as the load capacity calculation document (Ref. 7).

At the international level, the Working Group 7 of Sub-Committee One of ISO Technical

Committee 60 in charge of worm gears (ISO TC60/SC1/WG7) began work on this topic in 1992.

Development of the First ISO Technical Report on Worm Gear Geometry

The development of the unification of geometrical symbols of worm gears, and the associated calculation, began in 1992 in conjunction with the discussions over the concept of the load capacity of worm gears. It has since been decided to first develop a document

with definitions of the different profiles of cylindrical worm gears that serves as the basis of the next step in creating the load capacity ISO documents.

The Technical Report ISO TR 10828/ Worm Profiles Geometry was published in 1997. The main target was to describe and provide the basic equations of the axial profiles of the thread profiles for the five most common types of worms (Fig. 1):

Type A: Straight-sided axial profile

Type C: Concave axial profile formed by machining with a convex circular profile disc-type cutter or grinding wheel

Type I: Involute helicoid, straight generatrix in base tangent planes

Type N: Straight profiles in normal plane of thread space helix

Type K: Milled helicoid generated by bi-conical grinding wheel or milling cutter, convex profiles in axial planes

From the axial profile, its projection in different plane was also defined in this TR in order to be able to determine (Fig. 2):

- The normal profile, to determine the shape of the flying cutting tool for machining the worm wheel
- The rack profile in offset planes used to calculate the conjugate worm wheel profiles
- The transverse profile of the worm which can be used for the control

(The decision of the WG to not include in this document all the geometrical definitions of worm gears is because similar work on a load capacity calculation standard is in progress, including the minimum set of geometrical relations requested by this second document.)

Evolution of Load Capacity Calculation Methods for Worm Gears

Due to the complex geometry, the load capacity of worm gears has been developed over a long period on standards based on industrial experience with empirical factors with two main criteria: contact pressure (some time called wear) and bending of threads.

Analytical methods, including geometrical aspects, have been developed with the power of computers, and the first version of DIN 3996 was published in 1998.

ISO TC60/SC1/WG7 developed and published the ISO TR 14521—Calculation of Load Capacity of Worm Gears—in 2010; this standard is based on DIN 3996. The following conditions that can limit the rated load capacity are:

- **Wear:** such damage usually appears on the tooth flanks of bronze worm wheels
- **Pitting:** this form of damage may appear on the flanks of worm wheel teeth. Its development is strongly influenced by the load

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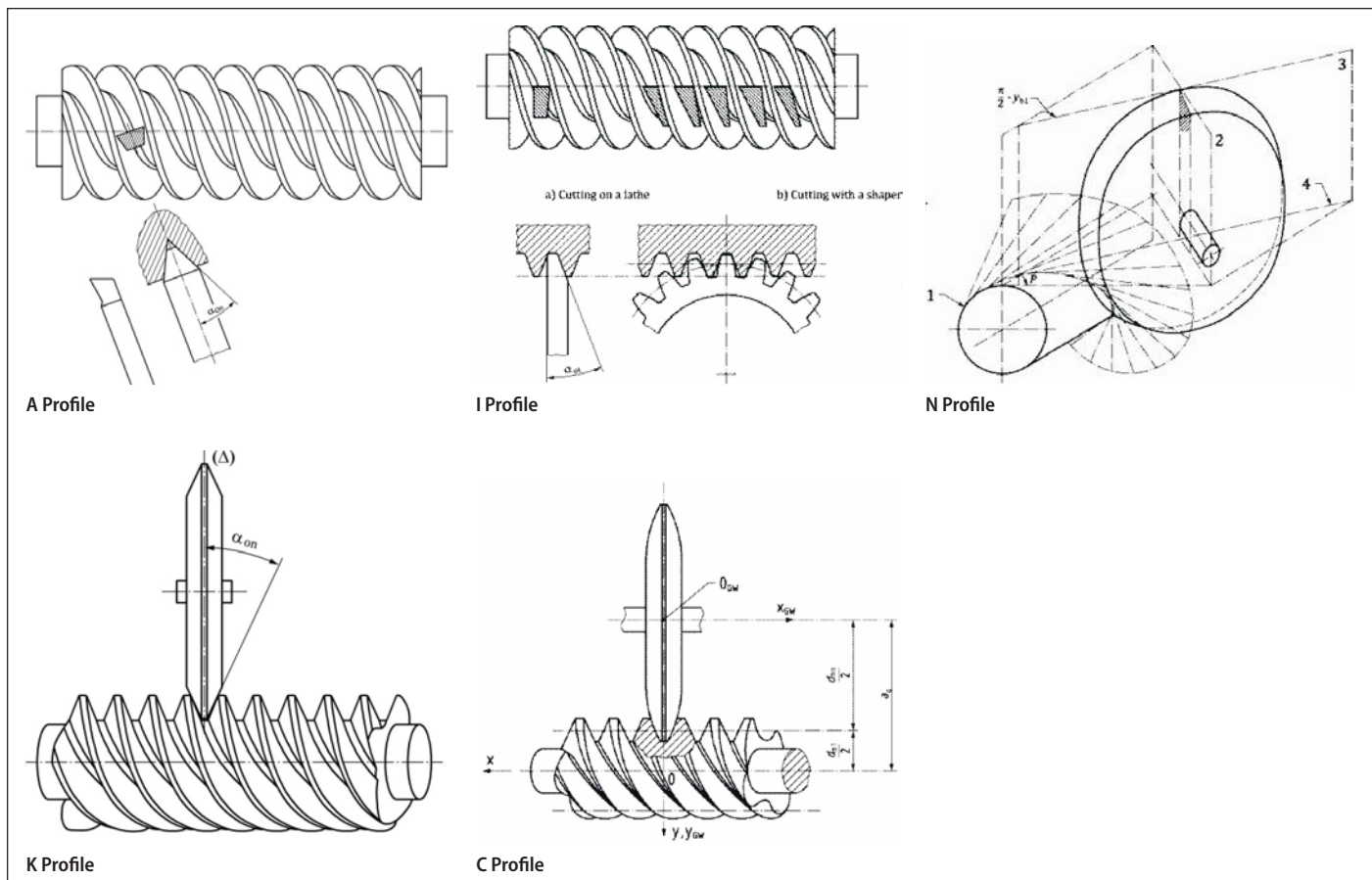


