MACHINE SCIENCE

ISSN: 2227-6912
E-ISSN: 2790-0479

2022, Volume 11, № 2. Pages 44 –53

COMPARATIVE SERVICE LIFE ANALYSIS FOR WORM GEARS ACCORDING TO DIFFERENT FAILURE CRITERIA

Iftikhar CHALABI

Department of Mechatronics and Machine Design of Azerbaijan Technical University, Baku, Azerbaijan

E-mail: i_chalabi@aztu.edu.az

Abstract: In this paper, a comparative analysis of the service life of worm gear wheels for various failure criteria is carried out. Based on this analysis, a method is proposed for selecting the worm gear wheel material, the values of the modulus, on which the bending strength of the teeth depends. Research has shown that increasing the bending strength of teeth is of great practical importance, both for one-sided and two-sided loads. This also makes it possible to significantly increase the lifetime of one-side loaded gear wheels by replacing their working flanks with non-working ones. it was proposed to replace the working surfaces of the teeth of worm wheels after a certain service life.

Keywords: *Worm gear, service life, failure, pitting, tooth breakage, wear*

Introduction. Worm gears are very widely used in the structure of many machines and equipment, because they have a large gear ratio, high kinematic accuracy, smooth and silent operation compared to other mechanical transmissions. One of the disadvantages of these gears is the high cost of their material and manufacture. Therefore, the implementation of preventive measures to prevent premature failures and increase the service life of the worm gear is very important.

The reasons for the failure of worm gears are very various. The type of failure generally depends on many factors, such as operating conditions, material properties, geometric dimensions, manufacturing accuracy, etc. For timely elimination of possible failure to the worm gears, as well as increasing the service life, it is particularly important to choose the right materials and geometric dimensions at the design stage and conduct regular maintenance work during lifetime.

In the considered literature [1–5], calculations of worm gears based on the main failure criteria are usually considered separately. And in this case, the corresponding designed life of worm gears for different failure criteria are not compared. Therefore, in most cases, it is not possible to judge which type of failure to the worm gears is more likely and may occur earlier than others. Some types of failure (for example, teeth breakage) are the most dangerous. And other types of worm gear's failure (for example, pitting on the teeth) do not always lead to complete destruction of the worm gear, and they can in some cases be prevented in various ways. Therefore, a comparative lifetime analysis for worm gears according to different failure criteria can be of great practical importance.

In [6, 7], the bending strength of the teeth of various worm gears is studied by modern methods. However, the assessment of the strength and service life of worm gears by other failure criteria is not considered. And in works [8-11], studies of worm gears are carried out according to the criteria for failure of the working flanks of the teeth. A comparative analysis of the service life of worm gears for various failure criteria in these works is also not considered.

An analysis of the literature on worm gears has shown that a comparative analysis of their service life for various failure criteria has not yet been considered. Such an analysis can be of great practical significance, and therefore, this paper is devoted to this important task.

Formulation of problem. The types of damage occurring on the worm gear tooth (Fig. 1) can be divided into two groups. The first group includes flank damage such as pitting, wear, scuffing and grooving, which basically occur on the working flank of the teeth of the worm gear as a result of contact stresses σ_H (Fig. 2). The second group includes damage to the tooth root (overload breakage or fatigue breakage), which occur as a result of the bending stresses (τ_F) occurring there, which are manifested on the working flank as tensile stresses, on the non-working flank as compressive stresses. Since worm gear wheel is one of the weakest elements of the worm gearing, these damages determine the service life of the entire gearing in the majority of applications.

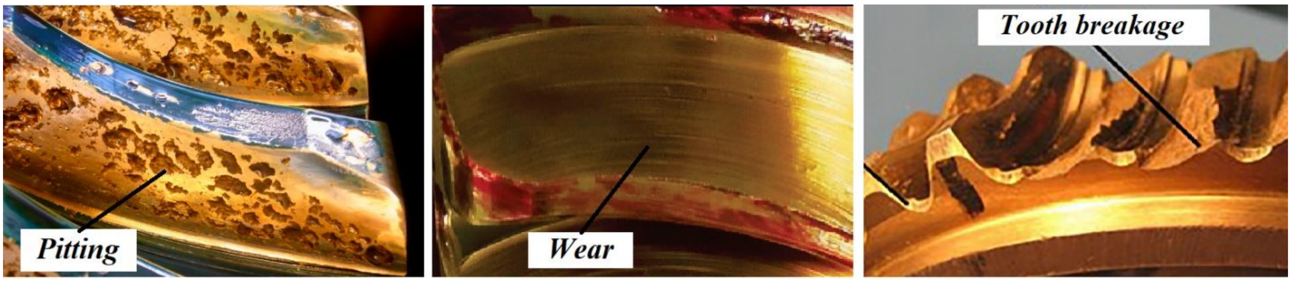

Fig. 1. Tooth damage to worm gear wheel [11, 12, 13]

