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SOME PROBLEMS OF FATIGUE LIFE OF MACHINE ELEMENTS AND STRUCTURES UNDER SERVICE LOADINGS

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The paper contains a review of problems connected with fatigue life of machine elements and mechanical structures, presented during the 5th National Conference "Fundamentals of construction, operation and testing of heavy-duty machines, including machines used in building engineering", Zakopane 1992. On the basis of information from the latest world literature directions of the further investigations on heavy-duty machines life have been proposed.

1. Introduction

Fatigue damage cumulation in materials belongs to the most important reasons of failure of machine elements and mechanical structures while their operating. At present it is possible to observe similar approach to solving problems of fatigue life in many fields (for example in construction of aircrafts, rockets, road and rail vehicles, ships, off-shore structures, turbines and energetic generators, buildings, chemical apparatus, different types of machines and their elements) (cf [11]). Service loadings, influence of environment in which the machine works as well as mechanical properties of materials and their shapes are the most important factors determining fatigue life.

This paper shows some problems connected with the fatigue life of machine elements and structures. The discussed problems have been selected from seven papers presented on the 5th National Conference "Fundamentals of construction, operation and testing of heavy-duty machines, including machines used in building engineering" (Zakopane, Poland, 1992), (cf [21]).

2. Uniaxial fatigue

In the paper presented by Malinowski [22], methods of fatigue life calculations for elements and fittings of excavators and loaders are discussed. These calculations are made during the designing process and preliminary tests of prototypes. In the calculations

- a rate of damage probability, a number of working cycles up to a fracture and a type of welding notch of the element are assumed,
- standard fatigue characteristics of the material $(S_a - N)$ are used,
- parameter A, equal to the half of a range of extreme stresses in the designed element and determined under static loadings, is applied,
- probability density function of service loading amplitudes is applied.

About 100 stress-time histories for single-bucket excavators and loaders provided with 14 types of fittings and used in typical service conditions were applied to determination of the generalized probability distribution of stress amplitudes. Cycles were counted employing the range pair method. The generalized probability density function of stress amplitudes $f(S_a)$ was assumed as a linear composition of two distributions: exponential distribution $f_1(S_a)$ and distribution of extreme values $f_2(S_a)$

$$\begin{aligned} f(S_a) = \frac{L-\nu}{L} f_1(S_a) + \frac{\nu}{L} f_2(S_a) = \frac{L-\nu}{L}\frac{k_1}{A}\exp(-k_1\frac{S_a}{A}) + \\ +\frac{\nu}{L}\frac{k_2}{A}\exp\Big[-k_2\frac{S_a}{A} + k_2k_3 - \exp\Big(-k_2\frac{S_a}{A} + k_2k_3\Big)\Big] \end{aligned} \tag{2.1}$$

where

ν – a number of cycles of machine work,

L – a total number of amplitudes in all working cycles ν,

k_1, k_2, k_3 – dimensionless coefficients.

It results from the measurements that the mean values of coefficients for excavators are: $k_1 = 12.9$, $k_2 = 7.96$, $k_3 = 0.74$, $L/\nu = 98$. A linear hypothesis of damage cumulation is applied to; fatigue calculations damages in the amplitude interval $S_a = (0.5 \div 1)Z$ are cumulated for the exponent $(m+2)$ of the $(S_a - N)$ curve and in the interval $S_a = Z \div 2.5A$ for m, where Z is a fatigue limit.

In future the presented method of calculations should be extended on a low-cycle fatigue and elements of fracture mechanics (cf [24,26]). The paper by Malinowski [22] gives a solid background for planning full-scale fatigue tests of complete elements on the special stands. In Poland a level of such tests is not too high yet comparing with foreign laboratories.

In further measurements of service loadings in excavators and loaders, cross-correlations between loadings should be also included. These cross-correlations cannot be neglected when multiaxial fatigue is considered (cf [7]).