Figure 1 Cylindrical worm gear profiles.

- transmitted and the load-sharing conditions.
- **Tooth breakage:** shear failure of worm wheel teeth or worm threads can occur when teeth become thin due to wear or overload.
 - **Worm thread and worm shaft breakage:** shaft breakage can occur as a result of bending fatigue or overload.
 - **Worm shaft deflection:** excessive deformation under load modifying contact pattern between worm and worm wheel.
 - **Scuffing:** this form of damage often appears suddenly and is strongly influenced by transmitted load, sliding velocities and the conditions of lubrication
 - **Working temperature:** when excessively high working temperature leads to accelerated degradation of the worm gear lubricant.

The calculation of load capacity is based on the following criteria, which are linked as follows:

- Criteria for pitting and criteria for worm shaft deflection are independent of others

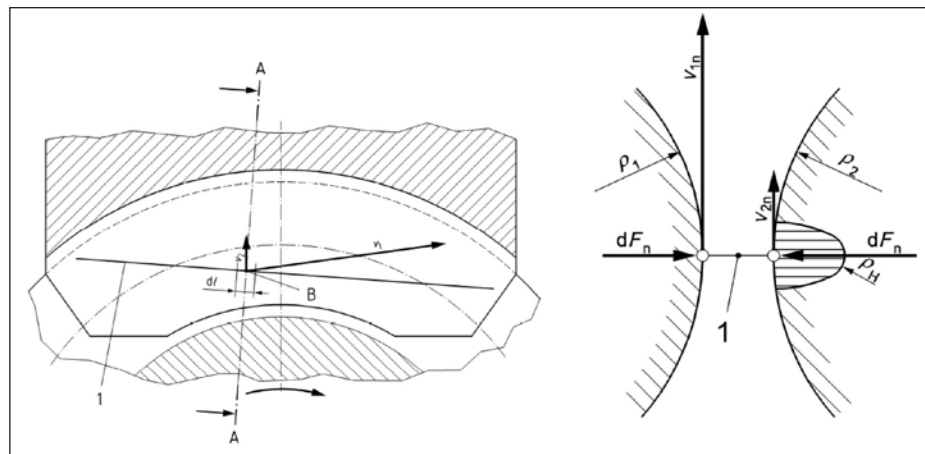


Figure 2 Contact line on a worm wheel and Section A-A normal to the contact line 1.

- Criteria for wear and criteria for tooth breakage of worm wheel are dependent because the reduction of worm wheel tooth thickness by wear is taken into account for the calculation of shear stress at tooth root

Criteria for wear is function of oil film thickness, which depends on oil film temperature, so the calculation of different type of power losses in order to determine the wheel bulk temperature, is the first step for wear calculation

In order to isolate the influence of geometry on the zone of contact and the lines of contact, it has been defined non-dimensional parameters:

- p_m^* = mean Hertzian stress
- h^* = mean lubricant film thickness
- s^* = mean slip path.

