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CHARACTERISTICS OF A NEW TEST RIG AND METHODOLOGY FOR CYCLIC TESTING OF GEAR TOOTH BENDING FATIGUE STRENGTH

CHARAKTERYSTYKA NOWEGO STANOWISKA ORAZ METODYKA PULSACYJNEGO BADANIA WYTRZYMAŁOŚCI ZMĘCZENIOWEJ ZĘBÓW KÓŁ ZĘBATYCH NA ZŁAMANIE

Key words: pulsator, gear, tooth fracture, S-N curve, nominal stress number (bending).

Abstract Tooth fracture is the most dangerous form of gear wear that excludes the gear from further use. In order to counteract the occurrence of this type of damage, it is very important to properly design the toothed gear. To calculate the gear tooth bending strength, a strength parameter called the nominal stress number σ_{Flim} is necessary. ISO 6336-5:2003(E) and available material databases provide σ_{Flim} values for the most popular engineering materials used for gears, including those for case-hardened steels. There is, however, no data for a new generation of nanostructured engineering materials, which are the subject of research conducted at the Tribology Department of ITeE – PIB. The σ_{Flim} parameter is most often determined in cyclic fatigue tests on toothed gears with specially selected tooth geometry. In order to determine the above strength parameter, a pulsator (symbol T-32) was developed and manufactured at ITeE-PIB in Radom. The article presents a new device, research methodology, and the results of verification tests for case-hardened steel 18CrNiMo7-6, confirming the correctness of the adopted design assumptions and the developed research methodology. The results of tooth bending fatigue tests are the basis for the selection of a new engineering material dedicated to gears, which later undergoes tribological testing.

Słowa kluczowe: pulsator, koło zębate, złamanie zęba u podstawy, krzywa S–N, granica zmęczenia.

Streszczenie Złamanie zęba u podstawy jest najbardziej niebezpieczną formą zużycia kół zębatych wykluczającą je z dalszej eksploatacji. W celu przeciwdziałania wystąpieniu tego rodzaju uszkodzenia bardzo istotne jest właściwe zaprojektowanie koła zębatego. Do obliczeń wytrzymałościowych kół zębatych na złamanie zęba niezbędny jest parametr wytrzymałościowy zwany granicą zmęczenia σ_{Flim}. Norma ISO 6336–5:2003(E), a także dostępne bazy materiałowe podają wartości σ_{Flim} dla najbardziej popularnych materiałów wykorzystywanych na koła zębate, w tym dla stali nawęglanych. Brak jest jednak jakichkolwiek danych dla nowej generacji nanostrukturalnych materiałów konstrukcyjnych, które są przedmiotem badań prowadzonych w Zakładzie Tribologii ITeE – PIB. Parametr σ_{Flim} wyznacza się najczęściej w badaniach pulsacyjnych na kołach zębatych o specjalnie dobranej geometrii uzębienia. W celu wyznaczenia powyższego parametru wytrzymałościowego w ITeE – PIB w Radomiu opracowano i wytworzono pulsator (o symbolu T–32). W artykule przedstawiono nowe urządzenie, metodykę badawczą oraz wyniki badań weryfikacyjnych dla stali konstrukcyjnej 17HNM – nawęglanej i hartowanej, potwierdzające poprawność przyjętych założeń konstrukcyjnych i opracowanej metodyki badawczej. Wyniki badań zmęczeniowych stanowią podstawę wyboru nowego materiału konstrukcyjnego dedykowanego na koła zębate, które później poddaje się przekładniowym badaniom tribologicznym.

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INTRODUCTION

Tooth fracture is the most dangerous form of gear wear, because it eliminates the transmission from further use, additionally exposing the user to danger. An image of a toothed wheel with broken teeth, coming from a mining gear transmission, is shown in **Fig. 1**.

Fig. 1. View of the gear with broken teeth after exploitation in a coal mine
Rys. 1. Widok koła ze złamanymi zębami po eksploatacji w kopalni węgla kamiennego

During operation, stress occurs, in particular bending, resulting in the formation of cracks at the root of the tooth, which can cause a total or partial fracture of the tooth. The occurrence of this type of damage can be caused by the combined effect of many material, technological, or operational factors.

