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DETERMINING AN ALLOWABLE WEAR OF WORM WHEELS

Summary

The paper presents an analysis based on the two criteria used to determine an allowable sliding wear of the worm gear side teeth and of the worm wheels. In both cases, two undesirable consequences occurred. The first one is a reduction in the tooth root cross section, which may lead to tooth fracture. The other one is the appearance of tooth tapers. Both consequences are undesirable in all types of gear transmissions because of the risk of tooth fracture.

Key words: *worm wheels, tooth fracture, tooth wear, tooth taper, tooth root*

1. Introduction

In comparison with all other gears, worm gears exhibit relatively high energy losses during operation in spite of their good features. This fact was a starting point for conducting this research. The aforementioned fact defined our research focused on an improvement in durability, cheaper manufacturing, and reduced energy losses. The worm gear durability is determined by a tribological system which includes the worm material, the worm wheel material, and the lubricant. The materials in this tribological system are similar to the materials in journal bearings. The worm screw – journal relation and the worm wheel – bearing lining relation, along with a lubricant as the third component, represent the key elements of the total no-load energy loss of a worm gear. Recently, efforts have been made to replace tin bronze with a more suitable bronze type, at least as far as the price is concerned. In this respect, aluminium bronze has been highlighted as particularly suitable. The current scientific research demonstrates a possibility of using aluminium alloys. More details on this topic can be found in the papers written by Niemann [1], Winter [2], Huber [3], Opalić [4, 5], Žeželj [6], and Panić [7].

The third factor in a tribological system is a lubricant, which has an important role both in wear reduction and in increasing the resistance to contact pressure. When synthetic lubricants appeared on the market, it was considered that they contributed to an improved contact pressure limit, but some authors proved the opposite, having compared the synthetic lubricants with the mineral ones, e.g., Huber [3], Wilkesmann [9], Opalić [4, 5], Žeželj [6], Panić [7] and Muminović [8].

In real conditions, the main problem is that the viscosity of a lubricant cannot be clearly defined. This issue has to be further clarified with respect to the working temperature and an allowable pressure at a critical point of contact between the worm and the worm wheel.

2. Calculation

When a worm gear is lubricated by immersion, there are two ways of determining the allowable wear:

1. wear restriction by protecting the worm wheel tooth root from fracture (i.e. from the critical weakening of the tooth root by the cross section reduction due to wear);

2. wear limitation through the tooth thickness at the worm wheel crown diameter (i.e. the undesirable pointedness of the tooth is avoided by a tip tooth thickness of $s_a > 0$).

In order to determine the allowable wear of the first possibility (i.e. a minimum cross section of the tooth root), the following calculation according to Niemann will be applied [2]:
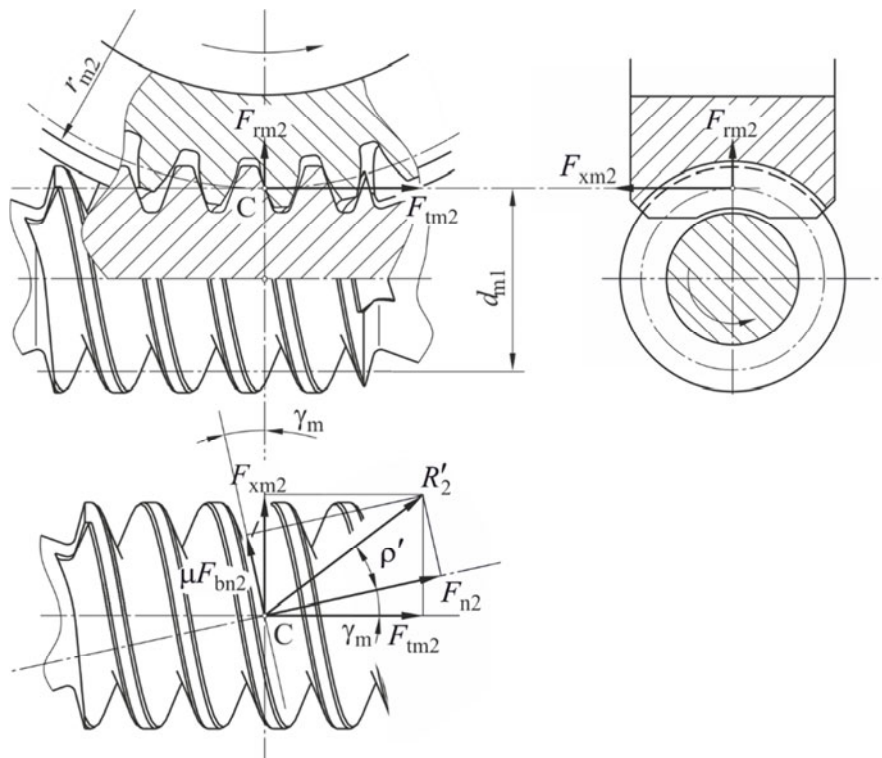