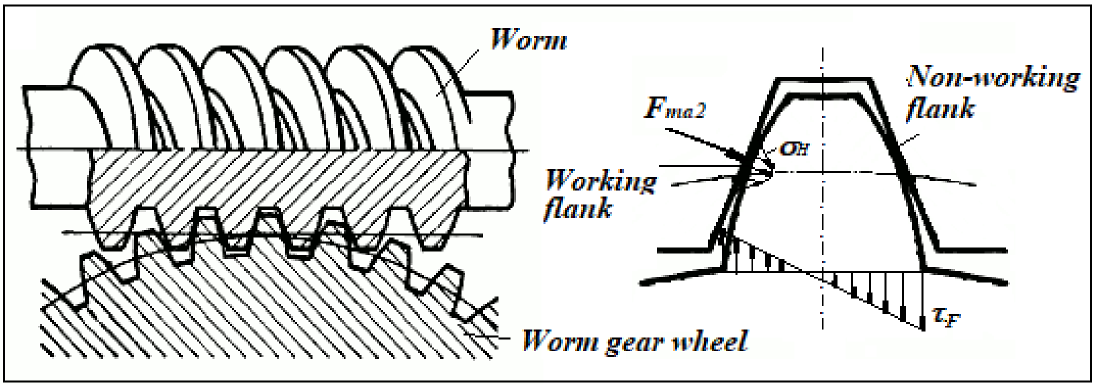

Fig 2. Operating stresses and types of worm gear teeth damage

If the worm gear wheel direction of rotation does not change during the entire operating time, the flank damage occurs only on the working flanks of the teeth. The non-working flanks are not loaded and remain without damage, regardless of the operating time. In [14], the question of increasing the service life of spur gearboxes by activating the non-working flank was considered. Due to the fact that worm gearboxes are characterized by increased sliding wear compared to spur gearboxes, an investigation is carried out in the present work on increasing the service life of worm gearboxes by changing the flank of the worm gear wheel teeth. The flank change of the teeth can be realized either by changing the direction of rotation or by inversion the worm gear wheel. Since tooth breakage can occur both on the tooth root of the working flank and the non-working flank, a comparative life assessment according to failure criteria tooth breakage and pitting (or wear) is required. An increase in service life by changing the tooth flank is only possible if the durability up to the occurrence of the tooth breakage is significantly greater than the service life up to the maximum (permissible) pitting (or wear).

It is known from practice that the main tooth damage of the worm gear wheel made of bronze, such as pitting, wear and tooth breakage, have mutual interactions [1]. When a critical pitting growth is reached, on the one hand, the abrasive sliding wear increases, on the other hand, increased wear can lead to a standstill in pitting development. Due to continuous wear, the pitting can even disappear again in many cases. Due to increased wear, the tooth thickness of the worm gear wheel is gradually reduced, as a result of which the probability of tooth breakage increases significantly. In [8], based on extensive studies, the service life of worm gear wheels was divided into three phases (Fig. 3). In phase I (pitting formation phase with load cycle number N_{LI}), only a small pitting value ($A_{P10} \leq 2\%$) and low mass removal (wear value Δm) are recorded. In the pitting growth phase II (load cycle number N_{LII}), the wear value increases insignificantly, although the pitting increases towards the end of this phase to the maximum value. In the wear phase III (load cycle number N_{LIII}), the pitting decreases and stagnates at a low value. But the wear value increases rapidly and leads to a significant weakening of the tooth root. This phase usually ends with the total failure of the worm gear set due to tooth breakage or a disturbed torque transmission behavior.

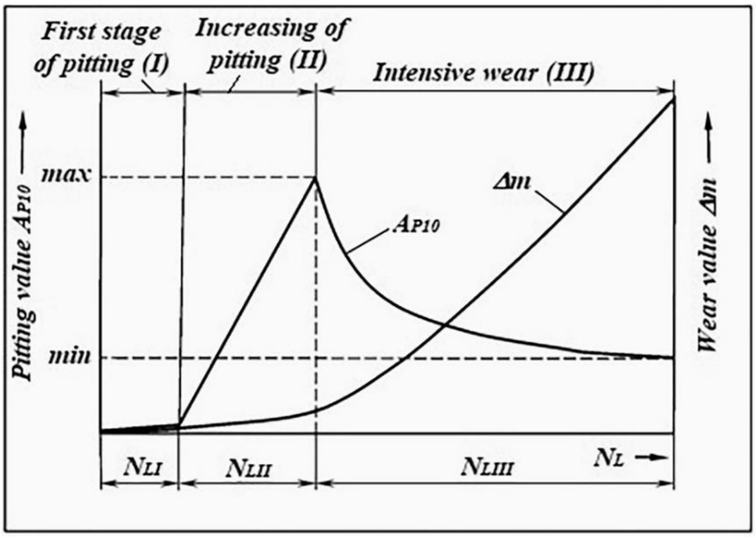

Fig. 3. The stages of failure on worm-gear wheel

Comparative analysis of service life. The activation of the non-working flank at the end of the pitting growth phase can lead to a significant increase in the service life, since at this point the wear value is still low and the teeth are little damaged. The stages of occurrence of failures in the teeth of a worm gearwheel are reflected in Fig. 4, a. As can be seen from the figure, damage to the tooth surface occurs, as a rule, in its working profiles. And non-working opposite profiles is practically not susceptible to damage after prolonged operation. Therefore, at the end of the second stage of operation, it is possible to increase the durability of the worm gear drive by replacing the working profiles with non-working ones. As can be seen from Fig. 4, b, with this method, the resource increase will be approximately in the amount of $N_{LI} + N_{LII}$.