Necessary action affecting the achievement of the required tooth bending strength is to carry out the appropriate design process. An indispensable condition for providing representative results of calculations of gears is obtaining appropriate values of strength parameters. Without these parameters, calculations would have large errors. This would result in an underestimation of the size of the tooth, which may result in catastrophic wear or overestimation, which increases material consumption, the cost of gear production, and the size of the transmission. It should be mentioned here that the issue of obtaining relatively small dimensions of gear transmissions is crucial, for example, for the mining industry, due to the limited size of mining excavations.

To determine the safety factor for tooth root stress, which is the result of strength calculations of gears, a parameter called the nominal stress number (bending) σ_{Flim} is necessary **[L. 1]**. The σ_{Flim} stress number is determined from the S-N (Wöhler) curve. It is best to determine this curve on the basis of gear tests. For example, devices called pulsators are used for this purpose. Pulsator cyclic tests are of a fatigue testing nature and rely on the fact that a tooth is loaded with pulses, with a specific frequency, until the tooth fracture occurs, and the number of cycles is determined based on this. In order to determine the full S-N curve, different levels of forces loading the teeth of the test gear are used.

The nominal stress number can be determined in a popular fatigue bending test on a simple specimen **[L. 2]**, but this method, although much cheaper, is recommended only for preliminary comparison of various engineering materials, because the geometry of the simple specimen does not correspond to the geometry of the tooth's root **[L. 1]**. In the area of the tooth root, maximum values of bending stress are observed due to the temporary working load at service. In addition, technological notches may occur in this zone, caused, for example, by the improper grinding of the tooth flank or structural ones resulting from the differentiation of the microstructure as a result of, e.g., surface hardening.

Currently, various design solutions of pulsators are found **[L. 3–5]**. The standard SAE J1619 **[L. 6]** gives details of the design of the test gear fixture, geometric parameters of the test gear, and the methodology for determining the S-N curve in fatigue tests.

At various research centres in the world, fatigue research is conducted in two ways. Some of the works are experimental, e.g. **[L. 7–9]**, while the rest use mathematical modelling tools **[L. 10–13]**.

Standard ISO 6336-5:2003(E) **[L. 14]** gives the σ_{Flim} values for the most common materials used for gears, e.g., for case-hardened steels. However, there is no data for a new generation of engineering materials. In the case of testing new materials, in order to know the exact value of σ_{Flim}, it is necessary to perform dedicated tooth bending fatigue tests. Sporadically appearing data in the literature, e.g. **[L. 15]**, most often refer to different materials; therefore, they can only be treated as approximations.

For a dozen or so years at ITeE – PIB in Radom, research has been carried out on the application of new, advanced materials for high-load gears. So far, they have focused on tribological gear studies **[L. 16–20]**, giving only partial information on the suitability of the material being tested. Since 2016, the Tribology Department of ITeE – PIB has been participating in performing a 5-year R&D application project (which aims to implement new engineering materials in the transmissions of mining conveyors to improve their functionality), and it was necessary to extend the research area with tooth bending fatigue tests required by the industry. Therefore, in the ITeE – PIB, a pulsator was developed and manufactured, with the symbol T-32 assigned. The methodology for testing tooth bending fatigue and the determination of the nominal stress number σ_{Flim} was developed. The device and research methodology meet the requirements of SAE J1619 **[L. 6]**.

The article presents a new device, research methodology, and results of verification tests of gears made of 18CrNiMo7-6 – classic engineering, case-hardened steel.

TEST RIG AND TEST GEAR

A photograph of the T-32 Pulsator, developed and manufactured at ITeE – PIB, is shown in **Fig. 2**.

Fig. 2. T-32 Pulsator
Rys. 2. Widok pulsatora T–32

The device is equipped with a hydraulic aggregate (on the left side), which sets the hydraulic oil pressure in the actuator operating part at the level of 280 bar. It includes the following: oil tank (about 250 l), oil pump, accumulator for stabilizing oil pressure, oil filter, oil cooler, as well as measuring and protection elements.

The main part of the pulsator consists of the following: hydraulic actuator, pressure sensor, tooth loading system, stator, head, and manipulator. The pulsator and hydraulic aggregate are mounted on anti-vibration elements. The test gear is enclosed in a transparent chamber.

The examined gear is mounted in a special fixture – **Fig. 3**. The tooth at the bottom is a supported tooth, and the tested tooth (broken) is at the top.

The elements of the control and measurement system are placed in the control cabinet (**Fig. 2** – on the right side of the pulsator). Setting the test parameters takes place from the touch panel. During the tests, the touch panel indicates the number of cycles and the minimum and maximum load value. To prevent shock loads, the minimum loading force is set to 10% of the maximum value.