Gear transmissions mechanisms, in turn, belonging to body-works of single-bucket tracklaying excavators M250H often fracture before the proper time. Thus, it was necessary to perform suitable tests and calculate their fatigue lifes. Taking these gears into account Krukowski [14] presented a method of fatigue life calculation and a special computer program, called WTRD 1. The method is based on the characteristics of service loadings measured under real working conditions and, on the other hand, is in conformance with international standards ISO/DIS 6336 and ANSI/AGMA 2003-A86. Histories of service loading were measured for 18 possible work variants of the considered machine and for at least 5 working cycles of turn mechanism in each variant. The obtained results of schematization of loading histories were presented as an experimental histogram of amplitudes in seven classes. Calculations consisted in determination of fatigue life for carburized and hardened gear wheels made of 17HGM steel. In the computer program a standard method of calculations for gear wheels against fatigue bending and pitting was used. Fatigue damages were cumulated according to the linear Haibach hypothesis. Fatigue life of particular gear wheels was expressed in numbers of cycles of changes in loading of teeth for the assumed life-time of the excavator (8000 hours). It was assumed that time of turn of the bodywork is about 70% of the total worktime of the excavator; since one cycle lasts usually 30 s, a number of working cycles is 672×10^3.

In the considered paper it was found that for factors of safety, which were usually assumed for bending (1.5) and pitting (1.2), reliable work of the gear transmission was possible only for static loadings and it was unreliable under fatigue loadings. This result of calculations agrees with the independent full-scale fatigue tests of gears and data concerning excavators. It means that the proposed method of calculations is correct and it can be applied for other gear transmission systems with similar loadings.

Sobczykiewicz [26] presented a method of fatigue designed welded elements of single-bucket hydraulic excavators, including problems of fatigue life, manufacturing processes and operating conditions. Three stages of designing process are distinguished here. The first stage includes preliminary designing and calculations concerning some selected points or sections in which a fatigue fracture can occur. Two groups of data are necessary for these calculations: data concerning strength of the welded joint, included in fatigue characteristics $(S_a - N)$ and data concerning service loadings expressed with the probability density function of amplitudes $f(S_a)$ of BETA type

$$f(S_a) = \frac{1}{B(p,q)} S_a^{p-1}(1 - S_a)^{q-1} \tag{2.2}$$

where

$B(p,q)$ – function BETA dependent on p and q,

p, q – parameters of probability density function.

Parameters p and q were derived on the basis of many (above 60) one-hour measurements of loading histories for several excavators operated under various working conditions. Cycles were counted with the rain flow method. At this stage of calculations a fatigue life is determined for a high number of cycles. Here a reliability assessment is also established on the assumption that fatigue life of the welded joint has Weibull probability distribution function.

The second stage begins when a prototype of the machine is ready and calculations of life time are more precise. Thus, many data are required at this stage. They are: low-cycle fatigue characteristics of the material, i.e. a cyclic stress-strain curve $(S_a - \varepsilon_a)$ and strain-cycling fatigue characteristic $(\varepsilon_a - N)$ as well as the registered histories of nominal service stresses which may be a basis of determination of local histories of strains and stresses in the notch with the Neuber or the Glinka-Wolski rules. At the first and second stages damages are cumulated according to the Palmgren-Miner hypothesis.

At the third stage elements of fracture mechanics are applied, namely fracture toughness and a curve of fatigue crack growth rate $(da/dN - \Delta K)$, expressed by the Paris' formula. History of service loadings is required as well, like at the second stage.

Computer programs were worked out for each stage. An information on loadings and strength is more and more precise at each following stage, so the prediction of fatigue life becomes more accurate, too. At the second and third stages the results of fatigue life calculations are compared with the results of tests for prototypes operating under real working conditions.

From the actual papers published in international journals it appears that a real progress can be observed in simulation of fatigue life of elements and structures. From these papers it the appears that the range of calculations proposed by Sobczykiewicz, is worth extending especially at the first stage of designing.

Rusiński et al. [24] discussed modelling of the fatigue crack propagation in welded joints of heavy-duty machines. A software package called RYSA was worked out employing the finite element method (FEM). This package can be used for determination of the basic parameters of fracture mechanics, i.e. stress intensity factor K_I, J_I – integral according to three methods, crack opening displacement and crack length, and intensity of stresses σ_i near the crack tip, respectively, at each step of calculations. In the computer program elastic-plastic problems are solved according to the plastic strain theory and the theory of plastic flow and using the method of succeeding increments of loading. Transition into plastic state is defined according to the Huber-Mises-Hencky hypothesis. The program uses special isoparametric 2-D and 3-D finite elements. The presented example of calculations shows some possibilities of RYSA.

It seems to be obvious that further works on numerical simulation of cracks in elements of heavy-duty machines can tend, among others, to

- graphical presentation of path of cracks under the influence of service loadings which are random and multiaxial,
- determination of critical dimensions of defects under the assumed loadings,
- determination of permissible loadings after identification of the formed cracks and defects during periodical inspections.