To assess how much the worm gear resource can be increased by the above proposed method, it is of particular importance to conduct a comparative analysis of service life according to the criteria of pitting and wear, as well as according to the criteria of pitting and tooth breakage. Considering that the first two stages of the worm wheel service life are associated with pitting, the following empirical expression was proposed in [1] to determine the maximum limit of the total duration of these two stages:

$$N_{HL} = N_{LI} + N_{LII} = 3 \cdot 10^6 \cdot \frac{v_{gm}}{v_{ref}} \cdot \exp\left[24,924 - 4,047 \cdot \ln\left(520 \frac{\sigma_{Hm}}{\sigma_{Hlim}}\right)\right]. \qquad (1)$$

Where v_{gm} is sliding speed on the center circle in the flank direction; v_{ref} is sliding speed in the test sample (v_{ref}=3 m/s); σ_{Hm} medium contact stress on the flank of worm wheel teeth; σ_{Hlim} is the endurance limit.

Sliding speed on the center circle in the flank direction can be determined on the basis of [1] as follows:

$$v_m = \frac{d_{m1} \cdot n_1}{19098 \cdot \cos \gamma_m}$$

Where d_{m1} is the reference circle diameter of the worm; n_1 is speed of rotation on the worm shaft; γ_m is pitch angle at the center circle of the worm.

According to [3] the current calculated stresses σ_{Hm} can be determined as follows:

$$\sigma_{Hm} = \sqrt{\frac{2T_2 \cdot K_A}{a^3}} \cdot Z_E \cdot Z_0$$

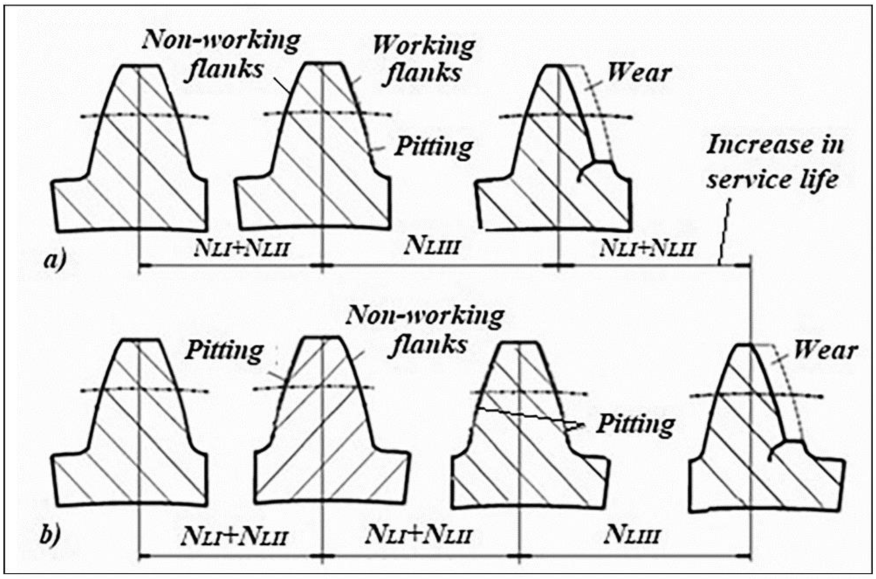

Fig. 4. The stages of failure on worm-gear wheel without replacing (a) and with replacing (b) of the working flanks of the teeth

Where T_2 is the moment on the worm wheel; K_A is the application factor; Z_E is the elasticity factor; Z_0 is coefficient that takes into account the increase in the contact line in the worm gear; a is centre distance of worm gear.

As it was noted, the third and at the same time the last stage of service life of the worm wheel may end as a result of two different reasons. First, let's consider the first case when the worm wheel loses its operability as a result of exceeding the permissible wear limit of the teeth. To determine the resource of the worm wheel by the permissible wear of the teeth, in [6] the following expression was proposed:

$$N_{WL} = N_{LIII} = \frac{\delta_{WD} \cdot E_g}{J_W \cdot s_m^* \cdot \sigma_{Hm} \cdot a} \tag{2}$$

Where δ_{WD} is limit value of flank wear, mm; E_g is the modulus of elasticity in worm gear, N/mm^2; J_W is the wear intensity; s_m^* is the medium glide path, mm.

