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FATIGUE TESTING OF TRANSMISSION GEAR

DOŚWIADCZALNE BADANIA TRWAŁOŚCI ZMĘCZENIOWEJ PRZEKŁADNI ZĘBATYCH*

This paper presents the results of experimental tests of fatigue life of selected gears, performed on a test stand equipped with a hydraulic universal testing machine. The tests were performed on skew and straight cylindrical gears, made of EN AW-2017A and EN AW-7057 aluminium, and 40HM steel. Moreover, fatigue life curves for selected gears were presented, and the mechanisms of the occurrence of damage were analysed. Relationships describing the maximum value of torque during a loading cycle in relation to the number loading of cycles until the gear is damaged were also proposed.

***Keywords**: Life cycle fatigue, gear box, experiment, cracking, damage.*

W pracy przedstawiono wyniki badań doświadczalnych trwałości zmęczeniowej wybranych przekładni zębatych, wykonanych na opracowanym stanowisku badawczym wyposażonym w hydrauliczną maszynę wytrzymałościową. Badania przeprowadzono na walcowych kołach o zębach prostych i skośnych, wykonanych ze stopów aluminium EN AW-2017A i EN AW-7057 oraz stali 40HM. Ponadto zaprezentowano wykresy trwałości zmęczeniowej wybranych przekładni zębatych oraz przeanalizowano mechanizmy powstawania uszkodzeń. Zaproponowano także zależności określające maksymalną wartość momentu skręcającego w cyklu obciążenia od liczby cykli obciążenia do uszkodzenia przekładni.

***Słowa kluczowe**: Trwałość zmęczeniowa, przekładnia zębata, eksperyment, pęknięcia, uszkodzenia.*

1. INTRODUCTION

Working loads of construction elements, especially cyclically changing loads, cause nucleation and the development of damage in the material, which often leads to fatigue destruction of the whole element [9, 10]. In the case of uniaxial or proportional biaxial loads, the damage cumulates on privileged surfaces; the life of material is determined on the basis of the results of standard tests presented in the form of fatigue curves [14, 15]. The prediction of fatigue life of construction elements that operate in conditions of nonproportionate loads (which occurs in the case of cylindrical gears) is a huge computational problem [5, 12]. The difficulties are connected with the necessity to formulate and experimentally verify general criteria descriptions allowing for the cumulation of damage on different physical surfaces, and to establish the surface of crack initiation and the crack criterion [10, 19].

The development of damage, and then the initiation of fatigue cracking in gears is particularly intensive in two areas: in the area of contact of gear teeth (from contact pressures) and at the base of the loaded tooth (from twisting and shearing).

In the former case, gear damage is the result of local crumbling on the surfaces of the mating teeth (mainly pitting wear caused by high values of contact, normal, and tangential stresses) [18]. In the latter case, damage to the element is the result of fracture to the teeth base (propagation of fatigue cracking until the whole of the tooth breaks off). It should be added that fatigue cracking in mating gears may appear both in the outer layer of the tooth, and inside the material - near the border between the outer layer and the core [4].

Prediction of the development of fatigue damage in gears as early as at the stage of their design allows to determine the lifespan of a given gear in conditions of normal operation, and avoid serious damage to the whole device. Fatigue calculations for gears usually consist in determining the fatigue life of the tooth base [7]. The computational procedure consists in determining the infinite fatigue life of a gear, which is expressed as the value of the normal stress at tooth base which the rim material can transfer without breaking it during at least 3x106 loading cycles [3]. This value is too small, as gears often work in such manner that the number of loading cycles is considerably higher. For comparative calculations, the values of infinite fatigue life obtained in tests of smooth samples at uniaxial tension-compression or uniaxial pulsation from-zero bending are used [17]. Another method for the determination of fatigue life of gear teeth requires the creation of a fatigue life curve on the basis of experimental tests of real gear pairs in operating conditions [10]. Most of the available papers connected with fatigue tests of gears are based on calculations with the use of the finite element method. There are fewer papers devoted to experimental verification and fatigue tests of real life gears.

This paper proposes an own design of a test stand that differs significantly from the solutions currently in use. Employing a hydraulic universal testing machine and a torsion torque sensor resulted in an accurate representation of mating of individual contact pairs [20]. The stand allows to determine the fatigue life of gears, i.e. the relationship between the maximum torsion torque during a cycle and the number of loading cycles causing the initiation of fatigue cracking on the contact surface or at the base of the tooth, using only one gear for the tests.