The pulsator is equipped with appropriate electrical and mechanical protections and a monitoring system of work parameters, e.g., the pressure and temperature of hydraulic oil, the presence of three phases, the hydraulic oil level, the piston rod position of the hydraulic actuator, etc. The operating status is indicated by the signal lamps placed on the control cabinet.

T-32 Pulsator has the following technical specification:

– Maximum load on the tooth: 100 kN,

a)

b)

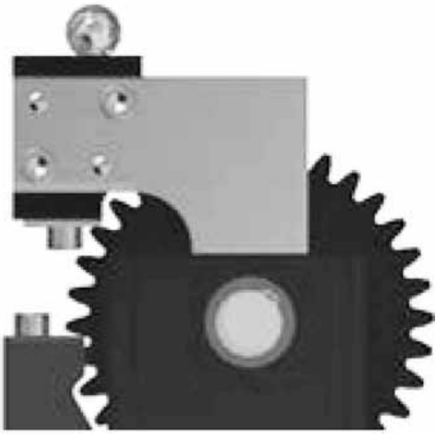

Fig. 3. Test gear fixture: a) photograph, b) model
Rys. 3. Uchwyt mocowania koła testowego: a) widok, b) model

– Cyclic load frequency: 30 Hz, and
– Maximum number of cycles after which the test will be terminated: 10 million.

During the tests, one of the teeth of the test gear made of the material under consideration is pressed and the number of loading cycles is counted until the tooth breaks. The applied load and the number of cycles are also measured.

A drawing of the test gear is shown in **Fig. 4**.

The test gear after the bending fatigue test is shown in **Fig. 5**.

The test gears were made of 18CrNiMo7-6 steel subjected to hardening and low tempering. The hardness of the surface of the gear tooth flanks was in the range of 60–62 HRC, and the hardness of the core 32–40 HRC.

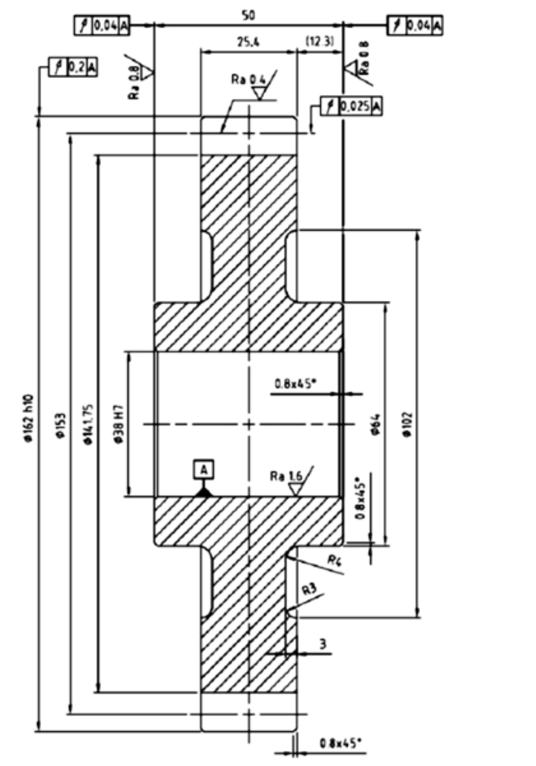

Fig. 4. Drawing of the test gear according to SAE J619 [L. 6]
Rys. 4. Postać konstrukcyjna testowego koła zębatego wg normy SAE J619 **[L. 6]**

Fig. 5. Test gear with a broken tooth
Rys. 5. Koło testowe ze złamanym zębem

TEST METHOD

In order to determine the σ_{Flim} stress number, it is necessary to perform a 4-stage test. These stages are as follows:

- The determination of the calibration curve: stress-load,
- The determination of the finite part of the S-N curve,
- The determination of the infinite part of the S-N curve using the Stair Case method, and
- The determination of the nominal stress number (bending) σ_{Flim}.

Determination of the calibration curve: stress-load

Because, in the S-N (Wöhler) plot, the vertical axis is the tensile stress at tooth root, and the load of the pulsator is given in force units, before the beginning of tests of a given material, a calibration curve in the coordinates of stress vs. load should be determined. The calibration curve is determined statically, using, for example, a tensile/compressive strength testing machine with a load up to 100 kN; the stresses under a given load are measured by a set of strain gauges stuck to the tooth root fillet of the pressed tooth (**Fig. 6**) and connected to a measuring amplifier.