The limit value of flank wear at the normal cross-section of the worm wheel tooth is determined on the basis of various criteria depending on the field of application of the transmission and operating conditions. In most literature, this measure is taken equal to the thickness of the tooth head $\delta_{WD}=s_{a2}$ (Fig.5). Within the framework of this condition, the permissible wear limit for a normal cross-section of a worm wheel tooth can be determined by the following expression [1]:

$$\delta_{WD} = m \cdot \cos\gamma_m \left(\frac{\pi}{2} - 2tg\alpha_0 \right)$$

Where m is module, mm; α_0 is the gearing angle along the dividing circle of the worm wheel, in standard gears is assumed $a_0 = 20^0$.

The wear intensity of the working flanks of the worm wheel tooth can be determined by the following expression [1]:

$$J_W = J_{OT} \cdot W_{ML} \cdot W_{NS}$$

Where J_{0T} is basic value of wear intensity; W_{ML} is the material-lubricant factor; W_{NS} is the starting factor. (In long-term operation, W_{NS}=1,0 is accepted).

The medium glide path on the tooth of the worm wheel s_m^* is determined depending on the shape of the worm profile and transmission parameters.

A comparative analysis of the predicted service life of the worm wheel according to the criteria of pitting and wear is of great practical importance. For the purpose of comparative analysis, a dimensionless coefficient $K_{WH} = N_{WL}/N_{HL}$ was applied. At the end of the second stage of operation, i.e., at the moment when the pitting effect gets the maximum value, it is possible to increase the transmission life by the value $\Delta N_L = N_{LI} + N_{LII}$ by replacing the working flanks of the worm wheel teeth with non-working flanks (Fig. 4).

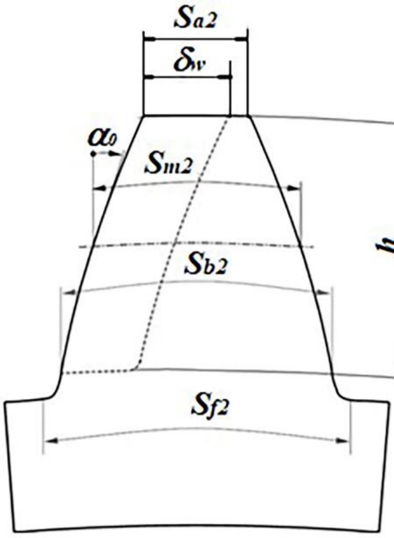

Fig. 5. Worm wheel tooth wear parameters

At this time, the following expression can be used to estimate the relative increase in service life:

$$\Delta L(\%) = \frac{N_{LI} + N_{LII}}{N_{LI} + N_{LII} + N_{LIII}} \cdot 100\% \tag{3}$$

Using the equations (1) and (2) in (3), calculations were performed on various values of the gear ratio (u) and the number of cycles (n_1) of the worm gear with an center distance a=160 mm, the torque on the shaft of the worm wheel T_2= 4900 Nm. The material of the worm was adopted hardened steel grade 16MnCr5 (surface hardness 58÷62 HRC), and the material of the worm wheel CuSn12Ni2-C-GZ (σ_{Hlim}=520 N/mm^2), obtained as a result of casting by centrifugal method.

The values of other parameters required for calculations are set according to the [1]. The results of the calculations are shown in Fig. 6. As can be seen from the figure, as the number of worm shaft cycles increases, the values of the relative increase in service life time also increases. This is due to the fact that at low worm speeds, the stages of formation and intensive pitting increase (N_{LI} and N_{LII}) are short, and the process of intensive wear (N_{LIII}) begins quickly. Calculations show that by replacing the working profiles of the wheel teeth with non-working profiles, when the rotation speed of the worm shaft n_1=3000 min^{-1}, it becomes possible to increase the transmission life by 40÷55%. Obviously, the gear ratio also has a significant impact on the relative increase in service life. With a large gear ratio, the possibility of increasing the resource becomes greater. At u=20.5, it is possible to increase the resource by 25-45% depending on the rotation speed, and at u=50 - by 30-55%.

One of the main conditions that with the proposed method it is possible to increase the life of the worm gear is the absence of the possibility of tooth breakage under the influence of bending stresses. Intensive wear of the working surface of the tooth at the third stage can lead to a decrease in its thickness and, consequently, breakage as a result of loss of bending strength. For this reason, a comparative analysis of service life according to the criteria of bending and contact stresses of the worm wheel tooth is of particular importance.

To determine the predicted service life by the criterion of bending stress, it is necessary to calculate these stresses. Based on [1], the bending stresses arising in the tooth of the worm wheel can be determined by the following expression:

$$\tau_F = \frac{F_{t2}}{b_2 \cdot m} \cdot Y_\varepsilon \cdot Y_F \cdot Y_\gamma \cdot Y_k \,. \tag{4}$$