To determine those non-dimensional parameters TR 14521 defines two methods:

Method C: Based on simplified formulae which are given in the TR and which are available in a cer-

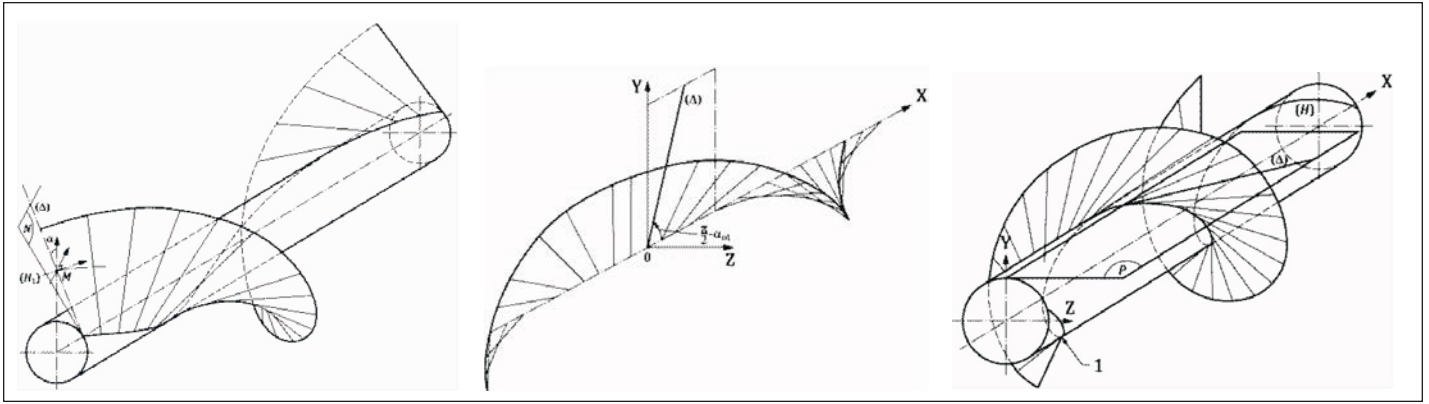


Figure 3 A, I and N profiles generated by a helical cylindrical displacement of a straight line.

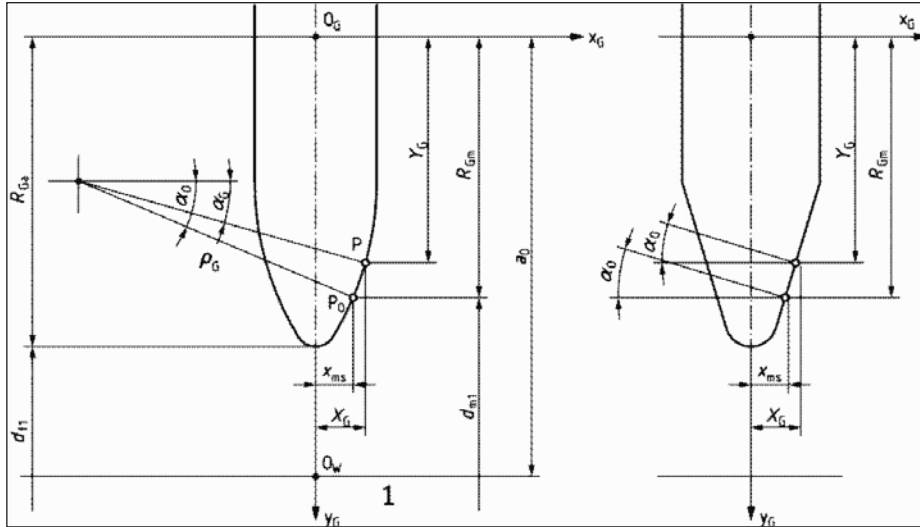


Figure 4 C and K profiles grinding wheel parameters.

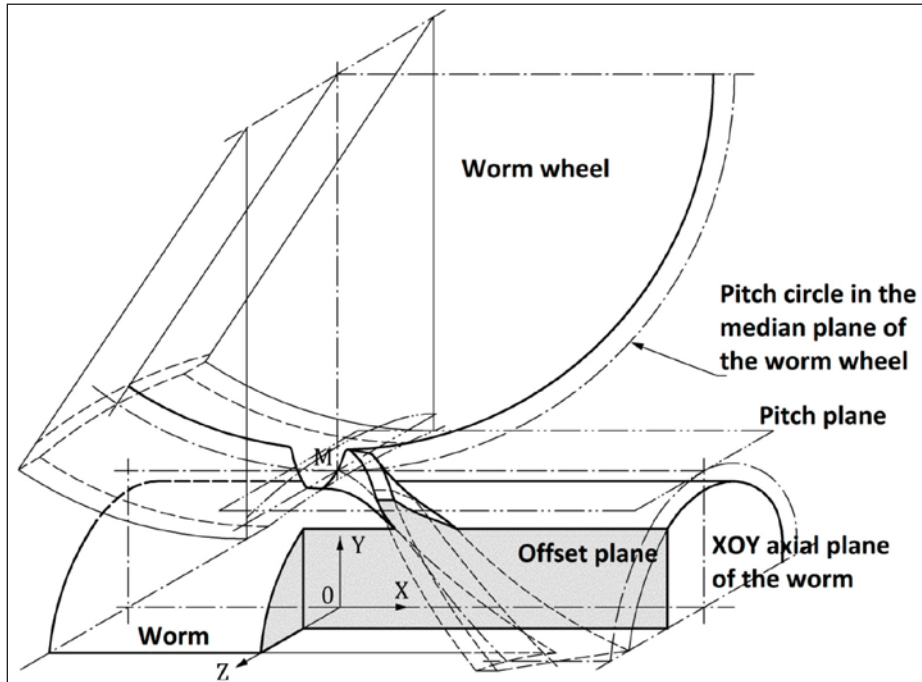


Figure 5 Pitch surfaces and offset plane.