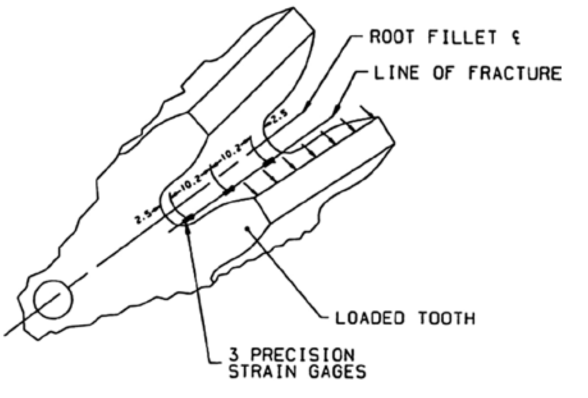

Fig. 6. Precision strain gages mounted in the tooth root fillet – according to SAE J619 [L. 6]
Rys. 6. Sposób naklejenia czujników tensometrycznych – wg SAE J619 **[L. 6]**

Figure 7 shows the measurement system used in the calibration.

The tensile strength S_u is read from the calibration curve. For the purpose of this procedure, it can be assumed that it is the tensile stress at the root of the tooth at the point of breaking the tooth or when the maximum load of the loading machine is reached – about 100 kN. This value is not the same as the tensile strength obtained in the classic tensile test.

Fig. 7. Measuring equipment
Rys. 7. Układ pomiarowy

Determination of the finite part of the S-N curve

The test procedure is based on SAE J619 **[L. 6]** and on the book **[L. 21]**. It assumes the following:
– Mounting the test gear in the fixture so that the tested tooth was not supported during calibration;
– Mounting the fixture with the test gear in the pulsator; and,
– Setting the load of the tooth corresponding to 0.85 S_u (load can be calculated from the equation of the calibration curve) and performing the run up to the fracture (failure) of the tooth together with determining the number of cycles leading to damage. (Note: if the number of cycles reaches 10 million and there is no failure, then the run should be rejected). The run should be repeated 4 times.

On the basis of the results from 4 runs x_i, the mean of the fatigue life should be calculated for the applied load from the following Formula (1) **[L. 21]**:

$$x_l = \frac{\sum_{i=1}^{n} \log x_i}{n} \tag{1}$$

where x_l – mean of the fatigue life (number of cycles) for the load exerted,
x_i – number of cycles to break a tooth in a given research run, and
n – number of run repetitions under a given load.

The next step is to set the load of the tooth to a value corresponding to 0.75 S_u. It is necessary to perform the run until the tooth breaks and note the number of cycles to break. One needs to repeat the run 6 times and calculate the mean of the fatigue life for the applied load from Formula (1) based on the results from the 6 runs.

The last step is to set the load of the tooth to a value corresponding to 0.65 S_u. It is necessary to perform the run until the tooth breaks and note the number of cycles to break. One needs to repeat the run 8 times and calculate the mean of the fatigue life for the applied load from Formula (1) based on the results from the 8 runs.

On the basis of the obtained averaged lives, a part of the S-N curve corresponding to finite fatigue life can be built. For this purpose, in the full logarithmic plot of stress vs. the number of cycles (S-N), one needs to mark the three determined means of the fatigue lives and lead the trend line to the intersection with the line corresponding to the cycle life, which, according to SAE J619 **[L. 6],** is 1 million cycles. This is shown schematically in **Fig. 8**.

At the point of intersection of the trend line with the line corresponding to the 1 million cycles, the $\sigma_{1\ mln}$ stress is determined, and, from the calibration curve, the corresponding $F_{1\ mln}$ load is calculated.

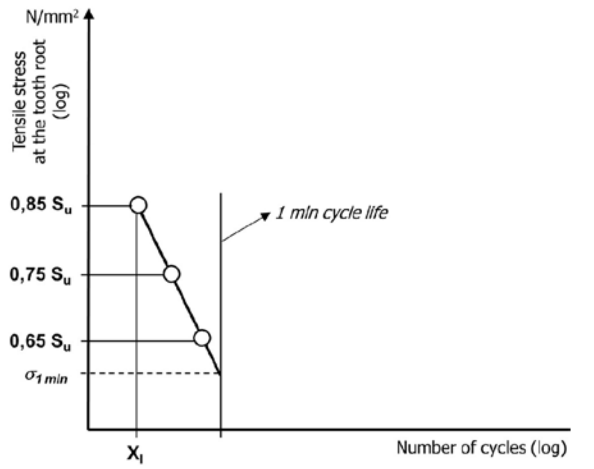

Fig. 8. Finite part of the S-N curve

Rys. 8. Krzywa S–N (wykres Wöhlera) – część odpowiadająca ograniczonej trwałości zmęczeniowej

Determination of the infinite part of the S-N curve using the Stair Case method

The procedure for determining the infinite part of the S-N curve is based on SAE J619 **[L. 6]** and on the book **[L. 21]**.