Fig. 1 Tooth load

$$F_{tm2} = \frac{p_x \cos\gamma_m \, b_2 \, \tau_{F\,lim\,T}}{S_F} = \frac{m_n \, \pi \cos\gamma_m \, b_2 \, \tau_{F\,lim\,T}}{S_F} =$$
$$= \frac{2s'' \cos\gamma_m \, b_2 \, \tau_{F\,lim\,T}}{S_F} \tag{1}$$

Where

F_{tm2}, N – the circumferential or tangential force on the worm wheel,

p_x, mm – the axial pitch of the worm,

γ_m, ° – the reference lead angle of the worm,

b_2 , mm – the facewidth of the wheel,

$\tau_{F\,lim\,T}$, N/mm² – the shear endurance strength,

S_F , - – the tooth breakage safety factor,

m_n , mm – the normal module,

s , mm – the initial tooth thickness,

s'', mm – the tooth thickness after a certain period of time.
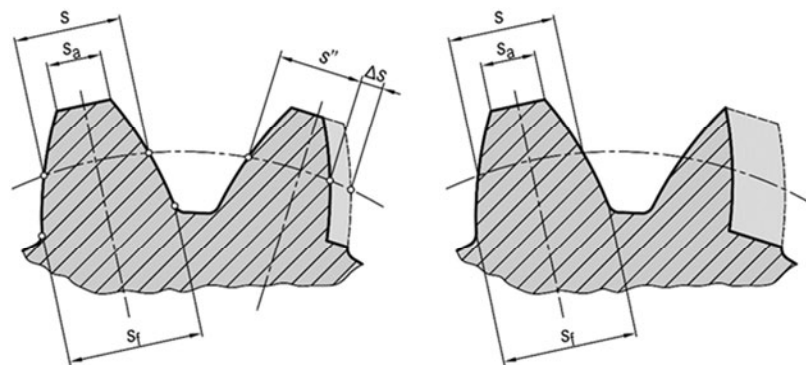

Fig. 2 A simplified image of a decrease in the tooth thickness due to an increased wear (cross-section)

During the worm gear service life, the tooth thickness s will be reduced to the size s'' (Figure 2), so after a certain period of time, the tooth thickness can be expressed as follows:

$$s'' = s - \Delta s \tag{2}$$

By introducing the output torsion moment T_2, the following expression will be obtained:

$$T_{2max} = \frac{(s - \Delta s)\,d_{m2}\,\cos\gamma_m\,b_2\,c_{gr}}{S_F} \tag{3}$$

leading to

$$\Delta s = \frac{m\,\pi}{2} - \frac{T_{2max}\,S_F}{d_{m2}\,\cos\gamma_m\,b_2\,c_{gr}} \tag{4}$$

After inserting the parameters of the tested gear in Equation (4), the values of allowable wear Δs_{lim} are calculated as the root strength criterion for different output torques and the root resistance to fracture for the material AlSn6.