2. STANDS FOR FATIGUE TESTING OF GEARS

Up to now, the most commonly used stand for experimental testing of the fatigue life of gears has been the power-closed-loop test stand, often called the circulating power stand [11]. A scheme of the "power-closed-loop" gears is shown in Fig. 1. It consists of two one-

(*) Tekst artykułu w polskiej wersji językowej dostępny w elektronicznym wydaniu kwartalnika na stronie ***www.ein.org.pl*** step gears with the same transmission ratios, the so-called test gears 1 and closing gears 2, two torsion bars 3 and 4, tightening clutch 5 and medium-power electric motor (generally 6 to 12 kW) 6. In the test gears there are the two tested gears, while in the closing gears – the gears closing the circuit whose life is much higher in comparison with the test gears. One of the key elements of the power-closed-loop stand is the loading unit. For this purpose, tightening clutch 5 is used the most commonly, enabling the turning of bars 3 of the gears with the appropriate torque.

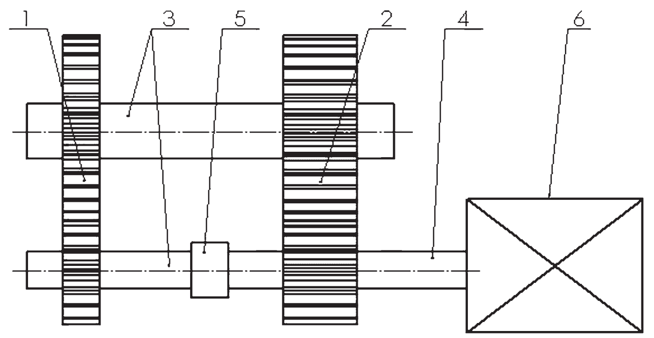

Fig. 1. Power-closed-loop stand 1 – test gears, 2 – closing gears, 3 – torsion bars, 4 – bar plug for the connection with the motor, 5 – tightening clutch, 6 – motor [11]

The scheme of the construction of the classic stand is shown in Figure 2. It consists of test gears 1, clutch or loading brake 3 mounted on one of the gear shafts, and motor 2 (e.g. electric), which forces the load torque of the gears. Classic stands find their use in dimensionally small gears, which yield small load torques.

In the latest version of the stand for testing toothed gears, a hydraulic method of loading is used. This allows to load the tested gears with a constant torque, similarly to the classic stand with mechanical tightening pre-set before commencing the test (Fig. 2). This solution allows to apply torsion torque in a changeable (programmed) manner, automatically during the performance of a test (without stopping the stand). This change may occur in a continuous, discrete, or even random manner.

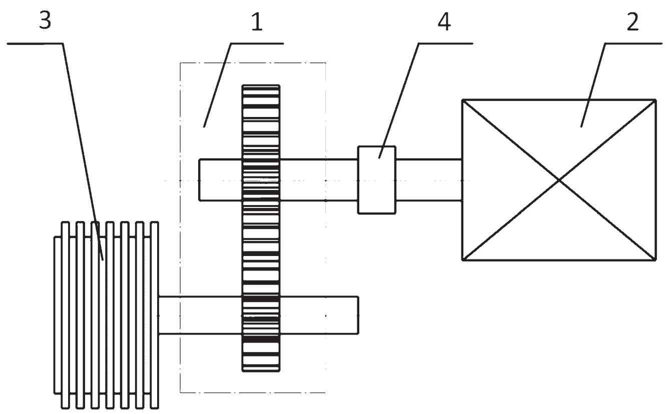

Fig. 2. Classic stand for toothed gears testing, 1 – body of the tested toothed gears, 2 – motor, 3 – loading clutch or electromechanical brake, 4 – clutch [3]

Stands for tests in the gigacycle range usually consist of: a computer, an analogue-to-digital converter, a device inducing oscillations (e.g. inductor), with a frequency of 20 kHz or more, and the tested toothed gears. A block scheme of this kind of stand is presented in Figure 3 [8].

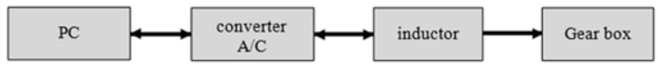

Fig.3. Block scheme of a stand for testing gigacycle fatigue life [8]

The results of the experimental tests on the described stands are, above all, the relationships between the maximum value of torsion (propulsive) torque during a loading cycle and the number of loading cycles until the destruction, obtained on the discussed stand.

Example fatigue characteristics are presented in Fig. 4. Line 1 represents infinite fatigue life. In the case of curve 2 the necessity for determining fatigue life also in the range of a very high number of cycles can be easily observed [13].

