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CALCULATION OF THE STRUCTURAL ELEMENTS OF THE BUCKET WHEEL EXCAVATOR WORKING WHEEL TRANSMISSION

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Abstract. *Bearing structure of the bucket wheel excavator is a complex mechanical system with prominent elastic properties of certain parts. The excavator boom, because of its steel wire-rope suspension and an unfavorable position of the working wheel, which is a large concentrated mass in constant contact with the excavated face, is affected by the complex dynamic loads, originating from the resistance to excavation and the movement of the total excavator mass. Designing the gear drive transmission of the bucket wheel excavator working wheel is a very complex designing task, that requires the implementation of modern design methods for its completion. High precision in calculating the structural elements of the working wheel transmission can be achieved through the simulation of the exploitation conditions. The paper gives an approach to the calculation of the elements of working wheel drive transmission based on the standardized spectrums of loads.*

Key words: *bucket wheel excavator, working wheel transmission calculations, load spectrum*

INTRODUCTION

The most significant factor in terms of safety and reliability of a mechanical system (such as the working wheel drive transmission) is the bearing capacity of its structural elements. Bearing capacity can be considered the measure of the transmission parts quality. Calculation of bearing capacity is done by comparison of working and critical load status in the transmission elements, so the quality evaluation is reduced to the evaluation of precision of working and critical loads values.

Precision of calculation of working loads depends on the appropriateness of applied methods and the ways of determination of the loads. The appropriateness is determined by its possibility to identify the status, time and value of biggest loads. The quality of the

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applied methods is tested experimentally. The value of the working load depends on the load, so therefore it is necessary to know the value, course of change, frequency as well as the probability of occurrence of the highest loads occurring during the exploitation in the observed part or element of the construction. The present test of resistance of machine parts to destruction is conducted by determination of nominal load, that is nominal working stress which corresponds the most frequent stress during the work. Exploitation conditions, that is occurrence of loads higher than nominal is taken into account in addition to the working condition factors which is chosen by approximation, from the appropriate tables. It is easy to conclude that such way of determining the working load is approximate, which brings about the inadequate dimensions of the structure elements.

High level of precision when dimensioning and testing to resistance to destruction can be attained through measuring the exploitation loads and identification of the load spectrum of vital transmission elements. Load spectrums are obtained on the basis of exploitation measurements of mechanical system in operation process for certain conditions, so each load spectrum has its occurrence probability. Choice of the valid spectrum is solved through introduction of several load spectrums representing certain working conditions, which facilitates a sufficiently precise evaluation for all the conditions within their scope.

2. BUCKET WHEEL EXCAVATOR WORKING WHEEL TRANSMISSION

Bucket wheel excavator is one of most important machines of the BTO system, and it is on the basis of its characteristics that the remaining components of the system are sized. Efficiency of the whole system depends mostly on the operation of the bucket wheel excavator. Shape of the bucket wheel excavator construction and its dimensions depend on the required capacity, way of loading the material and specific conditions of the pit, such as: stability of the terrain, hardness of material and permissible load of surface which supports the excavator.

Nowdays, there is a series of various bucket wheel excavators constructions, that differ only by the working wheel diameter, number and shape of the buckets on the wheel, position of working wheel drive transmissions in relation to the boom and the working wheel etc.

Bucket wheel excavators are produced mostly as unique products according to the conditions and characteristics of the operating environment where they are supposed to excavate coal or ore (Fig.1).

Working wheel and the driving system (electromotor-power transmission) is huge concentrated mass in a very unfavorable position (tip of the boom) which is at constant contact with the excavated material and so is exposed to dynamically complex loads.

Driving system (transmission) has a significant influence on the excavator construction, because, it is in close contact with the working wheel that is the boom and the entire excavator construction (Fig. 2).