The fatigue crack growth rate of the native material and the welded joints under different loadings are very important for determination of the fatigue life of machine elements. The overloads occurring during service, can, depending on their size and frequency, increase or reduce the fatigue life of the considered element.

In the paper by Borowiecki et al. [9], results obtained from experiments and calculations for the fatigue crack propagation in specimens made of 18G2A steel in the presence of overloads are presented. Compact (CT type) and flat (with central and edge notches and transverse butt joints) specimens were tested. Microscopic and macroscopic observations of the welded joints and the surface of fatigue cracks were carried out. A method of measurements of the crack length in a specimen with an edge defect was worked out. The method is based on measurements of crack opening displacement at the edge of the specimen. The method was applied to the tests of specimens under loading with constant amplitude and constant amplitude with overloads.

CT specimens were tested assuming three values of overload coefficients: 1.5, 1.75 and 2 and three spacings between overloads: 2000, 5000 and 20000 cycles, respectively. Flat specimens were tested under one combination of the overload coefficient (1.75) and the spacing (1000 cycles). All the experiments were realized for a constant ratio of stress (0.3). For numerical analysis of experimental data a calculation model of fatigue crack growth, based on the Forman equation, was used.

After the tests it was found that both overloads and the existing transverse weld cause life increased in relation to the tested specimens without overloads and non-welded. As for specimens with edge crack, their life increased almost two times owing to overloads and about 10% when the welded joint was made. Application of retardation models allows to describe the experimental fatigue crack growth rate on the basis of the assumed parameters. However, if large overloads occur (about 1.75 and more) the parameters of the model should be changed.

It was also found that the proposed method, consisting in determination of the fatigue crack length on the basis of measurements of crack opening displacement in a flat specimen with one-sided notch, reflects a size of the propagating crack very well. The paper by Borowiecki et al. [9] gives a new information about overloads and their role in the fatigue processses for 18G2A steel and welded joints. The information can be useful while explaining how the order of loadings influences a fatigue life. In the past some research workers tried to understand and describe this influence. Unfortunately, the problem has not been solved so far. The solution of this problem seems to be very important for formulation of the correct models of fatigue life under non-stationary random service loadings of machine elements and structures.

At present methods of non-destructive inspections for evaluating a fatigue life of materials and steel structures are developed. Besztak et al. [3] tried to ascertain if ultrasounds may be used for this purpose. They tested flat specimens made of 18G2A and St3SM steels with fatigue microdamages. The microdamages were characterized by different degrees of evolution controlled by a relative number of cycles up to fracture, i.e. a degree of fatigue damage cumulation. According to the Palmgren-Miner hypothesis it was equal to 0.23, 0.68, 0.91, and 1, respectively. Two parameters were considered, namely a velocity and a damping of ultrasonic wave (longitudinal ultrasonic waves passed through specimens along length 80 mm in the region of maximum fatigue damages). Level of the given stresses was S_a = 145 MPa with the stress ratio $S_{min}/S_{max} = 1/30$ and was corresponding to the fatigue life in the range of a high number of cycles (1.95×10^6 cycles for 18G2A steel and 2.2×10^6 cycles for St3SM steel). The investigations included measurements of the two mentioned parameters of ultrasonic waves in specimens with generated fatigue damage and in standard specimens without any damage. The velocity of the ultrasonic wave was determined indirectly, from measuremens of time with the error $\pm(0.1 \div 0.2)$ns and converters of frequency 4 MHz were used. The damping factor was measured with the error ± 0.3 db/cm for the frequency range 3-15 MHz with broad-bend heads and a system for spectral analysis of ultrasonic impulses.

From the measurements it results that in 18G2A steel a visible nonlinear decrease of ultrasonic wave velocity can be observed while increasing degradation of the material structure caused by fatigue loading. As for St3SM steel, such relation was not found because the obtained differences in the time of transition of the ultrasonic wave lied within the limit of error for the applied measuring method. Inversely, the damping factor is sensitive to changes in damage of this kind of steel and its variability versus a number of cycles is more visible for higher frequencies in the range $8 \div 12$ MHz. It seems to be interesting that this acoustic parameter is not very useful for evaluation of a fatigue degree of 18G2A steel.