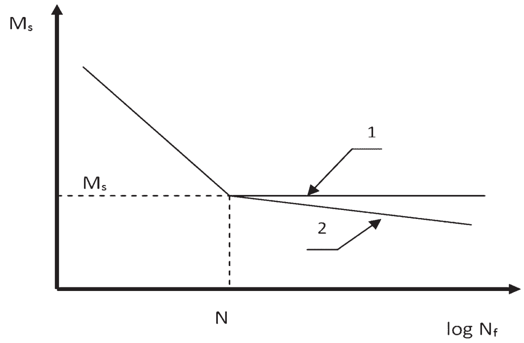

Fig. 4. Fatigue life curve of toothed gears; 1 – infinite fatigue life, 2 – gigacycle fatigue life [13]

3. METHODOLOGY OF TOOTHED GEARS TESTING

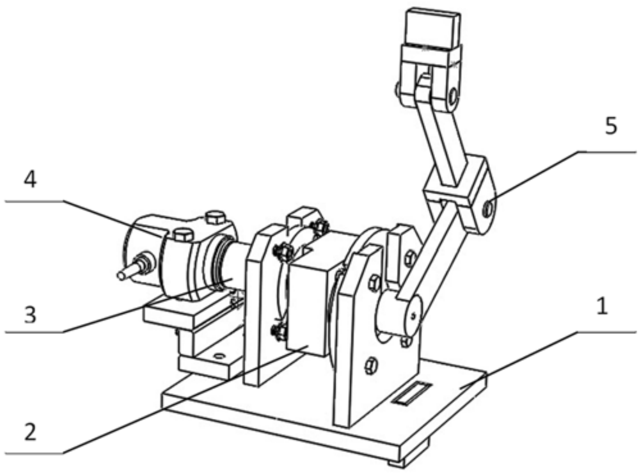

Fig. 5. Test stand device for the determination of the fatigue life of gears; 1 – mounting base, 2 – gears, 3 – clutch, 4 – torque sensor, 5 – crankshaft mechanism.

The paper presents an original stand for the determination of the fatigue life of gears. When designing it, the authors' aim was to reflect the operating and technical conditions of the tested pair of gears as accurately as possible. The presented test stand consists of base 1 mounted to the holder of the universal testing machine (MTS 322 Test Frame), which allows to apply a programmable load curve. To the base, by means of adequately profiled flanges, gears body 2 is mounted, inside which the tested pair of gears is located. The output shaft of the gears is mounted by means of clutch 3 with torque sensor 4 that collects data during the test. The drive shaft of the gears is connected through crankshaft mechanism 5 with a second holder of the universal testing machine, thus closing the kinematic chain of the load [20].

In the presented solution, in order to apply load, crankshaft mechanism 5 is used, which changes programmed reciprocating motion of the inductive actuator of the universal testing machine into rotational motion of the drive shaft of tested gears 2. The torsion torque at the output shaft is recorded in real time by means of strain torque sensor 4. The measurement circuit enables acquisition of the value of the measurement signal from the torque sensor and the value of the respective forcing load.

The presented stand allows to determine the fatigue life of toothed gears representing real operating conditions, i.e. the susceptibility of shafts, bearing nodes, and elements of torque transfer. Moreover, a single pair of mating gears allows to determine the whole fatigue characteristic of the gears. Each point on the fatigue curve is determined on a single pair of mating teeth, or two at most in the case of two pairs of teeth intermeshing. After finishing the test at a given load level, the gear shaft rotates, so that the next undamaged pair of teeth intermeshes, then the process of cyclic loading is repeated. Owing to this approach it is possible to reduce the cost of fatigue tests by lowering the number of test samples (gears).

The initial parameters in the presented stand model are: maximum torque $M_{s\max}$ applied by maximum linear displacement of the actuator of the universal testing machine to the drive shaft of the gears, and the frequency of load application *f*. The initial angle of rotation of shaft α_g is determined indirectly from the geometry of the crank mechanism. The influence of the following parameters on the fatigue life of the gears can be determined on the described stand: gear ratio, type of cooling and lubricating fluid, material properties, teeth shape, and processing technology. During the tests recorded are the torsion torque of the gears' output shaft versus time curve, the number of loading cycles, and the angle of rotation of α_g.

The scheme of the method for the determination of the fatigue life of wheels of the tested gears on the presented stand are shown in Figure 6. The view of the test stand is shown in Figure 7.

Fig. 7. View of the test stand; 1 – universal testing machine, 2 – body of the stand, 3 – tested gears

