ZESZYTY NAUKOWE POLITECHNIKI ŚLĄSKIEJ 1986

Seria: GÓRNICTWO z. 143 Nr kol. 8883

1st International Conference - Reliability and Durability
of Machines and Machinery Systems in Mining
1986 JUNE 16-18 SZCZYRK, POLAND

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RELIABILITY AND DIAGNOSTICS OF TOOTHED GEARS FOR BELT CONVEYOR DRIVES

Summary. The paper, together with [1], is an attempt to illustrate and connect problems associated with the mesh life computations, reliability and diagnostics of the gear units.

The paper presented an introduction to the reliability model of the toothe gear at assumption of a limited life of the mesh. Probable causes of high failrure rate of the gear units are analyzed. A description is given for the diagnostic procedure of gear units, basing on the classification of mesh states. A statistical verification is presented for the class of states of the gear units being in good condition.

The diagnostic procedure is now being used in the lignite mines, and 200 toothed gears are supervised using diagnostic devices under consideration in the paper. The data gathered will serve to verify the reliability model and give a basis for directions in the structural changes.

1. THE INTRODUCTION TO THE RELIABILITY MODEL OF CHANGES IN TOOTHED GEAR CONDITION

The elements of toothed gear, gear mesh and rolling bearings undergo a natural wear and damage. The damage and natural wear are a reflection of the technical condition of the gear, this condition being a subject of diagnostic measurement. The phenomenon of gear mesh or bearing damage is of a statistical character, i.e. it is possible to determine the percentage fraction of a large group of bearings which will undergo damage at a given load being equivalent to the definite number of the cycles of changes in load. The life of a given type of the bearing is determined by the rated life corresponding to 10-percent level of damage. As resulted from tests, a 50-percent life and maximum life are five and 20 times, respectively, in excess of the rated life.

It has been assumed that gear mesh life against pitting can change in the similar manner.

An illustration of the relation between load capacity of the gear mesh against bending and pitting, as well as the effect of load capacity of the bearing in function of a number of the cycles of changes in load, is shown in fig. 1. It illustrates the relation between logarithm of relative bending fatigue strength log Z/Zo and logarithm of relative compressive fatigue strength log k/ko in function of the logarithm of a number of changes in equivalent load. The line "m" illustrates the value of relative bending strenght at a 50-percent level of the reliability for a material used to make the gearing.

Assuming safety factor to be $\delta = 2$, the line "m" will be situated just as shown in fig. 1. The line "m" illustrates lower limit of changes in the strength, and line "g" represents upper limit of changes in the strength. The upper and lower lines correspond to the tenfold change in the material life, as accepted after [2]. Possible spread of the loss of gearing load capacity due to the pitting has been accepted just as for rolling bearings after [3], as given earlier. Assuming the limit number of cyles for the contact strength $5 \cdot 10^7$ to be adequate to the mean value of damage spread, the life corresponding to 10-percent reliability is adequate to 10^7 cycles, and the maximum life corresponds to $20 \cdot 10^7$ cycles. Assuming that a devolepod pitting will occur at an expected value of its occurrence, or $5 \cdot 10^7$ cycles, and the strength of a concrete gear unit relates to the lower limit (line d point A), a change in the condition of material behaviour will take place at thie point, and the tooth will behave as made of the material with a notch. The relation between the relative strength and number of cycles will be changed due to the change in line inclination (index m = 3). For comparison, the index has been assumed to be m = 8 for the material without notch. At point B (on line AB), the loss of bending fatigue strength may occur due to the breakage of the gear mesh. In case when the gearing strength corresponds to the maximum line "g" and similarly developed pitting will occur at $5 \cdot 10^7$ cycles the line CD illustrates the loss of gearing load capacity due to the breakage. The interval corresponding to the spread of life at an expected value of pitting occurence equal to $5 \cdot 10^7$ cycles is denoted as Δ_1. In extreme cases, the expectation may be that a process of the loss of gearing load capacity will occur according to the line 3 when pitting has been developed in the gearing already at a 90-percent life and in the second extreme case when pitting has been developed at the maximum life corresponding to $20 \cdot 10^7$ cycles, i.e. according to the line of load capacity loss due to the breakage 4. A full range of tooth failrure due to the breakage, assuming the pitting to be occurred earlier, is denoted as Δ_2. The above reasoning may serve to predict gearing damage due to the breakage if, using diagnostic procedures, a moment will be determined when pitting has been developed in the gearing.