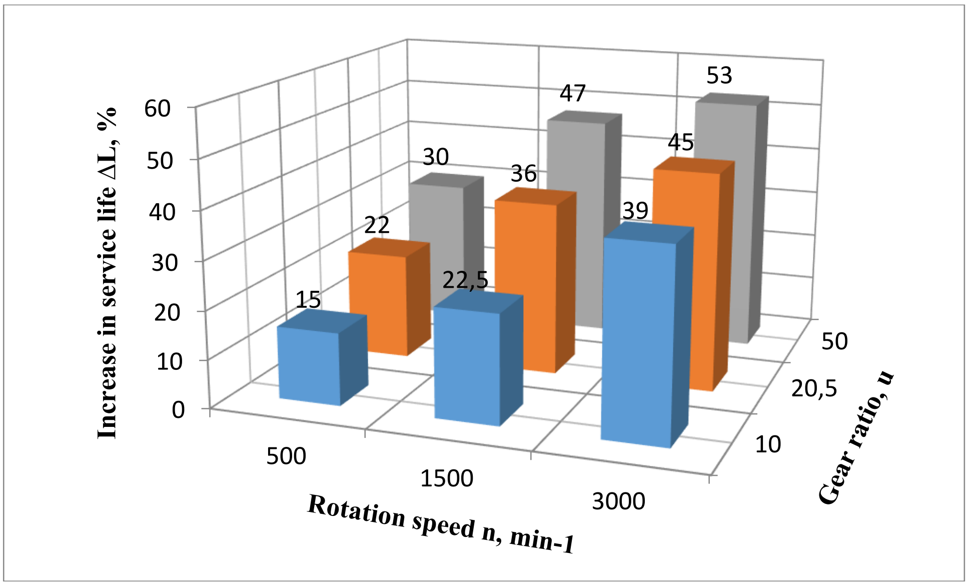

Fig. 6. The relative increase in service life time for worm gear with a=160 mm and T_2= 4900 Nm

Where F_{t2} is tangential force on the worm gear wheel; b_2 is width of worm gear wheel; Y_ε is the contact ratio factor for tooth root stress; Y_γ is the Helix angle factor for tooth root stress; Y_k is wheel rim thickness factor.

As can be seen from equation (4), the bending stress gradually increases from τ_{F1} to τ_{F2} depending on the wear of the tooth (Fig. 7). Where τ_{F1} is the bending stress, in the absence of wear (ΔS=0). Since the wear of the tooth at the first and second stages of operation is insignificant, we can assume that the value of the bending stress at these stages is equal to τ_{F1}. At the third stage of operation (the stage of intensive wear), the bending stress can gradually increase to a value of τ_{F2}, causing tooth breakage (if the wear values before tooth breakage do not exceed the permissible limit). Therefore, to simplify the calculations, we assume that the value of the bending stress at the last stage is equal to τ_{F2}. At the same time, it is also ensured that a certain margin of safety is obtained for bending stress.

In the case when the third stage of operation end due to the breakage of the worm wheel tooth under the influence of bending stresses τ_{F2}, the length of this stage can be approximately determined by the following equations:

$$\text{İf } \tau_{F2} \geq \tau_{F\lim}, \text{ then } N_{LIII} = N_{FD} \cdot \left(\frac{\tau_{F2}}{\tau_{F\lim}} \right)^{k} \tag{5}$$

$$\text{if } \tau_{F2} < \tau_{F\lim}, \text{ then } N_{LIII} = N_{FD} \cdot \left(\frac{\tau_{F2}}{\tau_{F\lim}} \right)^{-(2k-1)} \tag{6}$$

Where $\tau_{F\lim}$ is the endurance limit; N_{FD} is the number of cycles, corresponding to fracture of fatigue curve; k is the fatigue curve indicator. For worm wheels made of bronze, k=6 is accepted [6].

The Mayer-Haibach hypothesis [15] was used in equation (6) above when $\tau_{F2}<\tau_{F\lim}$.

Thus, taking into account the recommendations in [16], depending on the values of the bending stress acting at all stages of operation τ_{F1} and τ_{F2}, it is possible to determine on the basis of graph in Fig. 7 the predicted service life of the worm wheel, as follows:

$$\text{if } \tau_{F1} \geq \tau_{F\lim} \text{ and } \tau_{F2} \geq \tau_{F\lim}, \text{ then } N_{FL} = N_{FD} \cdot \tau_{F\lim}^{k} \frac{N_{LI} + N_{LII} + N_{LIII}}{\tau_{F1}^{k}\left(N_{LI} + N_{LII}\right) + \tau_{F2}^{k} N_{LIII}}, \tag{7}$$

$$\text{if } \tau_{F1} < \tau_{F\lim} \text{ and } \tau_{F2} < \tau_{F\lim}, \text{ then } N_{FL} = N_{FD} \cdot \tau_{F\lim}^{2k-1} \frac{N_{LI} + N_{LII} + N_{LIII}}{\tau_{F1}^{2k-1}\left(N_{LI} + N_{LII}\right) + \tau_{F2}^{2k-1} N_{LIII}}. \tag{8}$$

In the latter equations, the value of the $N_{LI}+N_{LII}$ should be determined by the formula (1), and the values of the N_{LIII} should be determined by the formulas (5) and (6).