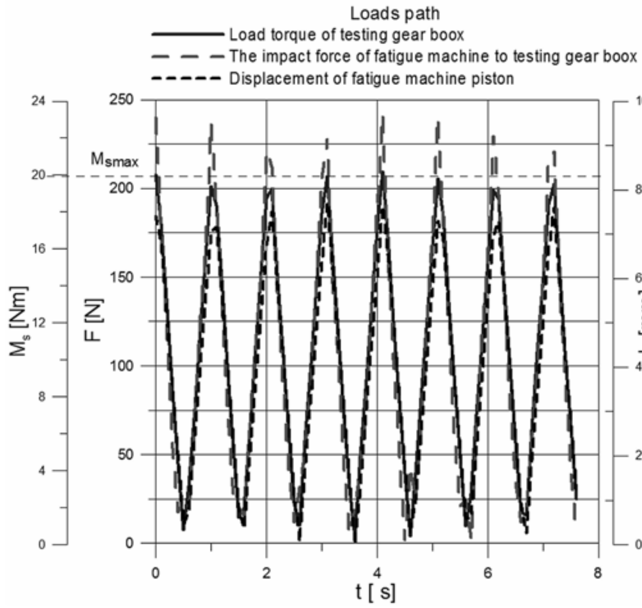

Fig. 8. Example load curves implemented in the performed tests

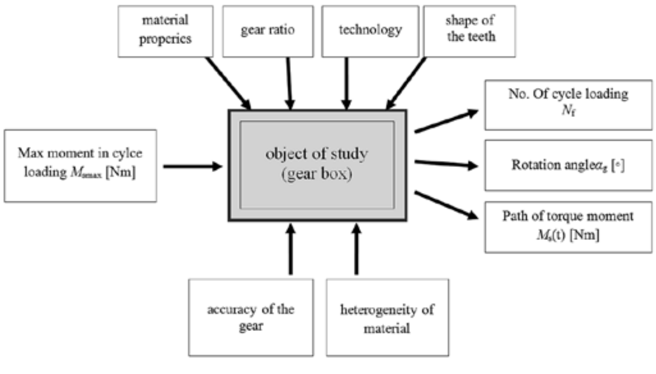

Fig. 6. Scheme of fatigue tests of gears

The testing process may be disrupted by imperfections in the workmanship of the tested gears, material inhomogeneity, and play in bearing nodes.

Figure 8 shows example curves of the from-zero torque loading the tested gears and the respective curves of force and displacement of the actuator of the universal testing machine. The character of the torque curve is similar to one occurring in real life operating conditions. Initiation and development of fatigue cracks causes an increase of the susceptibility of the mating gear teeth, and thus an increase of the angle of shaft rotation α_g. On its basis, the moment of gear damage caused by initiation and propagation of fatigue cracks in the tooth base or on the contact surface of teeth is determined.

4. Experimental tests of fatigue life of gears

The experimental tests were carried out on both straight and skew cylindrical gears. The gears made of EN AW-2017A and EN AW-7057 aluminium alloys had straight teeth with a normal module of 1.5 mm, while the gears made of 40HM steel had skew teeth and a module of 1 mm.

The fatigue tests were carried out in the laboratory of the Department of Mechanics and Applied Computer Science at Faculty of Mechanical Engineering of Białystok University of Technology. For each of the load levels, three repetitions were performed. The value of the determined loading cycles until fatigue cracking is initiated is shown in table 2 and Figure 10.

The next stage of the experiment were tests of skew gears. The results of the tests are presented in Figure 13 and table 6.

On the basis of the performed tests, two different mechanisms of fatigue damage to gears were observed. In the first one, cracking of the tooth base occurred. Fatigue cracks occurred above the bottom land of tooth. This mechanism occurred in the case of loads caused by torsion torque M_s with higher amplitudes. In this mechanism, the dominant stresses are those caused by teeth bending.

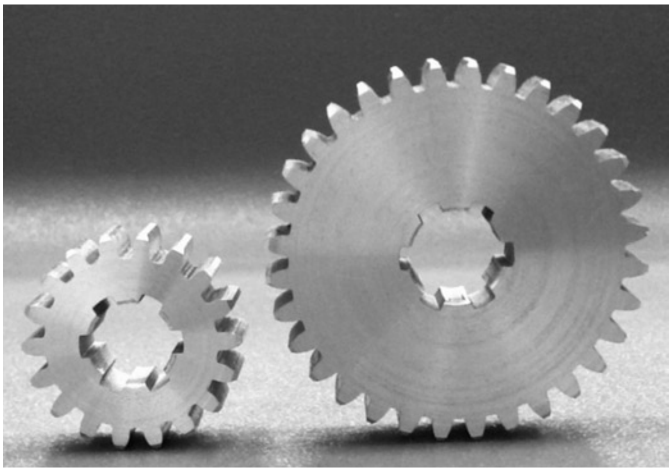

Fig. 9. View of straight-tooth samples used for the tests – gears no. 1 (Tab. 1)

Table 1. Parameters of straight gears – gears no. 1

Parameters	Gears I	Gears II
No. of teeth – Z	32	18
Tooth module – *m*	1.5 mm	1.5 mm
Pitch diameter – d_p	48 mm	27 mm
Extremal diameter – d_z	51 mm	30 mm
Surface roughness – R_a	0.63	0.63
Angle of intermeshing – *α*	20°	20°
Material	EN AW-2017A	
Technology of execution	boundary	
Lubrication	SAE 75W90	

