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DIN 3996: A New Standard for Calculating the Load Capacity of Worm 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

During the last years the load capacity of worm gears was raised about 30%. The reasons for this are the introduction of synthetic oils, optimization of the worm gear geometry and manufacturing improvements. The forthcoming new standard DIN 3996 "Calculation of load capacity of cylindrical worm gear pairs" takes into account these developments. This standard contains the following load capacity limits: wear, pitting, tooth breakage, temperature and worm deflection. Also, efficiency was taken into consideration. In most cases the calculation methods are based on results of recent investigations, which were performed on worm gear test rigs at the FZG. In the case of wear, pitting and tooth breakage test results and their influence on DIN 3996 are shown.

The calculation method for wear resistance is based on the fact that the wear intensity of a material/lubricant combination is a function of the lubricant film thickness and the lubricant structure. The main influence parameter on pitting resistance is the Hertzian stress. For tooth root strength the calculation method is based on a nominal shear stress theory, in case of worm deflection on the deflection curve of a cylindrical shaft.

The application of this new standard for gears in practice is discussed by recalculating some examples.

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1 Introduction

Modern worm gear drives are used more and more in heavy duty service, thus the knowledge of the load capacity limit is important. The main load capacity limits of gear drives with worms made of case carburized steel and worm wheels made of bronze are wear, pittings, tooth breakage, and temperature (see Figure 1).

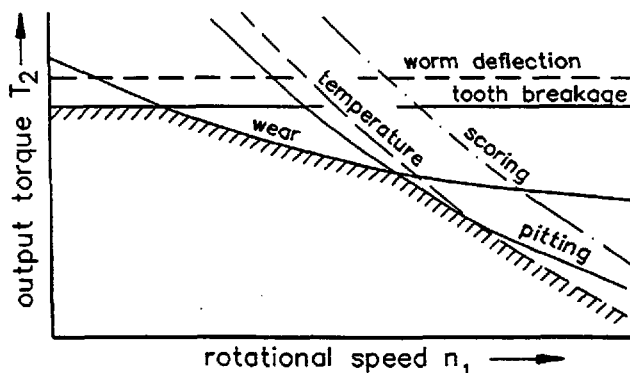


Fig. 1: Load Capacity Limits (acc. to /11/) as a Function of the Rotational Speed

Up to now in the case of worm gears, there was no German standard for calculating their load capacity. Other

standards like B.S. 721 /9/ and AGMA 6034-B92 /10/ have not been widespread in Germany. Many companies apply the calculation method published in [1].

Now, a DIN standard, whereby in industry and university have cooperated together, for gear drives has been developed, which takes into consideration the improvements of worm gears during the last decades. The forthcoming new standard DIN 3996 "Calculation of load capacity of cylindrical worm gear pairs" takes into account the failure limits mentioned above and also temperature. In the case of scuffing, research work has been not yet sufficiently carried out so as to introduce a reliable calculation method.

1.1 General Working Principle of DIN 3996

Although the calculation methods for the above mentioned kinds of damage have different origins, the basic principle of the calculation procedures is the same for each type of damage: For a recalculation procedure the operating conditions must be given or specified. With the geometrical data and the operating conditions of the gear drive actual values of the respective criterion for the dif-

ferent types of damage can be calculated. These actual values are compared to limiting values. The result is a safety factor S which can be determined acc. equation (1):

$$S = \frac{\text{limiting value}}{\text{actual value}} \quad (1)$$

If this safety factor shows low values a comparably high damage risk has to be expected, a high safety factor means low risk of damage. This is similar to the procedure for spur and helical gears in DIN 3990 and ISO 6336 or bevel and hypoid gears in DIN 3991.

1.2 Field of Application

In most cases the limiting values of the endurable loads have been determined in experiments with worm gears of different centre distances and different gear ratios. In these experiments the influence of various worm and wheel materials as well as different lubricants were investigated. The experiments were performed at the FZG (Forschungsstelle für Zahnräder und Getriebekonstruktion, Technical University Munich). The results are documented in /1/, /2/, /3/, and /4/. Further data on the load capacity is based on the operating experience of industrial companies.