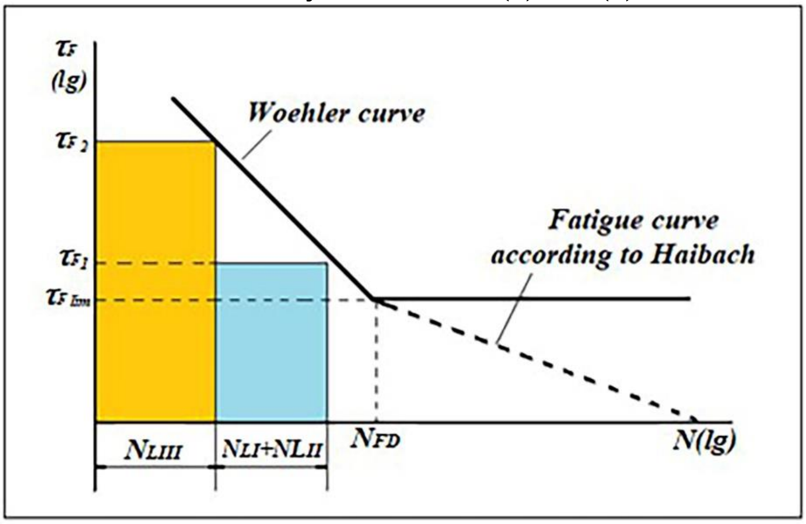

Fig. 7 Bending stress fatigue curves for worm wheel

To increase the service life by replacing the working flanks of the worm wheel teeth with non-working flanks after a certain period of operation, the estimated resource of the teeth according to the bending criterion should be sufficiently large than the predicted resource according to other criteria. Therefore, a comparative analysis of the worm wheel service life by the criterion of bending and contact stresses, as well as by the criterion of bending and wear is of particular importance. Let's consider a comparative analysis of tooth resources by strength conditions by bending stress and by contact stresses. To this end, it becomes necessary to study the dependence of the dimensionless parameter $K_{FH}=N_{FL}/N_{HL}$ on various factors.

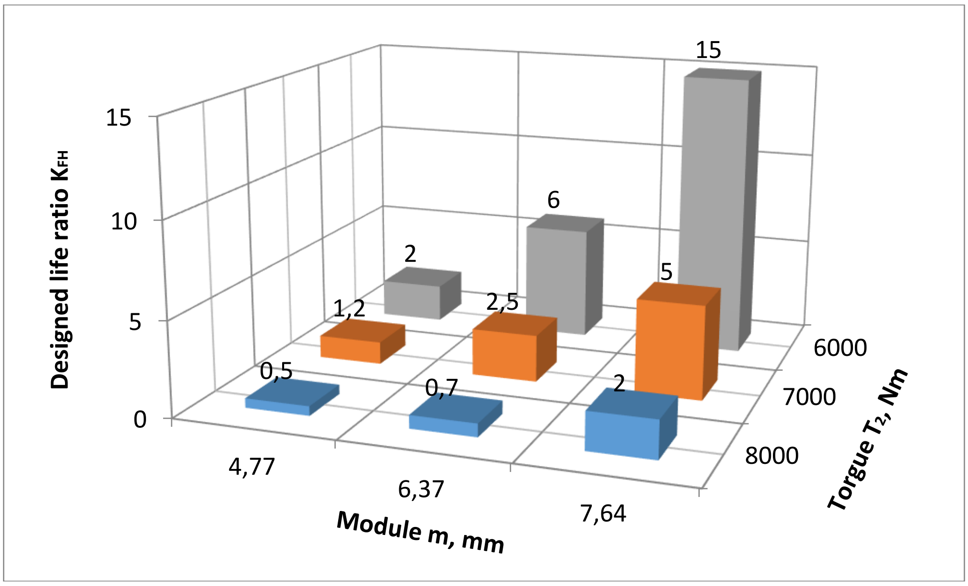

Fig. 8. Service life ratio K_{FH} for worm gear wheel made of CuSn12-Ni2-C-GZ for different modules

For a worm gear with an axial distance a_w=160 mm, a gear ratio u=20.5, the values of the K_{FH} parameter are set for various values of the torque (T_2), module (m) and the number of cycles (n_1) of the worm shaft. The material of the worm was adopted hardened steel grade 16MnCr5 (surface hardness 58÷62 HRC), and the material of the worm wheel CuSn12Ni2-C-GZ (σ_{Hlim}=520 N/mm^2, τ_{Flim}=100 N/mm^2, $N_{FD}=5\cdot10^6$), obtained as a result of casting by centrifugal method. The calculation results are shown in Fig. 8 and 9. As can be seen from the graphs, with an increase in the load on the worm wheel, the value of the service life ratio decreases. But at nominal load limits, the value of this parameter is greater than 2 with a module m=4,77 mm. At large values of the module (m=6,37 mm and m=7,64 mm), the K_{FH} parameter becomes quite high. And at values of the modulus m<4,77 mm, bending stresses become quite dangerous and the probability of tooth breakage from the effects of these stresses becomes higher. The rotation speed of the worm shaft (n_1) also has a fairly large effect on the service life ratio K_{FH}. Under heavy loads, the K_{FH} parameter decreases when n_1 increases.