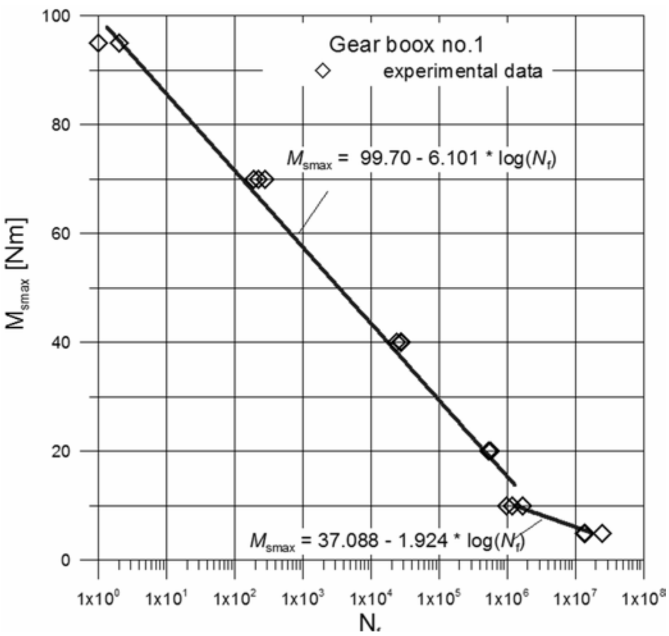

Fig. 10. Gears fatigue life curve – gears no. 1 (Tab. 1)

Figure 14 presents the trajectories of cracks observed in the tests and the respective values of maximum load. The curves of the trajectories of fatigue cracks at gear tooth base observed in the tests corroborate the results obtained on the basis of numerical calculations [2, 6, 13, 16].

Table 2. Results of tests of fatigue life of gears no. 1

Loading M_{smax} [Nm]	Number of loading cycles until the gear are damaged Nf		
	Sample1	Sample 2	Sample 3
95	1	1	2
70	190	280	226
40	24238	26854	27736
20	533304	524200	568112
10	1200542	986735	1699242
5	24523101	14121002	13569321

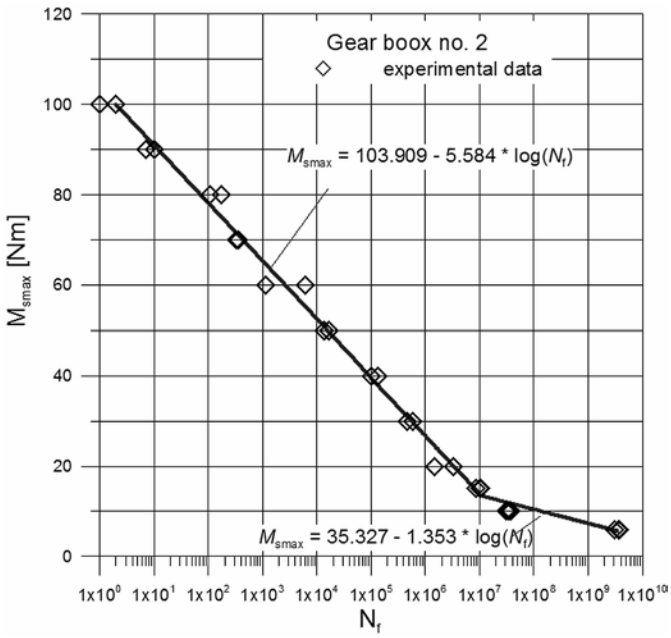

Fig. 11. Gears fatigue life curve – gears no. 2 (Tab. 3)

Table 3. Parameters of straight gears – gears no. 2.

Parameters	Gears I	Gears II
No. of teeth – Z	32	18
Tooth module – *m*	1.5 mm	1.5 mm
Pitch diameter – d_p	48 mm	27 mm
Extremal diameter – d_z	51 mm	30 mm
Surface roughness – R_a	0.63	0.63
Angle of intermesching – *α*	20°	20°
Material	EN AW-7075	
Technology of execution	boundary	
Lubrication	SAE 75W90	

Figures 15–16 present example fatigue cracks of tooth base (the first mechanism of damage to the toothed gear); both in the case of straight gears (Fig. 15), and skew gears (Fig. 16).

The second observed mechanism of gear damage is wear of the contact surface of a tooth. In this case wear to the gear is determined by the values of surface stresses. This type of damage was connected with the action of torsion torque with relatively small amplitudes on the gears.