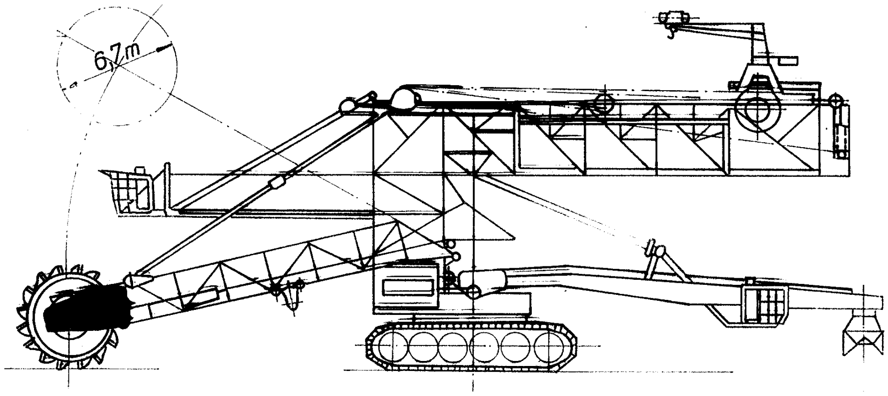


Fig. 1. Bucket wheel excavator

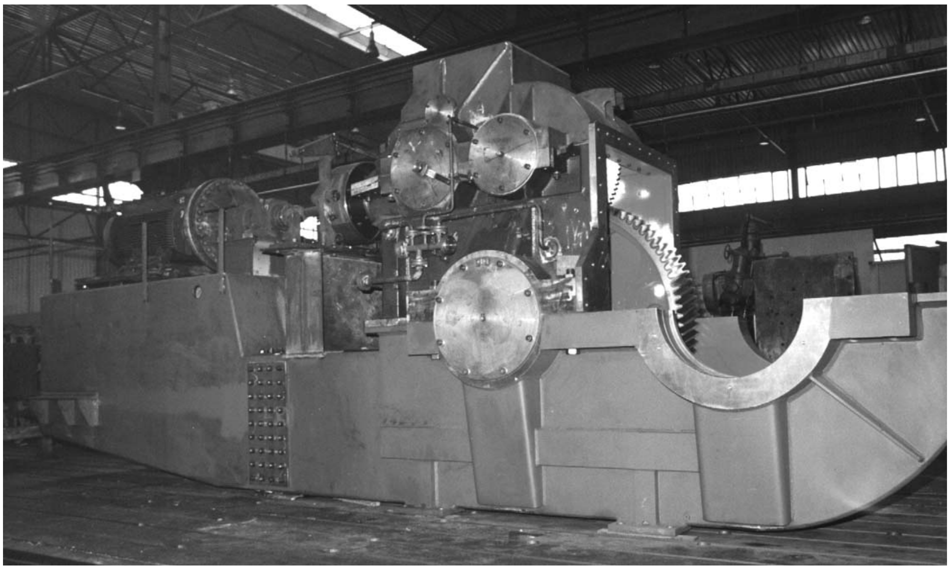


Fig. 2. Gear transmission of the bucket wheel excavator working wheel

Previous knowledge of load and deformation status, as well as the dynamic behavior of the transmission in excavation process would have the special significance so that the correct approach in designing its construction could be made.

Having in mind that the excavating process is periodical in nature, because of the buckets cyclic entering and exiting the excavated material, the precise determination of the torque on the output shaft through a mathematical function is impossible. In order to precisely determine the torque on the output shaft of the transmission, the tensimetric measurements of deformations need to be done, on whose basis one can calculate the value of the torque as a dominant parameter for calculus of all kinematics parameters of the transmission (gears, shafts, bearings, way of connecting the gear to the shafts, etc.).

Measurment results, expressed by normal deformation ε, can be turned into the tangential stress τ through the elasticity module E and Poisson's coefficient ν (1), which with polar moment of resistance of the cross section W_p determines the torque T (2) on the working wheel torque, which is required for the further calculation of the structural elements of the transmission.

$$\tau = \varepsilon \cdot E/(1+\nu) \tag{1}$$

$$T = W_p \cdot \tau \tag{2}$$

3. PROCESSING OF MEASURED VALUE - TORQUE

The most convenient from of representation of accidental processes of working loads characteristics, what the torque is on the shaft of the working wheel of the drive transmission and the loads for the corresponding probabilistic calculations of the structural elements of transmission is their discretization and statistical processing with the aim of obtaining the load spectrum. The load spectrum here comprises the representation of the results of the statistical processing of the accidental values (momentum on the output shaft of the transmission) in the form of the distribution functions of certain discrete values which characterize the working loads and stresses. Obtaining the corresponding load spectrums, and for the experimental evaluation of the stress in the elements of transmission in the area of weather* strength of the elements material, requires the following action:

- Choose the parameter whose value will be measured in the exploitational conditions and choose the procedure-methods of discretization. At working wheel drive transmission of the bucket wheel excavator, it is certainly the torque on the output shaft, as a dominant influential factor.
- Perform a statistical analysis of the data of the discretised criterion and their graphical and analytical description through the laws of probabilistic theory and mathematical statistics.