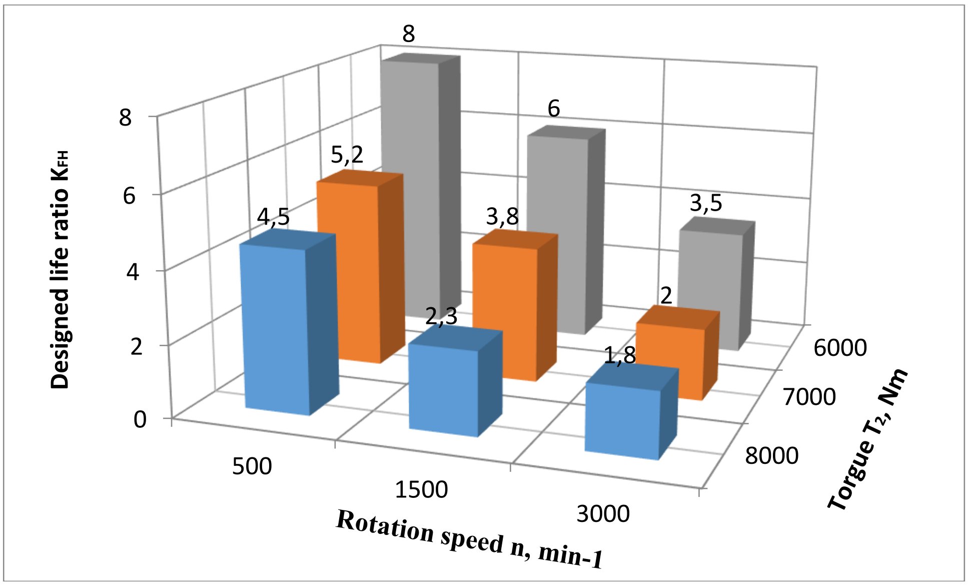

Fig. 9. Service life ratio K_{FH} for worm gear wheel made of CuSn12-Ni2-C-GZ for different drive speeds

In [8] reflects the results of a large number of experimental tests conducted at different values of the output shaft torque (T_2) and the rotation speed of the worm shaft (n_1) at different values of geometric and kinematic parameters (Table 1). The tests were carried out on a specially designed testing facility in the laboratory of the Institute “Machine Elements” of the Technical University of Munich. These experiments were carried out on worm gears consisting of a cylindrical worm with an involute profile made of steel grade 16MnCr5 and a worm wheel made of bronze grade CuSn12Ni-GZ, obtained by centrifugal casting. The analysis of the obtained results shown that in the worm gear tested, the value of the coefficient K_{FH} is greater than 1.

Table 1. The main parameters and indexes of service life of the tested worm gears

Basic transmission parameters	Rotation speed of the worm shaft n_1, min^{-1}	Output shaft torque T_2, Nm	Amount of wear Δm, q	Service life, 10^6 number of cycles		Service life ratio, K_{WH} (K_{FH})
				N_{HL}	N_{WL} (N_{FL})	
a=65 mm; u=20,5; m=2,5 mm	1470	210	12,36	13,96	27	1,94
	1470	290	47,9	1,99	26,46	13,3
a=100 mm; u=10,3; m=5 mm	500	1180	71,5	0,58	7,3	12,6
	500	1696	124,4	0,29	0,85	2,93
	1500	750	95,5	6,7	42,4	6,33
	1500	750	88,4	4,96	40,68	8,2
	1500	1180	108,8	2,94	10,71	3,64
	1500	1180	118	2,13	12,81	6
	1500	1180	83,98	2,51	6,49	2,6
	1500	1180	159	2,24	12,41	5,54

	1500	1696	88,6	1,31	3,05	2,33
	1500	1696	105,5	1,31	2,61	2
	2800	1180	174,9	5,26	22,26	4,23
	2900	1180	124,7	5,95	13,82	2,32
a=100 mm; u=20,5; m=4 mm	1500	700	81,7	6,06	19,48	3,21
	1500	700	140,2	2,73	13,04	4,77
	1500	700	89,7	5,27	17,93	3,4
a=160 mm; u=20; m=6,25 mm	2900	1470	396,3	2,51	12,54	5
	2900	1470	121	6,64	12,6	1,9

In general, the change in the tooth flanks of the worm gear wheel can be realized with the following procedures:

1. **Change in direction of rotation of the electric motor.** This very simple method can only be used in the same cases if the direction of rotation of the motor does not affect the operation of the machine.
2. **Inversion of the gear wheels.** This process can only be realized for worm gear wheels in general mechanical engineering, if the wheels are not made with shaft from one piece. In this procedure, the necessary maintenance work (disassembly, inversion and installation of the gearwheels) is to be implemented, and the direction of rotation of the electric motor is not required.