Table 1 The calculated values of an allowable tooth root thickness reduction as the root strength criterion

T_2, N·m	Δs_{lim} , mm	
	S_F=1	S_F=1.2
180	3.852	3.367
210	3.447	2.881
260	2.772	2.071
310	2.096	1.261
370	1.286	0.289

Based on Table 1, the relationship between an allowable tooth change and a specific wear can be obtained by calculating the volume loss due to wear and by multiplying it by the specific weight of the wheel tooth material.

The surface area of one worm gear tooth (i.e. the surface area of a projected wheel side) can be approximately calculated by the following expression (Figure 3):
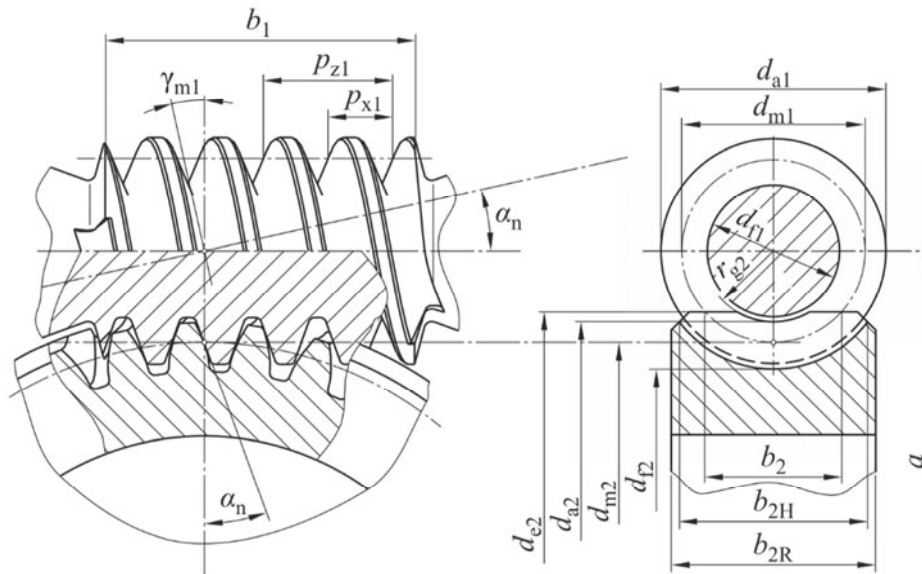

Fig. 3 The parameters for the tooth surface area calculation

$$A_{z21} = \frac{d_1 \pi m \arcsin(\frac{b_2}{d_1 + 2m})}{90 \cos \gamma_m \cos \alpha_n} \tag{5}$$

After inserting the values of the tested gear into Equation (5), the surface area of a single wheel tooth is obtained:

$$A_{z21} = 213 \text{ mm}^2$$

The calculated surface area is smaller than the actual one because of the tooth curvature [5]. By numerical integration we get an area of 256.8 mm² while by measuring a 3D CAD model, the area of 258.6 mm² [6, 7].

The total surface of wheel teeth amounted to:

$$A_{2uk} = z \ A_{z21 3D} = 9{,}309.6 \text{ mm}^2$$

The value of allowable specific sliding wear is obtained on the basis of a specific weight of 2.84 g/cm³ of the tested alloy and a service life of 12,000 hours, as well as on the basis of the tooth root strength criterion. The calculation results are presented in Table 2.

Table 2 The calculation results of allowable specific sliding wear as the tooth root strength criterion. (The values refer to a service life of 12,000 hours.)

