

An Industrial Approach for Load Capacity Calculation of Worm Gears (Verifying and Design)

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American Gear Manufacturers Association



TECHNICAL PAPER

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ABSTRACT:

The method proposed in this paper is based on an analytical rating method which has been developed some years ago by the author (see AGMA paper 88 FTM 6).

The calculation is based on the determination of the maximum of pressure between the mating teeth which is made by using a specific criteria for worm gears. This method can be used to verify the load capability of a worm gear but also to design a new gear covering several types of tooth profile.

Calculations are provided as examples and comparisons have been made with results obtained by the initial analytical method, and by standardized methods (AGMA, BS).

At the end of the paper a comparison is made with experimental results.

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AN INDUSTRIAL APPROACH TO LOAD CAPACITY CALCULATION OF WORM GEARS VERIFYING AND DESIGN

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1 INTRODUCTION

If the different rating methods usually used for designing worm gears ([1],[8] à [10]) consider their limited life because of **TEETH BREAKAGE** and **CONTACT PRESSURES**, experience shows that in most cases, it is the **SURFACE DEGRADATIONS OF THE WORMWHEEL** which limit the gearing life.

The main cause for **SURFACE DEGRADATION** is the fatigue phenomenon on the material layer submitted to **CONTACT PRESSURES**. This has brought us to develop a method of calculation allowing to evaluate the level and distribution of contact pressures along the flanks of the teeth.

As opposed to existing methods of calculations, the distribution of contact pressures is no longer supposed to be uniform.

The development that follows presents the method basis used as well as a simplified method which can easily be applied industrially either for verifying or design.

2 REMINDER OF ANALYTIC METHOD

For a worm gear, the cartographic calculation of contact pressures is obtained from a detailed analysis of meshing of a cylindrical worm with the mating conjugate wormwheel. In fact for this type of gear the gearing conditions evolve continuously and differ from one point to another in the same contact line; thus the name of analytical method ([2] to [4]).

The analysis of the meshing is performed at 2 steps:

- **STEP 1:** it aimed to determine for a relative position of the worm and the wormwheel the instantaneous load distribution along the meshing teeth. This distribution depends on the teeth geometry, the contact geometry and the materials making up the teeth. It is thus possible to calculate a pressure level for each contact point.
- **STEP 2:** by varying the relative position of the worm and the wormwheel it is then possible to study the contact pressure fields on the whole contact zone of the gear.

The principle on which is based the analytical rating method rests on determining the contact point for which the contact pressure field attains its maximum. The maximum load capacity admissible by the gear is thus limited by the pressure allowed by the most weaker material (generally that of the wormwheel) for the number of request cycles.

We have indicated in [2] that this type of calculation leads to expressing the admitted torque to the wormwheel C (in m.N) using a rather simple formula:

$$C = r_{w2} \cdot \left(\sigma_{Hlim} \cdot \frac{Z_H}{Z_E} \right)^2 \cdot Z_R$$

with:

- r_{w2} : operating pitch radius of the wormwheel (in mm)
- σ_{Hlim} : admissible pressure of the material (in MPa)
- Z_H : life factor
- Z_E : elasticity factor (in MPa to power 1/2)
- Z_R : pressure distribution factor of the worm gear (in square mm)

Z_R is composed of the global stiffness factor RDG_p which takes into account the global stiffness of the gear and the $(K_{t,jmint})_p$ factor which combines the mean equivalent stiffness $(RD_{eqm})_{t,j}$ and the equivalent curvature radius $(R_{eq})_{t,j}$ calculated at the maximum pressure amplitude contact point (index i, j) for the relative position of the worm and the wormwheel (index p):

$$Z_R = RDG_p \cdot (K_{t,jmint})_p$$

$$(K_{t,jmint})_p = \left(R_{eq} \cdot \frac{x_H}{RD_{eqm}} \right)_{t,j,p}$$

with:

- x_H : directive cosine of the normal to plan of contact

The power dissipated in the meshing under friction form is calculated from the determined load distribution and the sliding velocity which allow to calculate the friction coefficient in all contact point: from which we deduce instantaneous efficiency η . The torque transmitted by the gear is then expressed by:

$$C_{trans} = \eta \cdot C$$

In this type of calculation the difficult point is the determination of the pressure distribution factor of the gear Z_R for this requires important calculation means. To obtain a calculation method much easier to use, a simulated analytic method has been undertaken to establish a variation rule for the Z_R factor in function of the basic geometrical parameters of the wormgear.