tain range of basic geometrical parameter range: the equations apply to flank form I but can also be used in an approximate manner for tooth forms K, A and N. The approximation formulae for C-worm drives are derived from Bibliography (25) and from operational experience. The approximation equations indicated in 7.3.1, 7.3.2 and 7.3.3 apply to I-worm drives with $\alpha_n = 18... 22^\circ$, $x_2 = -0.5... +1$, $h_{am1} + h_{am2} \approx 2 \cdot m_{x1}$, C-worm drives with $\alpha_n = 20... 24^\circ$, $x_2 = 0... +0.5$, $h_{am1} + h_{am2} \approx 2 \cdot m_{x1}$ and $\rho/m_n \approx 5... 7$.

Method B: based on numerical determination with a detailed study in each point of the lines of contact in terms of radius of curvature, sum sliding velocity in the normal direction, and mean local sliding path (Fig.2).

How to Define a Standard for Geometry of a Worm Gear

Taking into account the complex geometry of worm gearing, and in order to apply Method B for all types of worm profiles, it was important to clarify the application of Method B and to enlarge the scope of TR 10828 to the definition of a complete standard on the geometry of worm gear.

Method B must also be used because the use of geometrical parameters like normal pressure angle, addendum coefficient, face width and external diameter for worm wheel evolve in a significant range from one gear manufacturer to another, and are not sufficiently explained in Method C.

So the revision of ISO 10828 was launched at the end of 2010 with a new

title — “Worm Profiles and Gear Mesh Geometry.”

Content of Revised ISO 10828

First of all, the different geometrical parameters for the worm and the worm wheel, as well as for the pitch surfaces defined previously in the ISO/TR 14521, have been revised and included in the revision. The goal is to delete those definitions in the future revision of ISO/TR 14521 in order to have a unique definition.

The main improvement concerns the following points:

1. Worm profiles have been split into two families, and axial worm profiles have been defined:
 - a. Profiles generated by a helical cylindrical displacement of a straight line that is available for A, I and N profiles (Fig. 3): For those three profiles a general equation has been set up; each profile is accessible by using the correct coefficients. The main parameter of those profiles is the radius in the axial profile of the worm.
 - b. Equations for profiles generated by a profiled disk grinding wheel, as for K and C profiles (Fig. 4): Each profile is accessible by the equation of the grinding wheel profile. The main parameter of those profiles is the radius of the axial profile of the grinding wheel. There is not a direct relation to obtain the axial profile of the worm.
2. In order to define the conjugate profile of the worm — i.e. — the worm wheel profile, the path of contact and the lines of contact — the rack profile of the worm in an offset plane is

determined from the axial profile. It means that the gear mesh of a worm gear set is studied in several offset planes parallel to the median plane of the worm wheel crossing the axis of the worm. It can be noticed that for the rack profile of the worm, the two flanks are never symmetrical. On one flank the pressure angle is increasing, while on the other the pressure is decreasing (Fig. 5).

3. With the definition of pitch surfaces:

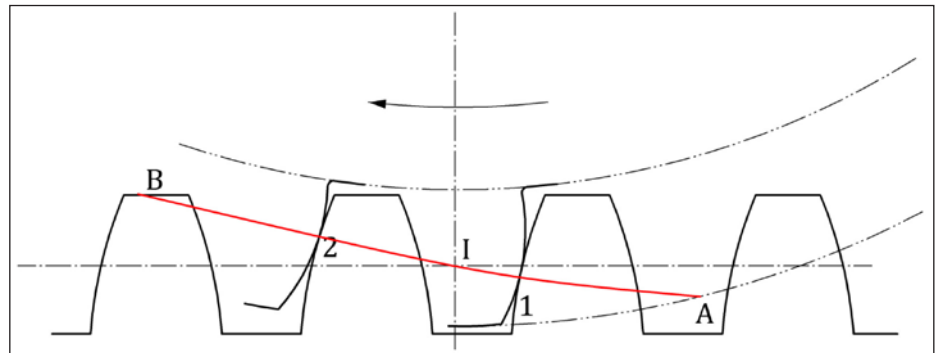


Figure 6 Active path of contact AB.