Basic characteristics of the accidental functions are: amplitude values, minimal, mean and maximum values, number and occurrence speed of the certain criterion for the defined working period, total number of changing cycles during lifetime etc. These methods of discretization to separate registered criterion of discuss process for statistical processing. They are based on the certain hypotheses which derive from the simplified physical representation of the damage accumulation in the material due to fatigue. They comprise the appropriate discretization of the accidental processes because to separate and statistical processing number of change or cycles definitely of criterion classic methods of theory reliably and mathematics of statistic.

3.1. Discretization of accidental processes

There is a large number of methods, which attempt to replace the real load and stress process change with simple process aimed at easier recognition and classification of the properties. It is a basic intention to make the approximation process in material due to fatigue, as close to the real process as possible.

Depending on the number of parameters, the discretization methods can have one, two or multitude of parameters. For discretization of dominant parameters, for the working wheel drive transmission, with respect to its specific position on the excavator boom, and to the system of connection to the working wheel, it is best to use methods with two parameters. These methods classify two variables, for example the amplitude and the mean values (of torque) or maximal upper and minimal lower values.

For discretization and defining of the torque values, the digital procedure is used, which comprises determination the accidental function $X = f(t)$ value in equal time intervals Δt. Discretization step can be determined from the relation (3)

$$\Delta t=1/(4\div 6)\cdot f_{max} \tag{3}$$

where f_{max}- maximal frequency of process oscillation.

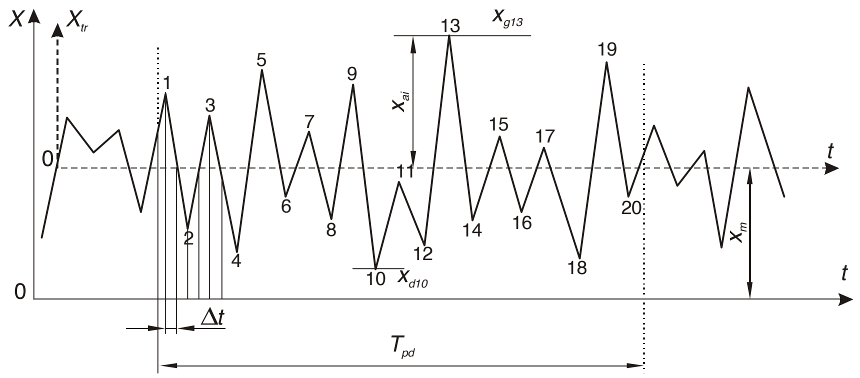


Fig. 3. Accidental function discretization

As to the time intervals Δt they have to be chosen in such a manner to obtain as realistic picture of deformation of the output shaft as possible, by calculation of the torque T. Time interval ought to be as short as possible, that is, in the recommended expression (3) smaller values should be taken. Discretization is done by three consecutive digital values (X_{i-1}, X_i, X_{i+1}), and the extreme values ($X_{ex} = X_{gi}$ or X_{di}) of accidental function and their number N_{ex}. are determined. Condition for singling out of those extremes is given following relation:

$$(X_{i-1}-X_i)(Xi-X_{i+1}) < 0 \;\Rightarrow\; X_i=X_{ex} \quad (\, i=1,2,3,...,N_{dg}) \tag{4}$$

where N_{dg} – is the number of digitalized points for the observed period of discretization of accidental function.

The processing procedure of the accidental processes comprises defining of the width and the number of classes in order to classify and statistically process the chosen parameters criterion. At some dissertation methods (which are the most appropriate for processing of the accidental processes of the working wheel drive transmission the bucket wheel excavators) it is done at the beginning of the discretization by sharing the range of the accidental processes into a number of equal classes. The number of classes is chosen so that the character of the classified criterion of the process is more clearly identified.

Error in statistical processing is reduced with the increase of the number of classes. The number of classes for such mechanical systems, as the bucket wheel excavator, is recommended to be $N_{k\,l} = 10 \div 20$.

For discretization of the accidental processes of the working load and stress changes aimed at creation of a basis for development of the calculation programs, there is a large number of methods: extreme values method, proper level cross-section method, span method, cycle method, etc.