The initiated investigations should be continued because owing to them it would be possible to work out a cheap method of prediction of the residual lifetime of the operated machines. This problem is very important from the practical point of view.

3. Multiaxial fatigue

Investigations on multiaxial fatigue have been carried out for about 100 years. Many mathematical models of limit states of material strength have been worked out so far. At present we know more than thirty fatigue criteria concerning multiaxial cyclic loadings. They are discussed, among others, in [8,12,13, 17,23].

In Poland investigations on multiaxial random fatigue were started 15 years ago and at present they are still at their preliminary stage [17,18]. Their development is rather difficult because:

- it is necessary to formulate correct mathematical models of fatigue criteria,
- new laboratory stands should be built,
- new research methods are necessary, especially algorithms for fatigue life determination.

Some machine elements under multiaxial service loadings were already tested in foreign laboratories.

At present two main directions can be distinguished in investigations on multiaxial random fatigue of machine elements:

- investigations of machine elements and structures under simulated or real service loadings,
- basic investigations of materials under a complex random stress state.

In the paper by Będkowski and Macha [7] the hitherto progress in investigations of multiaxial random fatigue has been presented. The authors discussed mathematical models, apparatus used in laboratories and preliminary results of tests verifying theory as well as they presented some new problems which should be solved in future.

3.1. Investigations of machine elements under service loadings

This direction of investigations tends to determination of the fatigue life of real machine elements or objects tested on laboratory stands, allowing to simulate service loadings. Sometimes these investigations of the real objects are carried out under extreme service loadings on a special proof ground. Such tests are ususally realized by big industrial laboratories and they are of a commercial character. The obtained results are seldom published.

Firms MTS and Schenck install hydraulic stands for multiaxial fatigue tests of vehicles, planes and rockets as well as their elements. They allow one to generate various service loadings with a certain number of servohydraulic actuators controlled by independent signals (cf [10]). For tests of cars and their elements, systems of 4,6,8,12 and 16 channels, respectively, are used. In a Spanish aeronautic laboratory CASA (cf [10]) we can find another example of multiaxial fatigue tests: here a horizontal stabilizer for the new Airbus A320 was tested. In Poland multiaxial fatigue tests are realized by car, aircraft and railway industries but these tests are not too much developed.

3.2. Tests of materials under complex random stress state

These tests are carried out in order to understand and mathematically describe the fatigue fracture in materials. First of all suitable criteria of multiaxial random fatigue should be formulated. Owing to these criteria a complex random stress state in the critical point of the machine element can be reduced to an uniaxial state and next, on the basis of standard characteristics of cyclic fatigue of the material, the fatigue life is calculated.

The first attempts on formulating the criteria of multiaxial random fatigue are presented in [17,18,19]. They consist in generalization of the known criteria for cyclic fatigue. In [8,17,18,19] five criteria for the random triaxial stress state are proposed and some limitations which make it impossible to use other criteria in the considered conditions are presented. From among the five criteria three are based on stresses, one on an elastic strain energy and the last one – on strains. The stress criteria were expressed as one generalized criterion of maximum shear and normal stresses in the fracture plane [8]

$$\max_t\{B\tau_{\eta s}(t) + K\sigma_\eta(t)\} = F \tag{3.1}$$

where

$\tau_{\eta s}(t)$, $\sigma_{\eta s}(t)$	–	shear stress in $\bar{s}$ direction and normal stress in the fracture plane with the normal $\bar{\eta}$, respectively,
B, K, F	–	constants for selecting the particular form of the criterion.

The $\bar{s}$ direction agrees with the mean direction of maximum shear stress $\tau_{\eta s max}(t)$.

In [13,16] the similar generalized strain criterion is formulated

$$\max_t\{b\varepsilon_{\eta s}(t) + k\varepsilon_\eta(t)\} = q \tag{3.2}$$

where

$\varepsilon_{\eta s}(t)$, $\varepsilon_\eta(t)$	–	shear and normal strains in the fracture plane, respectively,
b, q, k	–	constants for selecting a particular form of the criterion.

The fatigue fracture plane position is described in terms of mean values of direction cosines $\hat{l}_n$, $\hat{m}_n$, $\hat{n}_n$ $(n = 1,2,3)$ of the axis of principal strains or stresses. For determination of the expected fatigue fracture plane position three methods were proposed

- weight function method (cf [17,18,19,20]),
- method of equivalent stress variance (cf [4,5,6,20]),
- method of fatigue damage cumulation (cf [20]).