One of the reasons for an incorrect gearing operation is the loss of load capacity of rolling bearings. On the basis of design practice, the bearings of toothed gears are selected individually to have 90-percent life. However, the bearings decide on the reliability of a subassemby altogether. For toothed gears of conveyor drives under consideration, four bearings are selected to ensure a correct operation of a driving shaft. To ensure a correct operation of bevel stage six bearings have been selected. The determination of a resultant load capacity has no reflection in the structural computations. The relation between individual load capacity of the bearing and load capacity of bearing set can be determined approximately making use of the following example.

Let's assume that individual load capacity of the bearing has been selected in such a manner that its life attains $3 \cdot 10^9$ cycles, which corresponds to $5 \cdot 10^4$ hours of the gear life at 1000 r.p.m of the first shaft. It is shown in fig. 1 where broken line 1b is drawn which illustrates the individual life of particular rolling bearings. Assuming that a correct operation of driving shaft is provided by four roller bearings the relation between the individual life $L_1 = 3 \cdot 10^9$ cycles and bearing life L_1 is as follows

$$\frac{L_i}{L_1} = i^{-\frac{70}{27}}$$

for $i = 4$

$$L_4 = 0{,}0276\ L_1 = 0{,}0276 \cdot 3 \cdot 10^9 = 8{,}28 \cdot 10^7 \text{ cycles.}$$

To ensure a correct operation of the gearing six rolling bearing are needed. We assume that individual load capacity of bearings has been selected correctly, and it corresponds to the life of 50 000 hrs, and asseming further on that bearings are of a roller type we'll obtain as follows

$$L_6 = 0{,}0096\ L_1 = 0{,}0096 \cdot 3 \cdot 10^9 = 2{,}88 \cdot 10^7 \text{ cycles.}$$

Or, the life of bearings constituting a set of four bearings will be change over, relative to $3 \cdot 10^9$ cycles being assumed, to $8{,}28 \cdot 10^7$ cycles.

Because the set of six bearings is decisive for correct operation of the gearing an effect on the gearing life will reflect, already at $2{,}88 \cdot 10^7$ cycles, the load capacity of the bearing represented by broken line 2b. The hourly life corresponding to $2.88 \cdot 10^7$ cycles at 1000 r.p.m. is 480 hrs.

Fig. 4 illustrates a histogram of current input of the motors used to drive the gears. These are current intensities flowing through the motor during diagnostic measurements. The mean value of current input is 62.5 A, and the rated current input should be 118A, or the real load averages 0,53 of rated load. Considering the effect of load, the mean value can increase up to about 8.10^8 cycles (see fig. 1), which corresponds to the hourly life of about 5000 hrs. The presented model and analysis of causal connections is an introduction to the problem. Collection of suitable operating data on the life of elements will make it possible to prove or reject the above considerations.

2. FUNDAMENTALS OF DIAGNOSTIC PROCEDURE FOR TOOTHED GEARS OF BELT CONVEYOR DRIVES

A basis of diagnostic procedure for the gearing is band-like and selective nature of important vibration effects generated by the gearing interaction. The vibration effects producing a diagnostic signal may be described, among other things, by a spectrum of signal. Depending on the gearing condition, the signal spectrum forms a certain structure. The components of frequency of the tooth contact and their harmonics, and also components of the gearing run-out and their harmonics, enter into the composition of the structure of spectrum generated by the gearing. Apart from above mentioned components, the nonfrequency components can occur which reflect the interaction of particular teeth. The presented diagnostic procedure consists in the determination of gearing interaction condition by the component of gearing run-out and its harmonics.