After discretization of the accidental load processes and identifying the properties important for the fatigue of material, their statistical processing is undertaken. Processing comprises arrangement of the singled out properties with the aid of corresponding diagrams and tables, so that the regularity of distribution of the analyzed property could be determined. In discretization process, out of the singled out set of amplitudes X_{ai}, variation series is formed in the increasing series X_{a1}, X_{a2}, ... X_{an}. In such series, the number of amplitudes X_{uk},is known, and they comprise one load or stress section for the determined representative period of exploitation. In the next step, the division of the variation interval into a certain number of classes (k) is done, (the classes being of equal width ΔX_k) as well as arranging these amplitudes into classes by their values.

As a result of this processing, a pair of numbers [X_i, n_i] is obtained for each class, and it shows how many of n_i amplitudes are located in i class. The set of these value pairs is called the probability law of X_i variable, and the number of occurrences of singled out amplitudes by classes is called the frequency.

Very often, this statistic processing is conducted in relative coordinates, whose graphic interpretation give as histogram of the probability density function $f(X)$, histograms of cumulative increasing $F(X)$ and cumulative decreasing distributions $H(X)$. Cumulative decreasing distribution is the most frequently used form for presentation of the load spectrum. Absolute or relative values of the load or working stress amplitudes are applied on the ordinate, and the relative or absolute values of the number of occurrences of those amplitudes is applied on the abscissa. Because of the easier comparison of load spectrum with the Weller's curve of fatigue of material, the decreasing cumulative distribution is presented in the logarithmic scale of the abscissa. This change si presented in the form of the unit spectrum with the chosen number of changes $n_b = 10^5$. Value of the unit spectrum is chosen randomly, most often between (10^5 or 10^6).

4. PROGRAM SYSTEM FOR DESIGN OF GEAR POWER TRANSMISSION

System for design of gear power transmission is very complex and heterogeneous in structure. The system is developed on the modular principle, which facilitates execution, aided by a computer, of certain activities and tasks of a designer. The basic task of this system is to enable the integrated application of various program modules and systems developed by the authors and various companies, and which are intended for the automatization of certain activities in designing the gear power transmission. Because of this, the software platform of the developed system, relies on the maximum application of all available standards in the area of data exchange, communications and informatics.

The integrated program system for designing of power transmission **PTD** (Fig. 4) consists of three units.

1. program modules for calculation of power transmission elements,
2. program modules for calculation of rotary movement elements,
3. program modules for calculation joint of shaft-hub.

The first unit comprises the program modules for calculation of cylindrical, bevel and worm gears, friction, chain and belt transmission, the second unit comprises the program modules for shaft and roller bearing calculation, the third comprises the program modules for calculation of the wedges, groove joints, cylindrical pressed joints and pins.

Program module for calculation of gear couples consists of:

CPTD1 - program module intend for calculation and optimization of geometric-structural characteristics of the gears,
CPTD2 - program module for the final calculation of cylindrical and bevel gears,
PPTD1 - program module for conceiving of worm transmissions,
PPTD2 - program module for calculation of geometry and strength of worm couples.

Program module CPTD2 is intended for the final calculation of the gears and has the following capabilities:

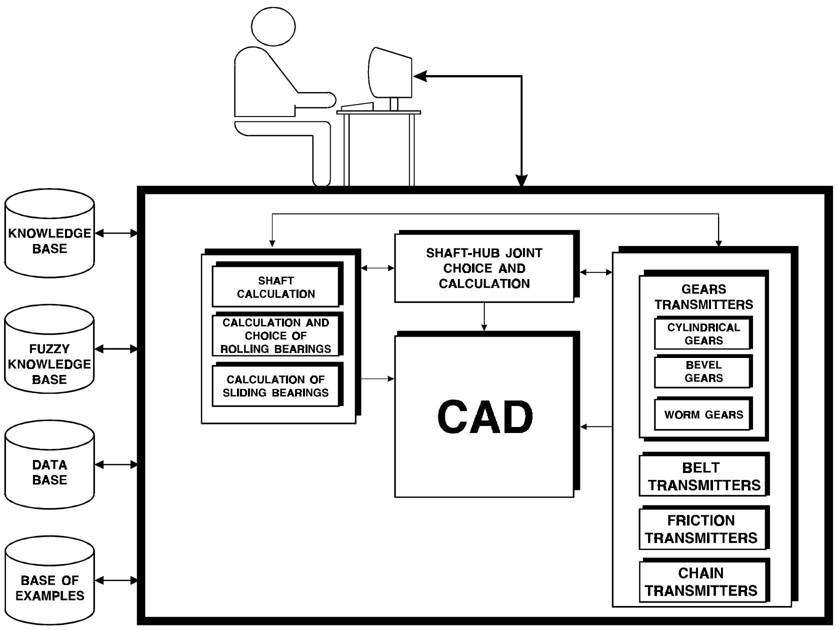


Fig. 4. Architecture of the program system for designing of power transmission

- calculation of precise geometry of cylindrical involute gears with exterior and interior teething for the given parameters of tools and calculation of bevel gears
- automatic choice of axial distance for previously set standard order,
- automatic determination of the bottom clearance and shortening of the head of a tooth for a whole number of addendum circle circumference,
- calculation of the working conditions factors through the load spectrum,

- calculation of bearing capacity according to the DIN-3990, AGMA - standard and ISO-recommendations,
- Calculation of elastic deformations of teeth,
- Calculation of bearing capacity in respect to pitting, forced and fatigue failure and scuffing,
- Calculation of bearing capacity through simulation of exploitation conditions,
- Calculation of bearing capacity of cylindrical and bevel gears with involute, straight and spiral teeth.

This paper gives the calculation of the bearing capacity of gears in non-stationary changeable exploitation conditions by the simulation of exploitation conditions.

Present examination of resistance to destruction of the mechanical parts is done by determination of the nominal load, that is nominal work load which corresponds to the most frequent load during the course of work. Exploitation conditions, that is occurrences of load higher than nominal is taken into account Through the factors of working conditions which is chosen approximately from the corresponding tables. It is not hard to conclude that such a way of determination of working load is the approximate one, what leads to inadequate dimensions of structure elements.

High level of precision in dimensioning and examination of resistance to destruction can be achieved through measuring the exploitation loads and an identification of the load spectrum of the vital structure elements. Load spectrums are obtained on the basis of exploitation measurements MS in the working process for certain conditions, so, therefore each load spectrum has its occurrence probability. Choice of the valid spectrum is solved through introduction of several load spectrums representing certain working conditions, which facilitates a sufficiently precise evaluation for all the conditions within their scope.

Load spectrums are usually expressed through the relative frequency of certain amplitudes $f(x)$ where

$$f(x) = \frac{\sigma_{ai}}{\sigma_{a\max}} \tag{5}$$

Load spectrum is also the basis for determination of critical stresses. At present calculation methods for the critical stress of dynamically stressed parts, the dynamic resistance which is obtained at constant ratio of highest and lowest stress and for destruction probability being around 0,5, is used. For more precise calculations of resistance of the parts to destruction it is necessary to introduce the resistance that corresponds to the manner and number of changes of working loads in the working life of the observed part – working durability.

Data on working durability are obtained by experimental examination of experimental parts or elements for certain spectrums of working stresses in laboratory conditions. Basic problem is the scope and duration of examination, conditioned by the need to examine a large number of load spectrums so that the more realistic data on the value and regularities of dissipation of examination results. That is why this problem is solved by finding the relationship of working and basic durability, that is by the application of the hypotheses of the damage accumulation in material. The basis of this hypothesis is the assumption that each stress s_i in the spectrum contributes to the damage proportionate to the number of cycles to destruction N_i, where $n_i < N_i$. Destruction occurs when the sum of all the damages reaches the value:

$$D = \sum \frac{n_i}{N_i} = \sum D_i \tag{6}$$

According to some hypotheses $D = 1$.

The experimental examination of this hypothesis for different stress conditions, however, showed significant departures. Because of that, different modification of this hypothesis were conducted, primarily in respect to the sum of the damage etc.

5. THE CALCULATION OF THE GEAR LOAD CAPACITY FOR THE CRITERIA OF ENDURANCE OF THE TOOTH FLANK AND ROOT

Calculation is based on the fact that each revolution of the gear has a corresponding damaging action. Damage depends on the value of the stress, and at small stresses it can be neglected. Working life which is obtained in such a way is the measure which characterizes the available resource of material. In order to conduct such a calculation, it is necessary to have the load spectrum, precise characteristics of material resistance and to apply the appropriate hypotheses on damage accumulation in material. There, the calculation will be conducted for Palmgren-Miner hypotheses where $D \approx 1$.