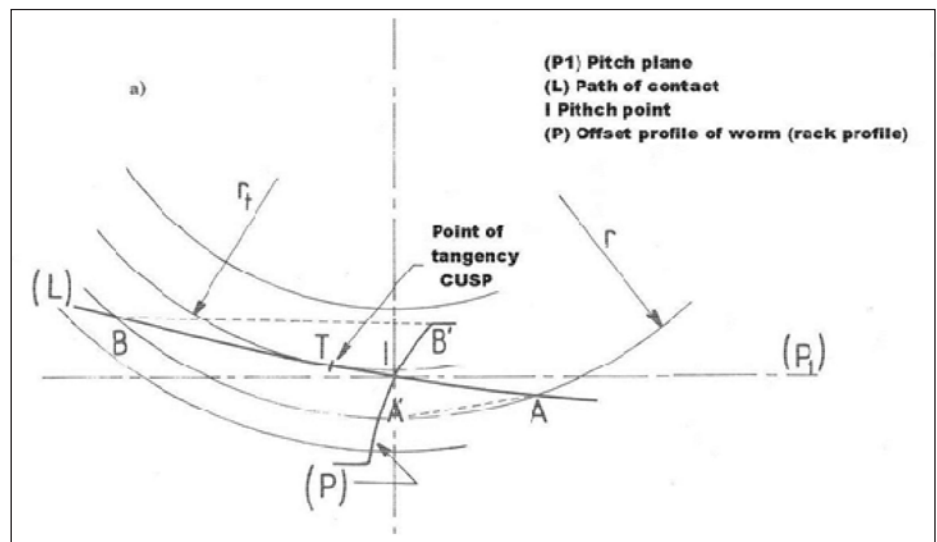


Figure 7 Cusp effect to limit path of contact to AT.

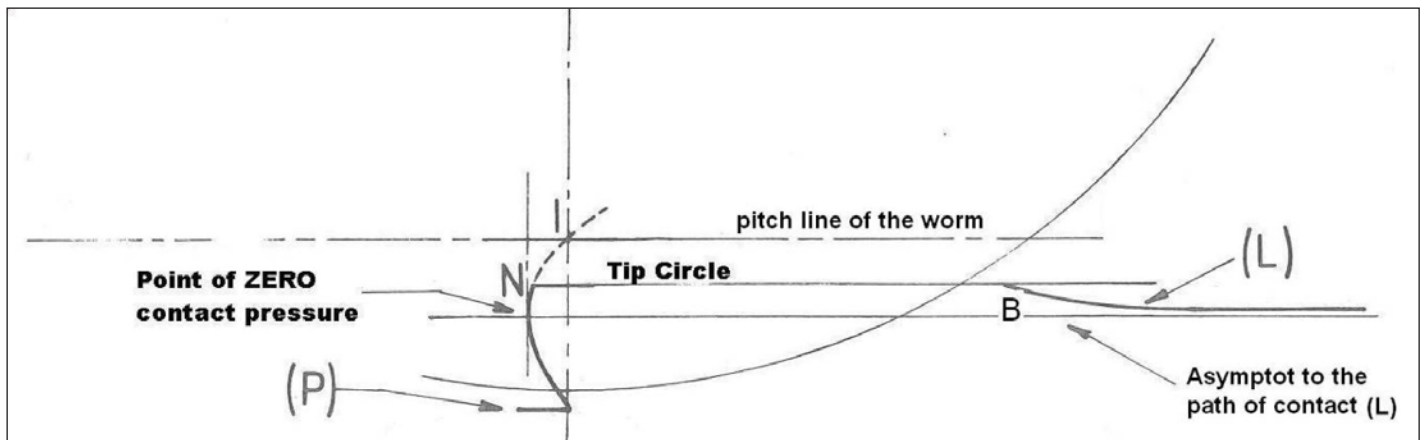


Figure 8 Limitation of gear mesh by the point of zero contact pressure angle.

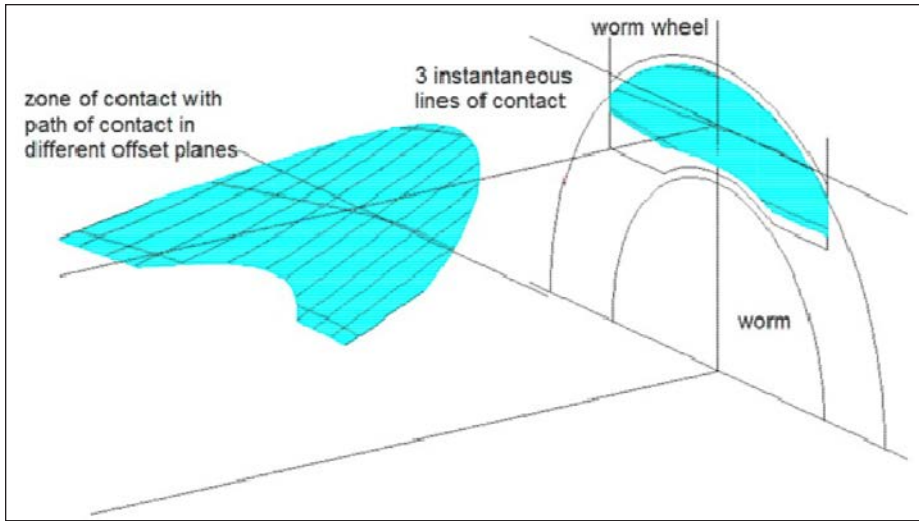


Figure 9 Path of contact, zone of contact and lines of contact for one relative position of worm and worm wheel.