Table 4. Results of tests of fatigue life of gears no. 2

Loading M_{smax} [Nm]	Number of loading cycles until the gear are damaged N_f		
	Sample 1	Sample 2	Sample 3
100	1	2	3
90	7	10	4
80	106	177	164
70	327	361	411
60	1154	6032	3456
50	16384	13880	10154
40	134104	100088	95860
30	576543	469870	720365
20	1448130	3214827	1148100
15	10050301	10456321	8451111
10	37267451	31267045	35409245
6	726745000	126745966	549245877

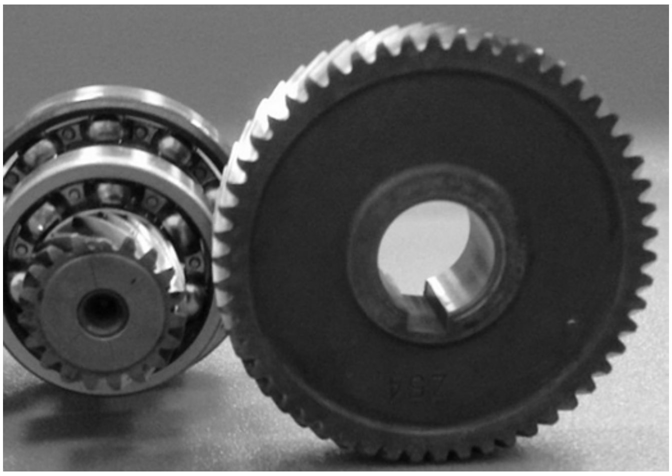

Fig. 12. View of the tested samples of cylindrical skew gears – gears no. 3 (Tab. 5)

Table 5. Parameters of skew gears – gears no. 3.

Parameters	Gears I	Gears II
No. of teeth – Z	54	19
Tooth module – m	1 mm	1 mm
Pitch diameter – d_p	55 mm	19 mm
Extremal diameter – d_z	57 mm	21 mm
Surface roughness – R_a	0.32	0.32
The angle of inclination of the teeth – β	10°	10°
Material	40HM	
Technology of execution	boundary	
Lubrication	SAE 75W90	

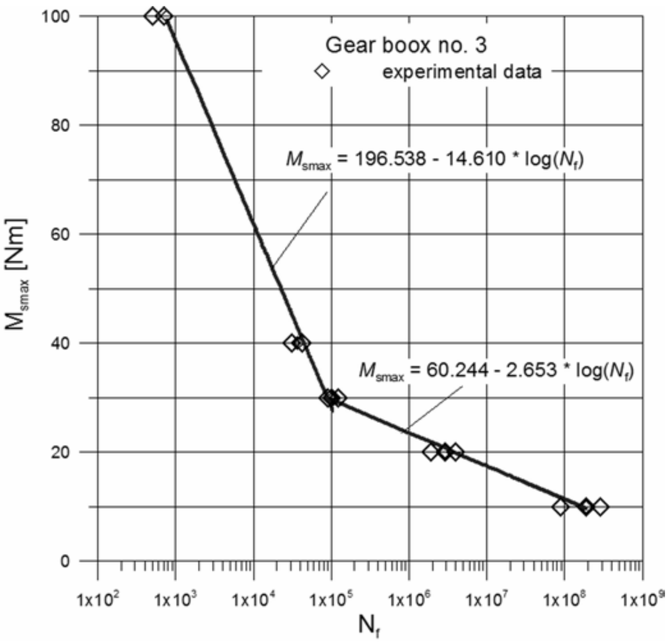

Fig. 13. Curve of fatigue life of gears – gears no. 3 (Tab. 5)

Table 6. Results of tests of fatigue life of gears no. 3

Loading M_{smax} [Nm]	Number of loading cycles until the gear are damaged N_f		
	Sample 1	Sample 2	Sample 3
100	702	511	1102
40	31007	42500	37607
30	90053	121653	100356
20	2915190	3913330	1915190
10	188407160	89466122	289221100

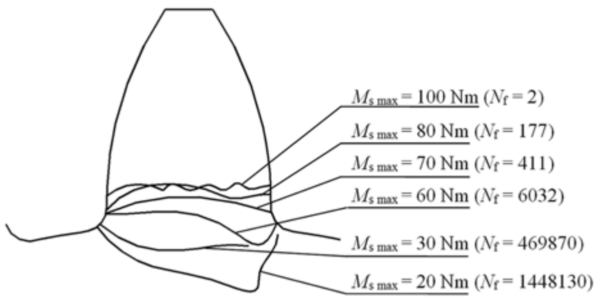

Fig. 14 Scheme of trajectories of fatigue cracks at tooth base in gears no. 2 (tab. 3)

During the tests typical mechanisms of wear of the surfaces of the pair of wheels in contact (pitting and adhesion) were observed [3]. In the case of the pitting mechanism of wear of teeth contact surface, micro-cracks progressing towards the inside of the wheel material were observed (Fig. 17a), while for the adhesive mechanism of wear, fragments of material were torn off without apparent additional micro-cracks (Fig. 17b).