A general algorithm for the fatigue life determination under multiaxial random loadings was formulated on the basis of criteria (2.1) and (2.2). It includes the following operations

- measuring, calculating according to MES or generating $\varepsilon_{ij}(t)$ or $\sigma_{ij}(t)$ $(i,j = x,y,z)$,
- determination of the expected fatigue fracture plane direction, i.e. $\hat{l}_n$, $\hat{m}_n$, $\hat{n}_n$, $(n = 1,2,3)$,
- calculation of histories of $\varepsilon_{red}(t)$ or $\sigma_{red}(t)$,
- cycle counting according to the rain flow method, the range pair method, the hysteresis loop method or the full cycle method,
- transformation of cycle amplitudes in relation to mean values,
- calculations of the damage degree and the fatigue life according to the assumed fatigue hypothesis (for example the Palmgren-Miner or the Haibach hypothesis).

In Poland stands for fatigue tests of round specimens under random torsion with bending (electromagnetic actuators were used here) and for fatigue tests flat cruciform specimens under biaxial independent loadings (servohydraulic actuators applied) were built for verification of the algorithm discussed above [20].

These two stands were equipped with generators of stochastic signals simulating service loadings and microcomputer cumulators of fatigue damages.

From the preliminary fatigue tests of cylindrical specimens made of 10HNAP steel under random torsion with bending (four combinations) as well as cruciform specimens of 10HA steel under biaxial tension-compression it results that the best agreement between fatigue life calculations and experimental data is obtained for the criterion of maximum normal stress in the fracture plane with the method of variance, the method of range pair and the Palmgren-Miner hypothesis [1].

From among foreign papers on multiaxial random fatigue those published in some last years should be distinguished [2,15,25, 27]. Leger [15] built a laboratory stand with three servo-hydraulic actuators for tests of round specimens; on the stand it was possible to control a torsional moment and bending moments independently in two perpendicular axes. However, the author has not given the obtained results of tests and a suitable theory. Sanetra and Zenner [25] tested round specimens under a combination of torsion and tension-compression and now they are analysing the results. Sonsino and Pfohl [27] applied a simplified procedure of fatigue calculations for redesigning a welded joint of the flange with the shaft in stirrers for fertilizers. The new procedure included maximum amplitudes from random histories of normal and shear stresses registered under operating conditions. It was assumed that the maximum amplitude of reduced stress was equal to the maximum amplitude of normal stress multiplied 1.14 times. Bennantine and Socie [2] suggest that fatigue damages should be cumulated in the fracture plane of the material (damages are the largest in this plane) with the Smith-Watson-Topper criterion in the maximum points of time-history of normal stress. This approach is the most similar to that applied in Poland.

From review of papers it appears that investigations of the multiaxial random fatigue in materials carried out by Polish research workers are the most advanced. Further investigations should tend to establishment of usability of the range of application of the proposed algorithm for fatigue life determination.

4. Conclusions

On the basis of the review of the discussed Polish papers and actual tendencies in investigations the following directions of works on a service life of heavy-duty machines can be proposed.

1. Computer aided stands for tests of machines and their fittings under simulated service loadings should be built.

2. Measurements of representative time histories of service loadings and their standarization for some selected types of machines are necessary.

3. It is necessary to continue tests of known and new materials giving development of fatigue and fracture data base.

4. Algorithms for fatigue life evaluation, including multiaxial random stationary and non-stationary loadings, must be improved.

5. It seems to be necessary to search new methods of non-destructive inspections allowing one to determine the remaining life-time of the operated machines. The well-known methods should be improved.

6. The computer aided assessement methods and control of localized fatigue damages should be developed.

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Wybrane problemy trwałości zmęczeniowej elementów maszyn i konstrukcji przy obciążeniach eksploatacyjnych

Streszczenie

Dokonano przeglądu zagadnień trwałości zmęczeniowej elementów maszyn i konstrukcji mechanicznych przedstawionych w siedmiu referatach na V Konferencji "Rozwój podstaw budowy, eksploatacji i badań maszyn roboczych ciężkich, w tym budowlanych", Zakopane, styczeń 1992. Na podstawie analizowanych prac oraz informacji zawartych w najnowszej literaturze światowej zaproponowano kierunki dalszych badań nad trwałością eksploatacyjną maszyn roboczych ciężkich.

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