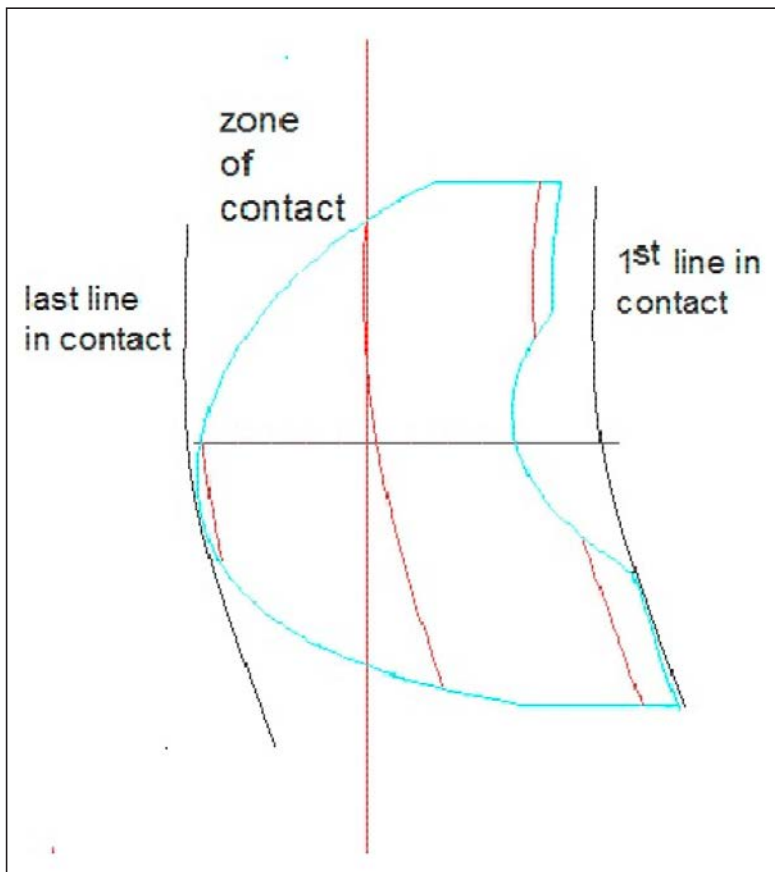


Figure 10 Contact ratio.

- a. For the worm: A plane parallel to the axis of the worm and the axis of the wheel with distance to the axis of the worm equal to the half of pitch diameter d_{w2}
- b. For the worm wheel the pitch cylinder is defined with the pitch diameter d_{w2}
- c. The pitch cylinder of the wheel is rolling without sliding on the pitch lane of the worm. The common tangent between the two pitch surfaces is called pitch axis; it is also the instantaneous axis of rotation.
4. In each offset plane, the rack profile of the worm, projection of the axial profile of the worm, allows to define the path of contact, which is the line along which the common point of contact between the flank of the worm and the conjugate flank of the worm wheel is moving during the gear mesh. From this line can be obtained the profile of the worm wheel. The shape of the path of contact can be concave or convex it; depends directly on the curvature of the rack profile of the worm. The pressure angle can be lower than zero. The zero contact pressure angle is a characteristic of worm geometry (Fig. 8).
5. Then it is important to determine the limits of the active path of contact. In principle it is limited by the outside active cylinder of the worm and the external circle of the external surface of the worm wheel (Fig. 6).
6. In certain circumstances the active path of contact can be limited by artifacts in the gear mesh due to:
 - a. Either by the presence of cusp (Fig. 7) when the path of contact is very flat when the pressure angle decrease and the contact exist only between A and T.
 - b. Either there is no active path of contact (Fig. 8) when the pressure angle is closed to zero; B is outside of the external surface of the worm wheel.
7. Before to determine the line of contact between the flanks of the worm. It is important to determine in each offset plane the limits of active path of contact as described in points 5 – 6. This defines the zone of contact. Then for each relative position of the worm