The test procedure is as follows:
– Load the tooth with force 1.2 * $F_{1\ mln}$. Carry out the run until the tooth breaks and note the number of cycles to fracture (failure). If the number of cycles <10 million, reduce the load to 1.15 * $F_{1\ mln}$. If the number of cycles ≥ 10 million, then the load should be increased to 1.25 * $F_{1\ mln}$. The load interval should not be greater than 5% of the $F_{1\ mln}$ value.
– Carry out runs on 15 to 25 teeth in the manner specified above; it is advisable to have an odd number or teeth tested. Sample data from the research, presented in the book **[L. 21]**, is shown in **Fig. 9**.
– Calculate the estimate of the mean and the scatter of the values (half-width of the confidence interval) of the stress number $\sigma_{F\infty,Pulsator,50\%}$ using the below given formulas **[L. 21]**. The estimation of the mean determines the infinite part of the S-N curve at the probability of 50% – **Fig. 10**.

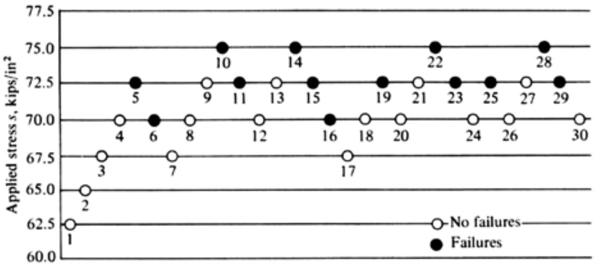

Fig. 9. Example data obtained using the Stair Case method [L. 21]; *No failures* – tooth survives 10 million cycles; Failures – tooth breakage occurs before 10 million cycles

Rys. 9. Przykładowe dane otrzymane metodą „schodkową" **[L. 21]**; *No failures – Brak złamania* – liczba cykli osiągnęła 10 mln bez złamania zęba; *Failures – Złamanie* zęba nastąpiło przy liczbie cykli mniejszej od 10 mln

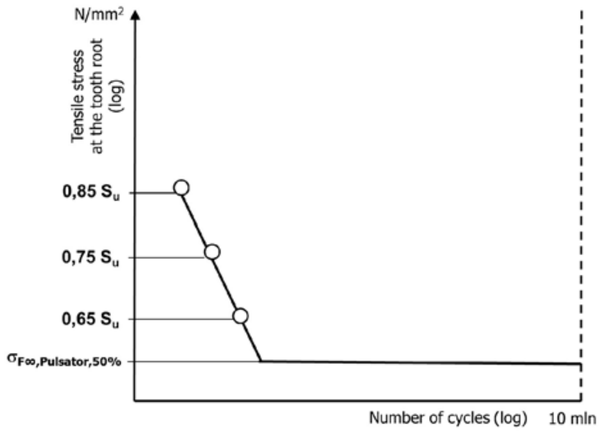

Fig. 10. The full S-N curve
Rys. 10. Pełna krzywa S–N (Wykres Wöhlera)

The calculation is made on these data (*No failure* or *Failure*), which is less. In the example shown in **Fig. 9**, there are fewer *Failures*.

In the first step, the identifier i has to be assigned to the given stress level S_i. The smallest stress has the identifier i = 0. Next, one needs to determine how many *Failures* were recorded for a given stress level (*Failures*, because there is less in the given example).

Determine the parameters A and B from Formulas (2) and (3):

$$A = \sum i n_i \tag{2}$$

$$B = \sum i^2 n_i \tag{3}$$

In the next step, the estimate of the mean should be determined from Formula (4):

$$\bar{x} = S_0 + d\left(\frac{A}{\sum n} \pm \frac{1}{2}\right) \tag{4}$$

where: "+1/2" is used when *No failure* is less frequent and "-1/2" is used when the *Failure* is less frequent,
d – stress interval corresponding to the load interval [N/mm²], and
S_0 – the lowest test stress [N/mm²].
The value $\bar{x}$ corresponds to the value $\sigma_{F\infty,\ Pulsator,\ 50\%}$.