5. PREDICTION OF FATIGUE LIFE OF GEARS

On the basis of the relationships between the maximum torsion torque in a loading cycle and the number of loading cycles causing gears damage obtained in fatigue test, and the performed analysis of the mechanisms of cracking and wear of gears, an attempt at preparing

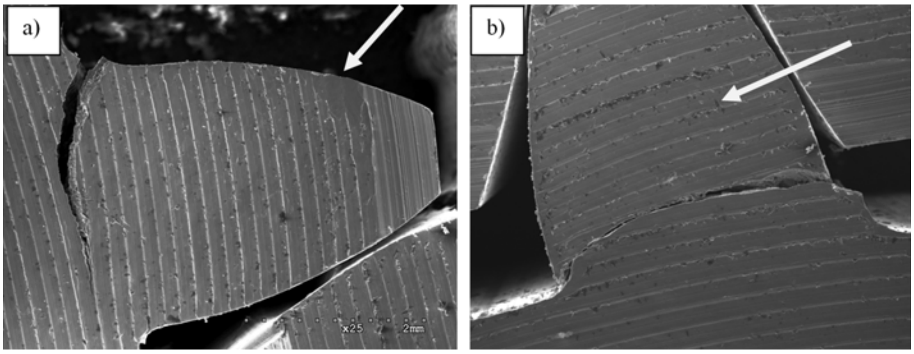

Fig. 15. Cracking in a tooth in gears no. 2 (tab. 3) that occurred as a result of the action of fatigue loads a) maximum torsion torque in a loading cycle M_{smax} = 30 Nm, number of loading cycles N_f = 469870, b) M_{smax} = 70 Nm, number of cycles N_f = 361 in 25× magnification

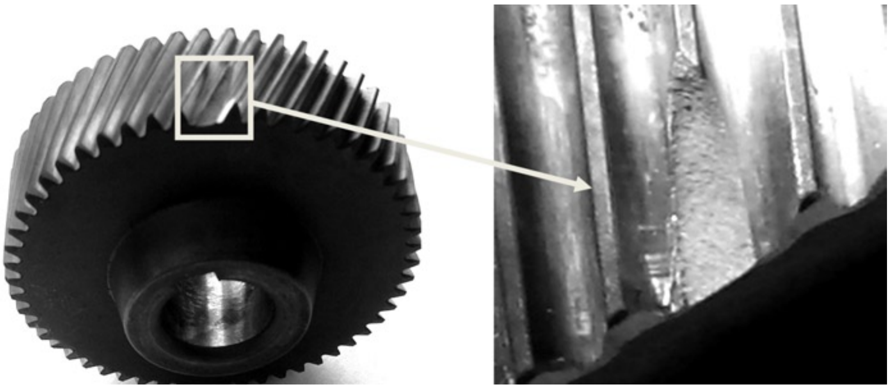

Fig. 16. Cracking of skew gear tooth caused by a cyclically changing load (from-zero pulsating cycle) a) maximum value of torsion torque in a loading cycle M_{smax} = 100 Nm, number of loading cycles N_f= 702

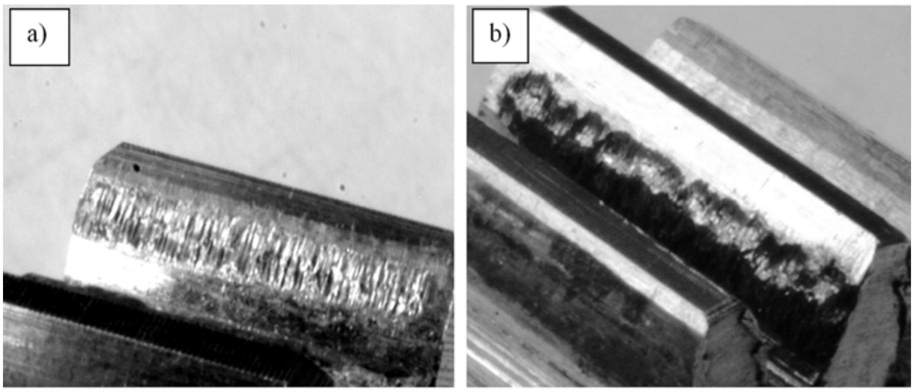

Fig. 17. Damage to tooth surface caused by a cyclically changing load, maximum value of torsion torque in a loading cycle M_{smax} = 20 Nm, number of loading cycles N_f = 3214827, b) M_{smax} = 10212 Nm, N_f= 35409245

semi-empirical relationships describing the fatigue life of gears was made. Figure 18 presents schematic curves of the fatigue life for both mechanisms of gears damage.