The calculation methods presented in DIN 3996 can be used for gear drives with worm and wheel materials and lubricants as listed below:

Worm materials:

- * Case carburizing steel (e.g. 16MnCr5), case hardened
- * Through hardening steel (e.g. 42CrMo4), flame or induction hardened
- * Nitriding steel (e.g. 31CrMoV9)

Worm wheel materials:

- * Centrifugally cast bronze: (e.g. GZ-CuSn12, GZ-CuSn12Ni, GZ-CuAl10Ni). These bronzes should have, as far as it is possible, a homogenous structure, the mean grain size should be below 150 μm .
- * Grey cast iron (e.g. GG-25)
- * Spheroidal graphite iron (e.g. GGG-40)

Lubricants:

- * Mineral oils containing mild additives
- * Polyglycols with basic oil EO:PO = 0:1 and 1:1 (EO $\hat{=}$ Etylenoxyde, PO $\hat{=}$ Propylenoxyde) and Poly-alphaolefins

2 Physical Parameters

Although the physical reasons of the worm gear failures are not researched in every detail, their main influence parameters are known. Important influence parameters are the mean Hertzian stress σ_{Hm} , the lubricant film thickness h_{minm} and the sliding path s_w . Different worm gear drives cannot directly be compared by these parameters because these are influenced by working conditions, the materials of worm and wheel and the lubricant.

In order to be able to compare the surface durability of worm gears, generalizing non-dimensional parameters are defined, p_m^* for the mean Hertzian stress, h^* for the mean lubricant film thickness and s^* for the mean sliding path. These relative parameters are independent of size, load, material and lubricant and thus only a function of the worm and wheel geometry. The derivations of these parameters are described in /7/ and /8/. These physical parameters can be determined using numerical methods. Because these methods (computer programs) are rather complicated, relatively simple approximating equations can be used instead. The following approximations are described from /5/. For cylindrical worm gears with involute flank form they can be calculated from the equations (2) to (4):

$$p_m^* = 1.03 \left(0.4 + \frac{x}{u} + 0.01 \cdot z_2 - 0.083 \cdot \frac{b_{2H}}{m_x} + \frac{\sqrt{2q-1}}{6.9} + \frac{q+50 \cdot (u+1)/u}{15.9+37.5 \cdot q} \right) \quad (2)$$

$$h^* = 0.018 + \frac{q}{7.86 \cdot (q + z_2)} + \frac{1}{z_2} + \frac{x}{110} - \frac{u}{36300} + \frac{b_{2H}}{370.4 \cdot m_x} - \frac{\sqrt{2q-1}}{213.9} \quad (3)$$

$$s^* = 0.78 + 0.21 \cdot u + 5.6 / \tan \gamma_m \quad (4)$$

It is obvious, that these non-dimensional parameters only cover geometrical data.

With these non-dimensional parameters and the working conditions the actual values of the Hertzian stress σ_{Hm} , the lubricant film thickness h_{minm} , and the sliding path s_w can be calculated with equations (5) to (7). Tooth strength and shaft deflection are calculated directly, as usual.

$$\sigma_{Hm} = \frac{4}{\pi} \cdot \left(\frac{p_m^* \cdot T_2 \cdot 10^3 \cdot E_{red}}{a^3} \right)^{0.5} \quad (5)$$

$$h_{minm} = 21 \cdot h^* \cdot \frac{C_\alpha^{0.6} \cdot \eta_{OM}^{0.7} \cdot n_1^{0.7} \cdot a^{1.39} \cdot E_{red}^{0.03}}{T_2^{0.13}} \quad (6)$$

$$s_W = s^* \cdot \frac{\sigma_{Hm} \cdot a}{E_{red}} \cdot N_L \quad (7)$$

It is the advantage of this method, that for each gear drive the non-dimensional parameters (equations (2) to (4)) only have to be calculated once. The wear and pitting resistance of different worm gear drives can be easily compared by means of these parameters.