T_2 (N·m)	Δms_{lim}, mg/h	
	$S_F=1$	$S_F=1.2$
180	8.487	7.418

T_2 (N·m)	Δms_{lim}, mg/h	
210	7.595	6.348
260	6.107	4.563
310	4.618	2.778
370	2.833	0.637

Referring to a different service life, the values presented in Table 2 will be calculated as follows:

$$\Delta ms_{\text{limxh}} = \frac{\Delta ms_{\text{lim}}\, 12{,}000}{L_{\text{h}}} \qquad (6)$$

where

Δms_{limxh}, mg/h –the allowable specific wear for the desired service life,

Δms_{lim}, mg/h – values in Table 2,

L_{h}, h – the desired service life.

Based on the calculated values of the allowable wear in accordance with the two criteria (i.e. the tooth root strength and the tooth taper), Figure 4 shows the allowable output torques correlated to the main shaft speed:

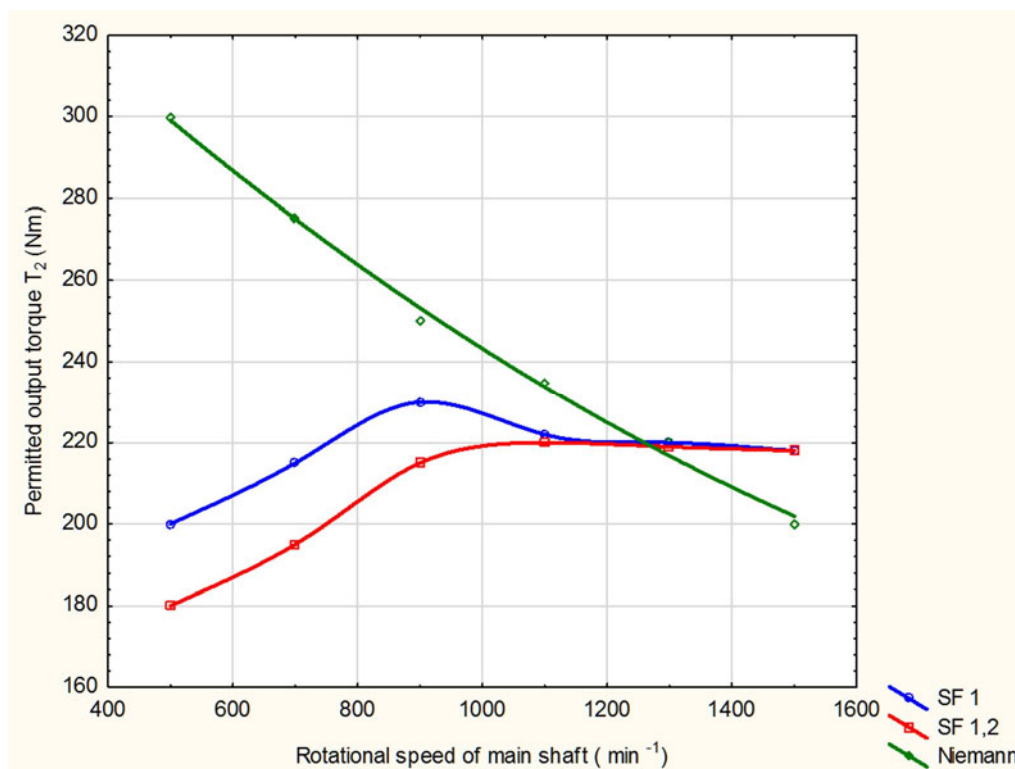AlSn6/34CrMo4 mineral oil, $T_{\text{u}} = 60°$ C.

Fig. 4 The allowable output torques of the tested gear according to the tooth taper and tooth root strength criteria [1, 5, 7]. S_F is the tooth breakage safety factor

3. Conclusion

The obtained results show that there is a relatively simple method for determining the allowable wear of the worm gear and the worm wheel teeth. Depending on the worm wheel material, it is possible to satisfy one of the criteria (e.g. the tooth taper or the tooth root strength) before satisfying the other. This paper presents the calculations for $s_a = 0$ (i.e. the theoretical taper limit), while in practice, the calculations will be performed with a minimum s_{amin} of $0.4m$.

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