Fatigue life in the case in question can be described with the following equations:

$$\begin{aligned} M_{\text{smax}}\left(N_{\text{f}}\right) &= M_{\text{fc}} - \eta_{\text{f}} \log\left(N_{\text{f}}\right) \qquad \text{dla } M_{\text{smax}} \geq M_{\text{p}}, \\ M_{\text{smax}}\left(N_{\text{f}}\right) &= M_{\text{wc}} - \eta_{\text{w}} \log\left(N_{\text{f}}\right) \qquad \text{dla } M_{\text{smax}} < M_{\text{p}}, \end{aligned} \quad (1)$$

where: M_{smax} – maximum torsion torque in a gear loading cycle, N_{f} number of loading cycles until the gears are damaged, M_{fc} – critical value of torsion torque in the gears (causing tooth base cracking), M_{wc}– computational value of torsion torque connected with the second mechanism of damage (wear of teeth contact surfaces), η_{f}, η_{w} – coefficients determined experimentally depending on the parameters of the tested gears for the first (fatigue cracking of tooth base) and the second (wear of tooth contact surface) mechanism of gears damage, respectively.

Table 7 compares the values of the parameters obtained in the tests in relationships describing fatigue life of the tested gears.

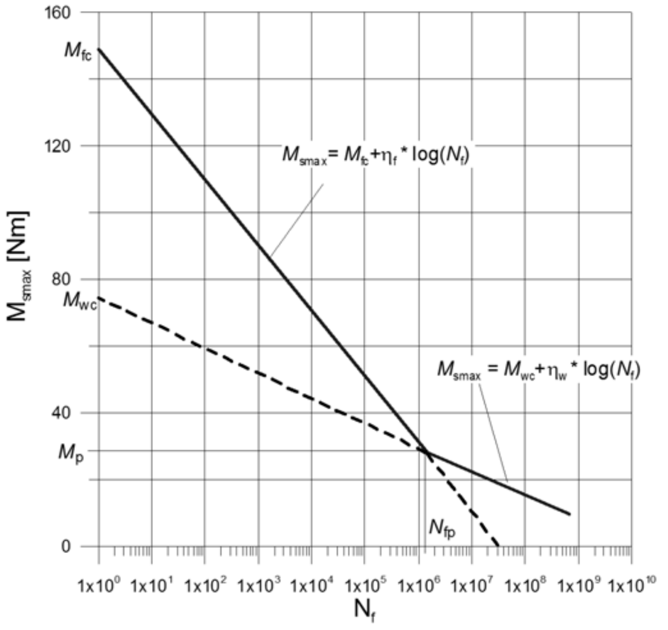

Fig. 18. Schematic formulation of the relationship between the maximum value of torsion torque in gears during a loading cycle and the number of loading cycles for two gears damage mechanisms: cracking of tooth base and wear of contact surface

Table. 7. Comparison of the values of parameters in relationships describing the fatigue life of the tested gears

Gear boox	Value of parameters in eq (1)			
	η_f [Nm]	M_{cf} [Nm]	η_w [Nm]	M_{cw} [Nm]
Gear boox no 1 (tab. 1)	-6,101	99,695	-1,924	37,088
Gear boox no 2 (tab. 3)	-5,584	103,909	-1,353	35,327
Gear boox no 3 (tab. 5)	-14,610	196,538	-2,653	60,244

6. SUMMARY

The paper presents a new stand for the determination of the relationship between the maximum torsion torque during a loading cycle and the number of cycles until gears are damaged, in which cyclically changing loads were applied by means of a hydraulic universal testing machine. On the presented stand it is possible to determine the fatigue life of gears reflecting real life operating conditions, i.e. the susceptibility of shafts, bearing nodes, and elements of torque transfer. Moreover, a single pair of mating gears allows to determine the whole fatigue characteristics of gears. Owing to this approach it is possible to reduce the costs of fatigue tests through lowering the number of test samples (gears).

The designed stand made it possible to perform fatigue tests of straight and skew cylindrical gears, in which the wheels were made from three different materials. The tests yielded information about the mechanisms of initiation and propagation of fatigue cracks, and the mechanisms of wear in gears. An analysis of the obtained results allowed to create semi-empirical relationships describing fatigue life of gears taking into consideration two mechanisms of damage: fatigue cracking of tooth base and wear of the contact surface of teeth.

It should be added that it is necessary to perform additional experimental tests of fatigue life of gears with other construction parameters and made from other materials, in order to verify the presented computational relationships. Recommended are also fatigue tests in the gigacycle range so as to determine the character of computational relationships in this range.

The obtained results of experimental fatigue tests (the relationship between the maximum value of load and the number of cycles until the gears are damaged, trajectories of fatigue cracking) may be used by other researchers to verify computational models, especially those employing the finite element method.

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