In the next step, the standard deviation s from Formula (5) is determined:

$$s = 1,62d\left(\frac{B\sum n - A^2}{\left(\sum n\right)^2} + 0,029\right) \tag{5}$$

under condition that

$$\frac{B\sum n - A^2}{\left(\sum n\right)^2} > 0.3$$

If the above condition is not met, the standard deviation cannot be determined.

In the last step, the limits of the confidence interval should be determined, in which the true mean μ is "located" with the probability of 95%, from Formula (6):

$$\bar{x} - \frac{Gs}{\sqrt{\nu}} t_{\alpha;\nu} \le \mu \le \bar{x} + \frac{Gs}{\sqrt{\nu}} t_{\alpha;\nu} \tag{6}$$

where: G – parameter read out from **Fig. 11**,
ν = n–1 (the number of degrees of freedom; in this case n = Σn_i),
$t_{\alpha;\nu}$ – value from the t-Student distribution, and
α – significance level.

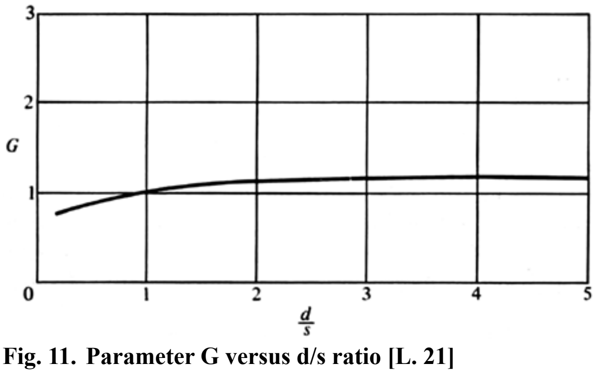

Fig. 11. Parameter G versus d/s ratio [L. 21]
Rys. 11. Wartości parametru G w funkcji stosunku wielkości d/s **[L. 21]**

Determination of the nominal stress number (bending) σ_{Flim}

On the base of **[L. 15]**, the nominal stress number (bending) σ_{Flim} is calculated from Equation (7):

$$\sigma_{F\lim} = \frac{\sigma_{F\infty,1\%}}{Y_{ST} * Y_{NT} * Y_{\delta relT} * Y_{RrelT} * Y_X} \tag{7}$$

where σ_{Flim} – nominal stress number (bending) accepted for design calculations of gears,
$\sigma_{F\infty,1\%}$ – stress number referred to the probability of 1%,
Y_{ST} – stress correction factor,
Y_{NT} – life factor,
$Y_{\delta relT}$ – relative notch sensitivity factor,
Y_{RrelT} – relative surface factor, and
Y_X – size factor.

According to **[L. 15]**:

$$\sigma_{F\infty,1\%} = \sigma_{F\infty,Pulsator,50\%} * 0.9 * 0.92 \tag{8}$$

where: 0.9 – correction factor taking into account the uncertainty of tests in the pulsator, and
0.92 – correction factor including the conversion of stress number 50% to 1%.

Thus:

$$\sigma_{F\lim} = \frac{\sigma_{F\infty,Pulsator,50\%} * 0.9 * 0.92}{Y_{ST} * Y_{NT} * Y_{\delta relT} * Y_{RrelT} * Y_X} \quad (9)$$

The method of determining the particular factors based on the ISO 6336-3:2006(E) standard **[L. 1]** is given below.

Stress correction factor Y_{ST}.

Because σ_{Flim} was determined using the standard reference test gears, $Y_{ST} = 2$.

Life factor Y_{NT}.

Its value is very close to the value 1 with the assumptions regarding the limit of cycles in the pulsator: 10 million cycles.

Relative notch sensitivity factor $Y_{\delta relT}$.

For the notch parameter $q_s = 2.5$, $Y_{\delta relT} = 1$.

Relative surface factor Y_{RrelT}.

For case-hardened steel and expected roughness of the tooth flanks in a mining conveyor $R_z = 1.1$ μm, it has been calculated that

$$Y_{RrelT} = 1.674 - 0.529\left(R_z + 1\right)^{0,1} = 1.104 .$$

Size factor Y_X.

For case-hardened steel and expected normal module of the gears in a mining conveyor $m_n = 10$ mm, it has been calculated that

$$Y_X = 1.05 - 0,01\ m_n = 0.950.$$

RESULTS

Calibration curve

The calibration curve is shown in **Fig. 12**.