Load spectrum is approximated to the appropriate blocks, where each torque T_i has a corresponding number of load change cycles n_i. On the figure 5, apart from the load spectrum load, there is also the Weller's curve line of permissible load on a gear.

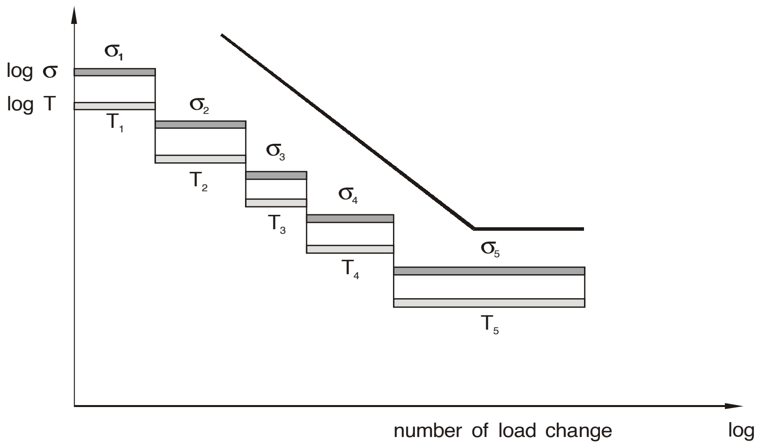


Fig. 5. Load spectrum

For each level of load spectrum T_i a corresponding stress on the flank of the tooth is determined according to:

$$\sigma_{Hi} = Z_H \cdot Z_E \cdot Z_\varepsilon \cdot Z_\beta \cdot Z_{B,D} \sqrt{\frac{2000 T_i}{d_1^{\,2} b} \frac{u+1}{u} K_{vi} \cdot K_{H\beta i} \cdot K_{H\alpha i}} \tag{7}$$

where: Z_H - zone factor,
Z_E - elasticity factor,
Z_ε - contact ratio factor,
Z_β - helix angle factor,
$Z_{B,D}$ - single pair tooth contact factor for the pinion, for the wheel,
K_{vi} - dynamic factor,
$K_{H\beta i}$ - face load factor,
$K_{H\alpha i}$ – transverse load factor.

In expression (3) the application factor K_A, is not used because it is taken into account throughout the load spectrum.

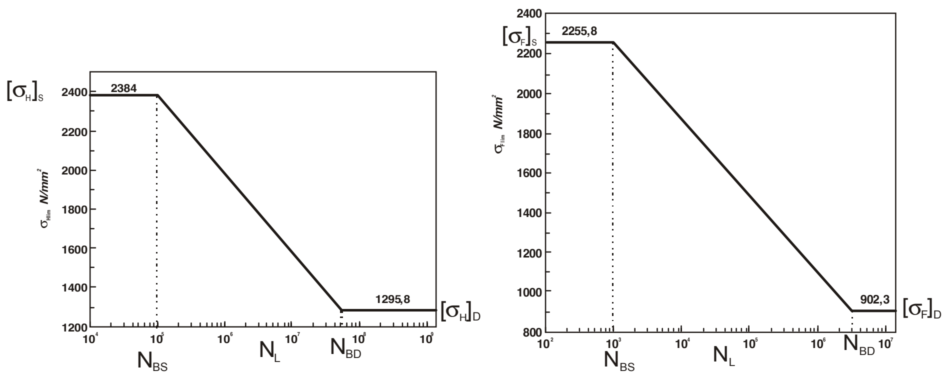
The material resistance characteristic is determined through the Weller's curve, that is damage line.

Inclination of Weller's curve (fig 6.a) is:

$$p = \frac{\log \dfrac{N_{BD}}{N_{BS}}}{\log \dfrac{[\sigma_H]_S}{[\sigma_H]_D}} \tag{8}$$

The number of load change cycles to failure for any level N_i equals:

$$\log N_i = p \cdot \{\log[\sigma_H]_S - \log[S_H \cdot \sigma_{Hi}]\} + \log N_{BS} \tag{9}$$



a) b)

Fig. 6. Damage lines for damage statistical probability of 10%

Working load in the root of a tooth for each level of the load spectrum T_i is determined through

$$\sigma_{Fi} = Y_{Fa} \cdot Y_{Sa} \cdot Y_\beta \cdot Y_\varepsilon \frac{2000 T_i}{d_1 \cdot b \cdot m_n} K_{vi} \cdot K_{F\beta i} \cdot K_{F\alpha i} \tag{10}$$