3 Efficiency and Coefficient of Friction

An important influence on the performance of worm gears is their efficiency. When calculating efficiency, the coefficient of friction must be known. Main influence parameters on the coefficient of friction are the materials of worm and wheel, the lubricant, the sliding conditions along the contact lines and the equivalent diameters. Because the coefficient of friction in the mesh cannot directly be measured in a worm gear test rig¹, a three disk test rig was designed and built to simulate the worm gear contact and investigate the coefficient of friction under sliding conditions of any point along the contact lines (the three disk test rig is described in /6/). Figure 2 shows coefficients of friction, measured on the three disk test rig, along the contact lines in the case of a worm gear with centre distance $a = 100$ mm and ratio $u = 20.5$. For this example the Hertzian stress was set to a constant value ($p_H = 300$ N/mm²) along the contact lines. This Figure shows, that the coefficient of friction along the contact lines - beginning from the inlet - steadily increases to the centre of the flank. There it has a maximum and then drops to lower values at the outlet. From these measured local values μ mean values μ_{zm} can be determined. This mean coefficient of friction shows values, which correspond with values, calculated from measured efficiency values of real worm gear drives.

¹ In worm gear test rigs the coefficient of friction can be recalculated from the measured efficiencies.

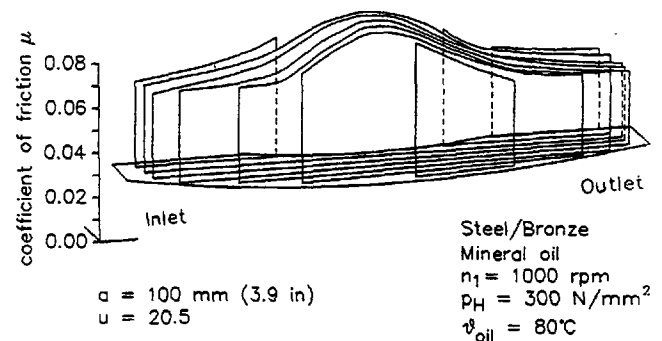


Fig. 2: Coefficient of Friction along the Contact Lines of a Worm Gear Drive with Centre Distance $a = 100$ mm and Gear Ratio $u = 20.5$ from Three Disk Simulation

Based on the investigations, performed on the three disk test rig and on the worm gear test rig, the mean coefficient of friction μ_{zm} can be calculated with equation (8):

$$\mu_{zm} = \mu_0 \cdot Y_S \cdot Y_G \cdot Y_W \cdot Y_R \quad (8)$$

In Figure 3 the mean basic coefficient of friction μ_0 , included in the standard DIN 3996, is documented.

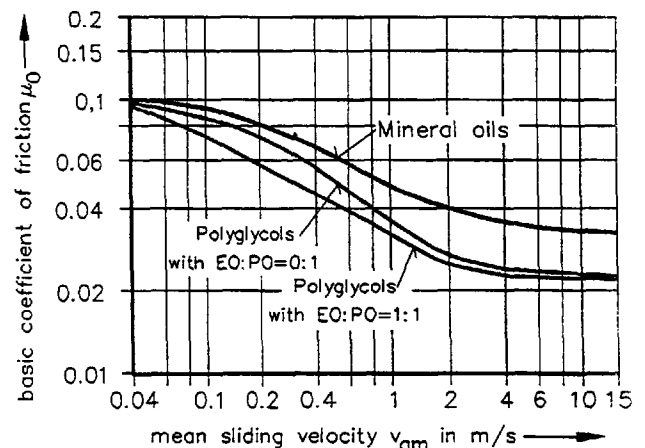


Fig. 3: Basic Coefficient of Friction μ_0

This Figure shows that the basic coefficient of friction μ_0 tends to lower values with increasing sliding velocity. In comparison to lubrication with mineral oil polyglycols show - especially at higher sliding velocities - lower values for the basic coefficient of friction.

The influence of the centre distance, geometry (contact line positions), the worm gear material and the surface roughness on the mean coefficient of friction μ_{zm} can be taken into account with influence factors, which are described by the equations (9) to (11):