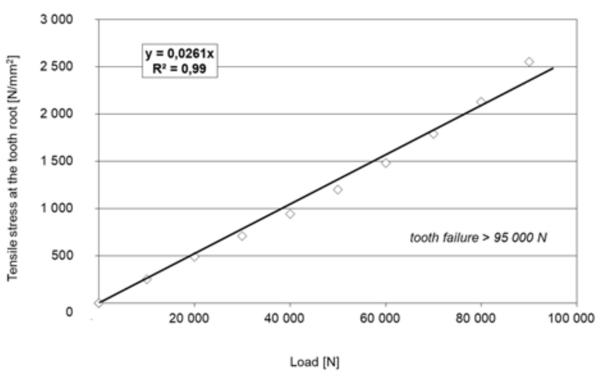

Fig. 12. Calibration curve for the test gears made of case-hardened steel 18CrNiMo7-6

Rys. 12. Krzywa kalibracyjna dla kół testowych wykonanych z nawęglanej i hartowanej stali 17HNM

It can be observed that the coefficient of determination $R^2 = 0.99$. It indicates a very good fit of the trend line to the measurement points.

From **Fig. 12**, the tensile strength $S_u = 95\ 000 * 0.0261 = 2\ 480$ N/mm² was determined for the maximum achieved static force of 95 000 N. As mentioned before, this value is not the same as the tensile strength obtained in the classic tensile test.

Testing at the stress level relating to the finite part of the S-N curve

Table 1 presents test results for stresses corresponding to the finite part of the S-N curve. The means x_l were determined from Formula (1).

Table 1. Results of tests and calculations at the stress levels relating to the finite part of the S-N curve

Tabela 1. Wyniki badań i obliczeń przy naprężeniach odpowiadających ograniczonej trwałości zmęczeniowej

Root stress [%S_u]	Root stress level [N/mm²]	Number of cycles until fracture, x_l	Mean of the number of cycles, x_l
85	2108	985	1323
85		1426	
85		1475	
85		1478	
75	1860	2568	3296
75		3627	
75		3173	
75		4062	
75		2948	
75		3622	
65	1612	6982	9087
65		6832	
65		6314	
65		6851	
65		21745	
65		10258	
65		11309	
65		8934	

Figure 13 presents the results in a graphical form in a full-log plot.

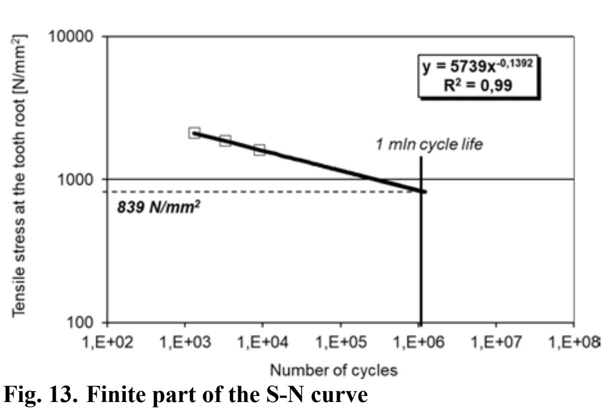

Fig. 13. Finite part of the S-N curve

Rys. 13. Krzywa S–N – część odpowiadająca ograniczonej trwałości zmęczeniowej

Figure 13 shows that the coefficient of determination $R^2 = 0.99$. It indicates a very good fit of the trend line to the measurement points.

Extrapolation of the line to intersection with the cycle life of 1 million allowed the determination of the stress $\sigma_{1\ mln}$, and from the calibration curve the corresponding load $F_{1\ mln}$. They are respectively: 839 N/mm² and 32 134 N.

Testing at the stress level relating to the infinite part of the S-N curve

Figure 14 presents test results at stresses corresponding to the infinite part of the S-N curve.

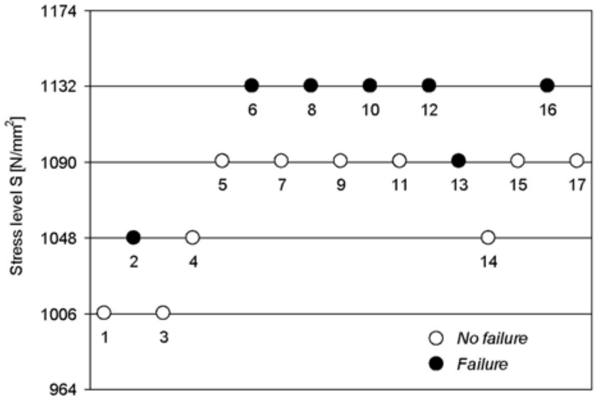

Fig. 14. Test results using the Stair Case method
Rys. 14. Wyniki badań metodą „schodkową"

Figure 14 shows that there were fewer *Failures* in the result set. There were 7 out of 17 results.