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example number	1	2	3	4	5	6
Assumption: $ha1^*=ha2^*=1$ $hf1^*=hf2^*=1.2$ $he2^*=0.5$						
Type of flank ZA ZK ZJ ZC ZI ZC	ZA	ZK	ZI	ZC	ZI	ZC
number of threads z1	2	2	2	2	2	2
axial module mx1 mm	4	4	4	4	4	4
diameter factor q1 -	9	9	9	9	10.28	10.28
normal pressure angle α_n°	20	20	20	20	20	20
reference lead angle of worm γ_{m1}° 12.53 11.01	12.5288077	12.5288077	12.5288077	12.5288077	11.0095079	11.0095079
worm reference diameter dm1 mm	36	36	36	36	41.12	41.12
worm root diameter df1 mm	26.4	26.4	26.4	26.4	31.52	31.52
worm tip diameter da1 mm	44	44	44	44	49.12	49.12
number of teeth z2	41	41	41	41	39	39
worm wheel profile shift coefficient x2	0	0	0	0	0.36	0.36
worm wheel reference diameter dm2 mm	164	164	164	164	158.88	158.88
worm wheel root diameter df2 mm	154.4	154.4	154.4	154.4	149.28	149.28
worm wheel tip diameter da2 mm	172	172	172	172	166.88	166.88
worm wheel outside diameter de2 mm	176	176	176	176	170.88	170.88
effective wheel facewidth b2H mm	30	30	30	30	30	30
wheel rim width b2R mm	30	30	30	30	30	30
outside diameter of the grinding disk dgr mm		309.6		309.6		309.6
profile radius of grinding disk p mm				24		24
centre distance a mm	100	100	100	100	100	100

type of flank	ZA	ZK	ZI	ZC	ZI	ZC
pm* Method B	0.8561	0.8687	0.8726	0.8536	0.8887	0.7377
pm* Method C	0.9496	0.9496	0.9496	0.7415	1.0797	0.8432
h* Method B	0.076	0.0745	0.0743	0.0855	0.079	0.0986
h* Method C	0.06891	0.06891	0.06891	0.08948	0.0658	0.0854
s* Method B	31.4293	30.8437	30.7635	33.8615	33.134	40.6564
s* Method C	30.285	30.285	30.285	36.215	33.659	40.253
Cusp in the gear mesh	no	no	no	yes	no	yes

Difference in % for pm*	-9.85%	-8.52%	-8.11%	15.12%	-17.69%	-12.51%
Difference in % for h*	10.29%	8.11%	7.82%	-4.45%	20.06%	15.46%
Difference in % for s*	3.78%	1.84%	1.58%	-6.50%	-1.56%	1.00%

- and the worm wheel the lines of contact can be determined (Fig. 9).
- The contact ratio of the worm gear can be determined by comparison of the position of the worm when the 1st line of contact is in contact with the beginning of the zone of contact and the position of the worm when the last line of contact is in contact with the end of the zone of contact (Fig. 10).
 - In each point of contact it is then possible to determine:
 - From the geometry: the normal vector, the tangential plane, the main and equivalent radius of curvature
 - From the kinematic conditions the rolling and sliding velocities
 - Revision of ISO/TR 10828 develops all the equations and algorithms to determine all the characteristics developed in points 1–9. For better understanding—and in order to be able to test algorithms—*MathCad* files

have been developed in the framework of the ISO TC60/SC2/WG7.

Examples of the Application of Revised ISO 10828

The following table summarize different calculation examples run for a same set of data with different worm profiles A, I, N and C for a same reference lead angle for examples 1 through 4 and for another reference lead angle for examples 5 and 6:

It can be observed that ISO/TR 14521 Method B (based on Revised TR 10828) and Method C provide significant differences for the non-dimensional parameter p^* , s^* and h^* —which can justify the clarification brought by the Revised TR 10828 in the application of Method B by detailed complex geometry calculations.

In those calculations the face width of the worm wheel has been divided in 53 offset planes and 20 relative po-

sitions between worm and worm wheel have been studied.

For p^* and h^* Method C is on the conservative side (oversizing gear set)—but not for s^* . In the case of cusp (C Profile), Method C is not applicable.

To illustrate the impact of cusp, the details of Example 4 are presented in Figure 11. It can be observed that the reduction of the zone of contact (in particular, there is an area where the lines of contact do not reach the outside diameter of the worm). On the left, the gear mesh in the median plane of the worm wheel emphasizes the phenomena. The other consequence is that when cusps exist, the equivalent radius of curvature is close to zero for the points on the border of the zone of contact (Fig. 12). It means that it is recommended to avoid cusp in gear mesh by increasing normal pressure angle.