Conclusions. Based on the research conducted, the following conclusions can be made:

1. The service life ratio of worm gear wheels according to the criteria of bending and contact stresses depends on the geometric and kinematic parameters of transmission, mechanical characteristics of materials. Increasing the module and wheel width allows you to increase the service life ratio;

2. Replacing the working profiles of the teeth with non-working ones after a certain period of operation, ensuring the necessary strength for bending stresses by the correct choice of geometric parameters and materials, increases the lifetime of the worm wheel gears;

3. It has been proved by calculations that due to this procedure, the service life of the worm gear can be increased up to 60%. For this, the flank change must be realized at the end of the pitting growth phase.

Acknowledgements.

The paper was created as part of a cooperation with the Chair of Machine Elements at the Technical University of Munich (FZG). Therefore, the author would like to thank the staff of the FZG for their help, especially Mr. Philipp Roth.

REFERENCES

[1]. DIN 3996: *Tragfähigkeitsberechnung von Zylinder-Schneckengetrieben mit sich rechtswinkling kreuzenden Achsen.* Beuth-Verlag, 2012, Berlin, Köln. (in German)

[2]. Khurmi, R.S., Gupta, J.K. *Machine Design*, Eurasia Publishing House, 2005, Ram Nagar, New Delhi.

[3]. Decker, K. *Maschinenelemente: Funktion, Gestaltung und Berechnung*. 2018, Carl Hanser Verlag, München, Wien. (in German)

[4]. Monz, A. *Tragfähigkeit und Wirkungsgrad von Schneckengetrieben bei Schmierung mit konsistenten Getriebefetten.* (Load capacity and efficiency factor of worm gears lubricated with consistent gear greases), 2012, Ph.D. thesis TU Munich. (in German)

[5]. Kishore, K., Mukhopadhyay, G., Adhikary, M. *Metallurgical Analysis of Premature Failure of a Phosphor Bronze Worm Wheel.* J. Fail. Anal. Prev. 2018, 18, pp. 1321–1326. https://doi.org/10.1007/s11668-018-0554-x

[6]. Thiele, R. *Untersuchungen zur Zahnfußbeanspruchung von Schneckenrädern und Entwicklung eines Tragfähigkeitsnachweises auf Basis der Zahnfußschädigungshypothese*. (Investigations on the tooth

root stress of worm gears and development of a load-bearing capacity verification based on the tooth root damage hypothesis). 2006, Ph.D. thesis TU Chemnitz:. (in German)

[7]. Reißmann, J., Leidich, E. *Calculation of the tooth root strenght of worm wheels on local stresses. Garching: International conference on gears*, VDI-Berichte, 2013. ISBN: 978-3-18-092119-0

[8]. Rank, B. *Versuche zur Grübchentragfähigkeit von Schneckengetrieben: Untersuchungen an Zylinder-Schneckengetrieben* - Grübchenbildung, Verschleiß. – München: TU München. FVA-Forschungsvorhaben 12/IV. Heft 494, 1996. (in German)

[9]. Heilemann, J. *Tragfähigkeit und Wirkungsgrad bei unterschiedlichen Schnecken-Zahnflankenformen unter Berücksichtigung der Oberflächenhärte und Härtetiefe* (Bearing capacity and efficiency for different worm tooth flank shapes, taking into account the surface hardness and hardness depth), 2005, Ph.D. thesis TU Munich. (in German)

[10]. Weisel, C. *Schneckengetriebe mit lokal begrenztem Tragbild* (Worm gearboxes with locally limited tooth bearing), 2009, Ph.D. thesis TU Munich. (in German)

[11]. Sigmund, W. *Untersuchung und Simulation des Verschleißverhaltens von Schneckengetrieben mit unvollständigem Tragbild* (Investigation and simulation of the wear behavior of worm gears with incomplete tooth bearing). 2015, Ph.D. thesis TU Munich. (in German)

[12]. Mautner, E.M., Stemplinger, J.P., Stahl, K.: FVA-Nr. 503 II - Heft 1162 - *Schnecken-getriebe-Baugrößeneinfluss II Abschlussbericht* - *Verschleiß- und Grübchentragfähigkeit von großen Zylinder-Schneckengetrieben mit optimierter Radbronze*. 2015, Forschungsvereinigung Antriebstechnik e.V., Frankfurt/Main.

[13]. Tschalabi, I. *Lebensdauererhöhung von Schneckengetrieben durch Flankenwechsel der Radzähne* / I. Tshalabi, E. Leidich, J. Reißmann //ANT Journal. 2012, Nr. 12, pp. 14-17.

[14]. Chalabi, I. *Comparative Service Life Analysis for Gears According to Different Failure Criteria.* J. Fail. Anal. Prev. 2020, 20. pp. 2137-2144. https://doi.org/10.1007/s11668-020-01029-y

[15]. Haibach, E. *Betriebsfestigkeit-Verfahren und Daten zur Bauteilrechnung*, 2006, VDI-Verlag GmbH, Dues seldorf.. (in German)

[16]. Naunheimer, H., Bertsche, B., Ryborz, J., Novak, W. *Automative Transmissions: Fundamentals, Selec tion, Design and Application*, 2011, Springer-Verlag, Berlin Heidelberg.

Received: 14.08.2022
Accepted: 02.11.2022