Material resistance characteristics are determined through the damage curve shown on the fig. 6.b, which implies that

$$p = \frac{\log \dfrac{N_{BD}}{N_{BS}}}{\log \dfrac{[\sigma_F]_S}{[\sigma_F]_D}} \tag{11}$$

Number of load change cycles to the failure for any level N_i is

$$\log N_i = p \cdot \{\log[\sigma_F]_S - \log[S_F \cdot \sigma_{Fi}]\} + \log N \tag{12}$$

The algorithm of the calculation factor of safety from pitting S_H and factor of safety from tooth breakage S_F is shown in figure 7. The calculation is iterated until the damage is the following bounds $0{,}95<D<1{,}05$.

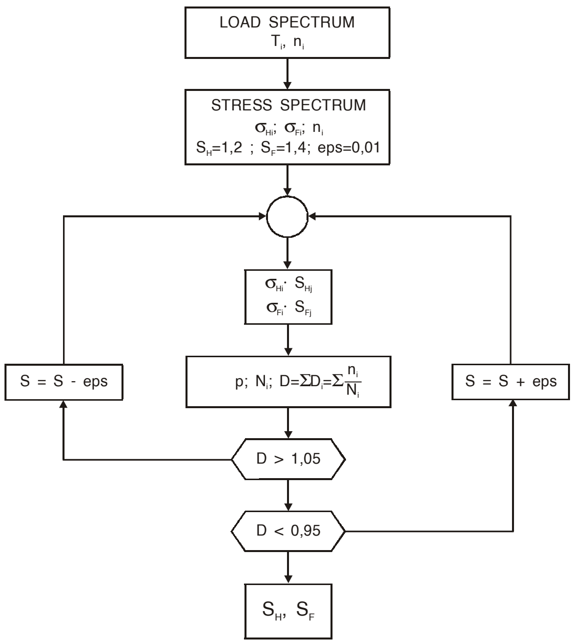


Fig. 7. The algorithm of the calculation factor of safety from tooth breakage and factor of safety from pitting

6. CONCLUSION

From the previously said, it can be concluded:

1. Because of the precision of calculation of working loads, it is necessary to know the value, course of the change, frequency as well as occurrence probability of the highest stresses occurring in the working life of the observed part of the structure.
2. Having in mind that the excavating process is periodical in nature, because of the buckets cyclic entering and exiting the material being excavated, the precise determination of the torque on the output shaft through a mathematical function is impossible.

3. The value of the real load of the structure elements of the working wheel transmission are obtained by the exploitation examination, and then the load spectrums are formed by statistic methods.
4. Calculation of the gears by the simulation of the exploitation condition, is based on the fact that each revolution of the gear results in a corresponding damage, that is the calculation is conducted for Palmgeren-Miner hypothesis.
5. At the Faculty of Mechanical Engineering of Niš, the integrated intelligent system for simultaneous design of gear power transmission is developed. This program system facilitates the calculation of the gears through the simulation of the exploitation conditions.

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PRORAČUN STRUKTURNIH ELEMENATA PRENOSNIKA RADNOG TOČKA ROTORNOG BAGERA

Dragan Milčić, Slobodan Miladinović

Noseća konstrukcija rotornog bagera predstavlja složeni mašinski sistem sa jako izraženim elastičnim svojstvima pojedinih delova. Strela bagera, zbog vešanja izvedenog čeličnim užadima i nepovoljnog položaja radnog točka, koji predstavlja veliku koncentrisanu masu u stalnom zahvatu sa otkopnim masivom, izložena je opterećenjima složenog dinamičkog karaktera, koja potiču od otpora kopanja i pokretanja celokupne mase bagera. Konstruisanje zupčastog prenosnika za pogon radnog točka rotornog bagera, kao složenog mašinskog sistema, predstavlja složen projektno konstrukcijski zadatak za čije rešavanje je neophodna primena savremenih metoda konstruisanja. Visok stepen tačnosti pri proračunu strukturnih elemenata prenosnika radnog točka rotornog bagera može se postići simulacijom eksploatacionih uslova. U radu je dat pristup proračuna elemenata prenosnika za pogon radnog točka rotornog bagera zasnovan na normiranim spektrima opterećenja.

Ključne reči: prenosnik radnog točka rotornog bagera, proračun, spektar opterećenja