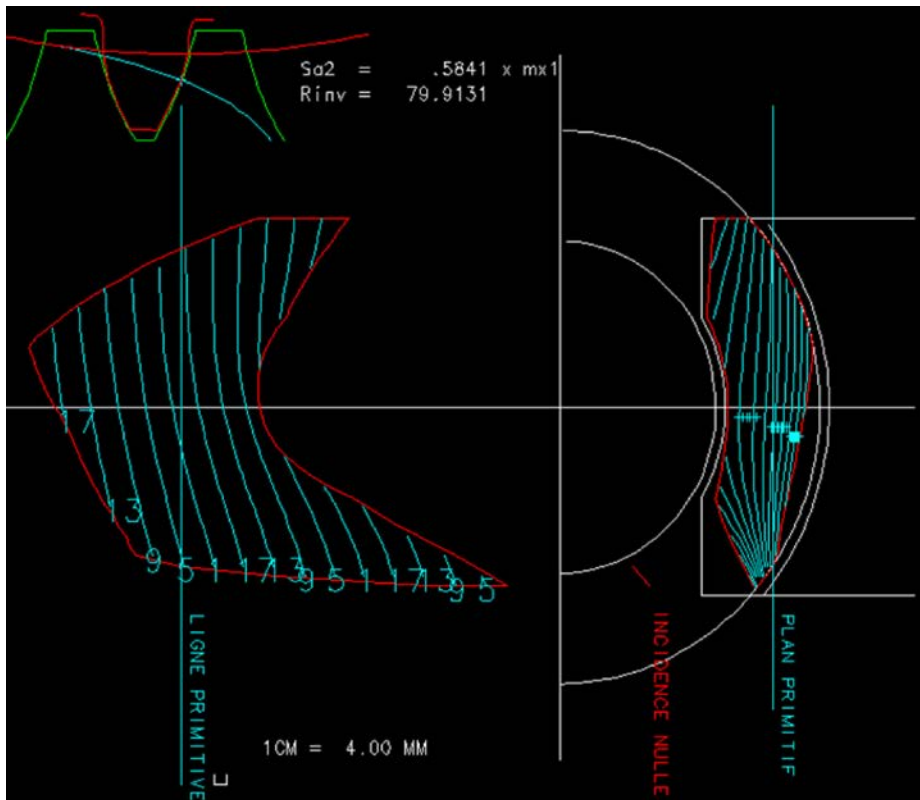


Figure 11 Example with cusp effect and reduction of zone of contact.

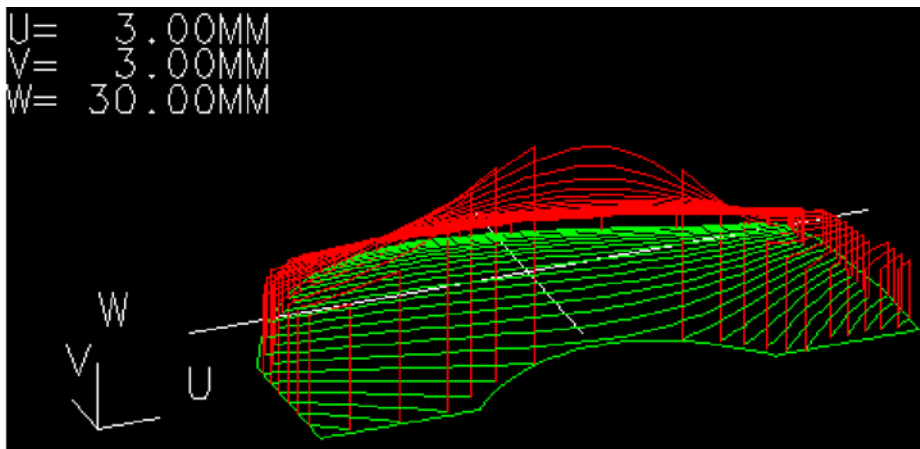


Figure 12 Equivalent radius of curvature in projection on worm wheel flank.

Dr. Michel Octrue is a CETIM (Centre Technique des Industries Mécaniques) expert in the field of mechanical power transmission as a specialist in the behavior of mechanical components (gears, roller bearings, etc.), with particular expertise in projects involving mechanical power transmission components and their integration in machinery and gearboxes for automotive and transportation devices. His experience covers the different stages — design and calculation; choice of tolerances; selection of materials and heat treatment; and development and validation by numerical simulation or testing devices. He is also involved in testing strategy and test bench design for mechanical components. Octrue is a graduate engineer from ECAM-Lyon, where he subsequently earned his doctorate in engineering. Among his technical achievements are the design and realization of software and training sessions to transfer the knowledge in the fields of gear technology (gear design, gear rating, gear tolerances and gear behavior in noise and vibration; the development of test benches for endurance to characterize cylindrical and worm gears; the design and realization of several test benches for gearboxes and different machine elements; the participation and animation of working groups in standardization activities in the framework of ISO TC60 – Gears; and the development of methodologies to reduce gear noise at the source. Octrue also serves as a technical expert for ANVAR, is speaker (20 int'l pubs), and has authored two books on worm gears (New Method for Designing Worm Gears and An Industrial Approach for Load Capacity Calculation of Worm Gears (Verifying and Design)).



Possible Extensions Provided by Revised TR 10828

The geometry of worm gears can be extended for any type of grinding wheel disk profile; one simply introduces the mathematical function of that profile and all the theory is still available.

The other possible extension is for the future evolution of ISO/TR 14521, in which at the moment the contact pressure distribution is considered as a constant value. However this assumption is far from reality, and industrial results have shown that this was not correct.

This type of calculation (radius of curvature, normal vector, etc.) can be used also for other mechanical components — profiles for roller screws, for example.

Conclusions

- The complex geometry of worm gear is now clearly detailed and Method B of ISO/TR 14521 is more accessible.
- This Technical Report will bring a better knowledge and understanding of the behavior of worm gears for the future. **PTE**

Acknowledgement. Thanks to all members of ISO TC60/SC1/WG7 for their active participation in the elaboration of these standards.

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