In the procedure, the measurements called routine and identifying ones are distinguished. The routine diagnostic measurements consist in measuring the root-mean-square (rms) value of vibration velocities and accelerations mm/s and m/s^2, respectively.

These quantities are measured in the following frequency bands: 10-100 Hz, 1000-3500 Hz, 3500-10 000 Hz. The components of gearing run-out and harmonics are encountered in the band 10-100 Hz, and components of tooth contact and harmonics - in the band 100-3500 Hz. As evidenced by practice, most information is available concerning the gearing condition in the band 100-3500 Hz. The identifying measurements are carried out by means of narrow band filters in the bands 10-100 Hz. 100-3500 Hz, and the band width of narrow band filter is 10 and 30 Hz, respectively. With identifying measurements it is possible to determine, by way of signal separation, from which stage of the gearing of multi-stage transmission the signal is generated.

A basis to take diagnostic decisions is the classification of states, fig. 2, and the course of parameters in time (trend of parameters). The classification was originated as a result of determination of a direct relation between the real technical condition of gear unit and vibration parameters. The classification is now being verified by the statistical methods. The interpretation of classes is of a techno-economic nature. The classes of condition for 100-3500 Hz should be interpreted as follows:

- class "A", normal operation of gear unit, to which correspond parameters up to 45 m/s^2, 20 mm/s, reflects the deferentiation of gear units regarding the accuracy of their execution; it reflects primary factors [4]. Also, class "A" comprises parameters reflecting initial wear and failure of bearings, which have no essential effect on the gearing operation, and it is not economic to put these gear units out of operation for their renewal.
  The parameters reflecting primary factors are very difficult to be separated from secondary factors (occurring in the first period of operation).
- class "B", i.e. "hazard for gear unit" relating to the range of 70 m/s^2 and 25 mm/s, reflects parameters of a gear unit in which rolling bearings should be replaced (it relates to the high-speed bevel stage) This replacement prevents gearing damage.

A decision taken to replace these bearings results from economic reasons. As a result of this operation, an economic effect is achieved which approximates the value of bevel gearing.

- class "C" i.e. "great hazard for gear unit" ranging to 90 m/s^2 and 30 mm/s, relates to the states which correspond to the further degradation of rolling bearings and gearing; there is a greater hazard for the gear unit to be damaged due to overload.
- class "D", i.e. "extreme hazard for gear unit" ranging to above 90 m/s^2 and 30 mm/s; the gear units should have repaired to avoid failure.

The repair of gear unit in such condition consists in the replacement of bearings and toothed wheels, reboring of bearing seats. The values 90 m/s^2, 30 mm/s delimit possible failure prevention.

It is worth saying that the values of acceleration and velocity are averages obtained from four measuring points; such a procedure is justified by the results of [4]. The parameters of presented classes, fig. 2, relate to the gear unit load which corresponds, in case under consideration, to an intensity of current input by the motor equal to 100 A.

As evidenced by tests, there is a relation between the load and diagnostic parameters of the gear unit. The diagnostic parameters of gear unit being in suitable technical condition do not show, or show to an inconsiderable extent, an effect on load. Fig. 3 illustrates the effect of load in terms of current intensity (A) on diagnostic parameters. In fig.3 a histogram is shown for vibration parameters, this histogram being obtained from measurements of 145 gear units. The histogram relates to th vibration parameters of acceleration ranging from 100 to 3500 Hz. The mean values of these parameters are equal to $a_{śr}$ = 14,8 m/s^2; standard deviation is S_A = 6,53 m/s^2.

In fig. 3, the limit is determined.

$$a_{śr} + 3\ S_a = 14,8 + 19,59 = 34,43\ \mathrm{m/s^2}$$

This limit has been accepted as that for a gear unit of suitable quality. The point of intersection X is a point where border-line of parameters of the gear unit having suitable quality and medium line of currant intensity intersect, this intensity being maintained durign diagnostic mrasurements. The histogram of current input by the motor, this input being provided durign measurements, is shown in fig. 4. Point X indicates the upper limit for gear unit of suitable quality. At point A, the broken line is drawn which takes the effect of load on diagnostic parameters into account.