On the basis of **Fig. 14,** the data in **Tab. 2** has been compiled.

Based on the data from **Tab. 2**, from Formulas (2), (3), and (4), parameters A, B, and the estimate of the mean $\bar{x}$ were determined:

A = 18
B = 50
$\bar{x} = 1093$ N/mm².

Table 2. Stress identifiers i, stress levels S_i and number of *Failures* n_i for a given stress
Tabela 2. Identyfikatory naprężeń i, poziomy naprężeń S_i oraz liczba n_i uszkodzeń dla danego naprężenia

i	S_i [N/mm²]	n_i
0	1006	0
1	1048	1
2	1090	1
3	1132	5

Because the parameter:

$$\frac{B\sum n - A^2}{\left(\sum n\right)^2} = 0.531, \text{ which is higher than } 0.3,$$

the standard deviation has been calculated – Formula (5):

$$s = 38 \text{ N/mm}^2.$$

From **Fig. 11**, for the ratio of d/s = 1.1, it has been read out that the parameter G = 1.05.

For probability of 95% and number of degrees of freedom $\nu = 7 - 1 = 6$, half-widths of the confidence interval have been determined from Formula (6):

$$\frac{Gs}{\sqrt{\nu}} t_{\alpha;\nu} = 40 \text{ N/mm}^2.$$

Value of the nominal stress number (bending) σ_{Flim}

In the previous step, the estimate of the mean has been calculated: $\bar{x} = \sigma_{F\infty,Pulsator,50\%} = 1\,093$ N/mm².

The following values were adopted for further calculations:

$$Y_{ST} = 2;\ Y_{NT} = 1;\ Y_{\delta relT} = 1;\ Y_{RrelT} = 1.104;\ Y_X = 0.950$$

From Formula (9) the nominal stress number (bending) σ_{Flim} could be determined as follows:

$$\sigma_{Flim} = 431 \text{ N/mm}^2.$$

The corrected confidence interval half-widths could be calculated in the analogous manner, which ultimately allows the following conclusion:

$$\sigma_{Flim} = 431 \pm 16 \text{ N/mm}^2.$$

SUMMARY

The article presents a new test rig – T-32 Pulsator for breaking a gear tooth, research methodology, and the results of verification tests of test gears made of 18CrNiMo7-6 case-hardened steel. The verification was made on the basis of the determined nominal stress number (bending) σ_{Flim}.

Standard ISO 6336-5:2003(E) **[L. 14]** gives the values of σ_{Flim} similarly treated steel in the range from approx. 312 to 525 N/mm². The highest values are assumed if there is certainty about compliance with the requirements of elemental composition, purity, physical properties, grain size, and phase composition of the steel from which the gears will be made, if an ultrasound test is provided to examine the continuity of the gear material structure, if the surface hardness of each gear will be tested, showing compliance with the recipient's requirements, if the correct core hardness will be demonstrated, if a magnetic test is provided to examine the continuity and quality of the material structure of the gear, and if the compatibility of the phase composition of the tooth surface and the core with the requirements is examined.

The value of σ_{Flim} obtained in this work is 431 N/mm², and therefore lies within the above range.

The manufacturer of the gears used for testing, for the bending strength calculations of gears made of

18CrNiMo7-6 steel, takes the value of σ_{Flim} = 430 N/mm^2. It is a value almost identical to the one determined in the research.

Thus, the determined value of the nominal stress number (bending) σ_{Flim} is correct. This indicates the proper operation of the T-32 Pulsator and the correctness of the developed research methodology.

It should be emphasized that the working conditions of the pulsator do not reflect the conditions of actual work of the gears, where there is one-pair meshing and double-meshing engagement. The most reliable test results are obtained on back-to-back test rigs. However, such studies require several test rigs working simultaneously and many months of research. Research on a single pulsator may, however, be completed in less than two months. What is more, these types of tests, in addition to typical gear tests, are approved by ISO 6336-3:2006(E), Method B **[L. 1]**.

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