As can be seen from fig. 3, the predicted upper limit for class A is drawn by a full line. The limit for class A was determined basing on visual inspection of relatively small number of gear units but on the basis of a direct relation between technical condition and diagnostic parameters. The broken line may be considered as a line of statistik verification.

As evidenced by practice experienced until now in using the diagnostic procedure to evaluate gearing condition it is also necessary to introduce the load factor. This factor is defined to be

$$\psi = \frac{a_2 - a_1}{A_2 - A_1} \qquad \frac{\mathrm{m/s^2}}{\mathrm{A}}$$

where: a_2 and a_1 - accelerations in the band 100-3500 Hz and motor current intensities A_1 and A_2 corresponding to them, a_2 - higher value of acceleration. Apart from above mentioned parameters, the procedure requires vibration parameters to be determined in the bands 10-100 Hz for velocities and accelerations -parameters to be determined in band 3500-10 000 Hz. The diagnostic procedure presented is now being used to supervise some 200 high-power gear units applied to drive belt conveyors.

3. DEVICES FOR DIAGNOSTIC TESTS OF TOOTHED GEARS

The diagnostic measurements of toothed gears (i.e. measurements of vibration parameters) are carried out by means of suitable diagnostic devices designed and produced in small series by Poltegor Institute/Wrocław.

For more details of the devices see [5]. The devices are designated as follows: UPD1 (universal diagnostic device), PPD1 (ready-to-hand diagnostic device). The devices UPD1 and PPD1 are able to analyse the components of gearing run-out and their harmonics, and also the frequencies of tooth contact and their components.

The devices UPD1 and PPD1 are able to measure the root meansquare value of velocities and accelerations of vibration up to 100 mm/s and 316 m/s^2, respectively.

The device UPD1 has a wide-band filter with constant lower frequency 10 Hz and stepwise tunable upper frequency ranging from 100 to 300 Hz, with a step of 50 Hz. This filter is intended to find the root-mean-square value of low frequency vibration (gearing run-out) originated from high-speed stages of toothed gears. Additionally to the above filter, the device UPD1 has a filter with constant upper frequency 3500 Hz and variable lower frequency ranging from 100 to 300 Hz with a step of 50 Hz.

This filter is intensed to measure the root- mean-square value of component vibration of the gearing (as a rule it relates to the first high-speed stage of the gearing). The device has also a standard filter of measuring range 10-1000 Hz which can be used to determine vibration parameters according to the standard ISO [6].

Apart from a/m wide-band filters, the device UPD1 is able to analyse a signal by means of stepless tunable filters. One of them analyses the signals ranging from 10-100 Hz, band width 10 Hz.

The second one analyses the signals ranging from 100 to 3500 Hz, band width 30 Hz.

The device PPD1 is able to analyse roughly the signal by dividing the band ranging from 10 to 10 000 Hz in three subranges, f.ex. 10-100 Hz, 100-3500 Hz, 3500-10 000Hz. The device PPD1 is now being produced to have a shape more convenient to handle in on-site conditions, as compared to that presented in [5].

4. SUMMARY

The presented reliability model of toothed gears is developed basing on data to be at disposal of an designer when selecting structural features of the toothed gear. The model illustrates how, basing on data being at disposal of the designer (i.e. Zo, Ko, m), on assumed safety factor and diagnostic data, to make use of it to predict gearing damage due to the breakage.

A moment is determined when the developed pitting occurs in the gearing. If working time of elements is known, from the beginning of their operation, a degree of hazard for gear units to be damaged due to the teeth breakage can be found. In future, when data concerning types of failure to the bearings and gearing, as well as those concerning working times of elements, will be at disposal, the model will be used to obtain data enabling structural changes to be made in order to increase the gearing life.

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Recenzent: Prof. zw. hab. inż. Jerzy ANTONIAK

Wpłynęło do Redakcji: luty 1986 r.

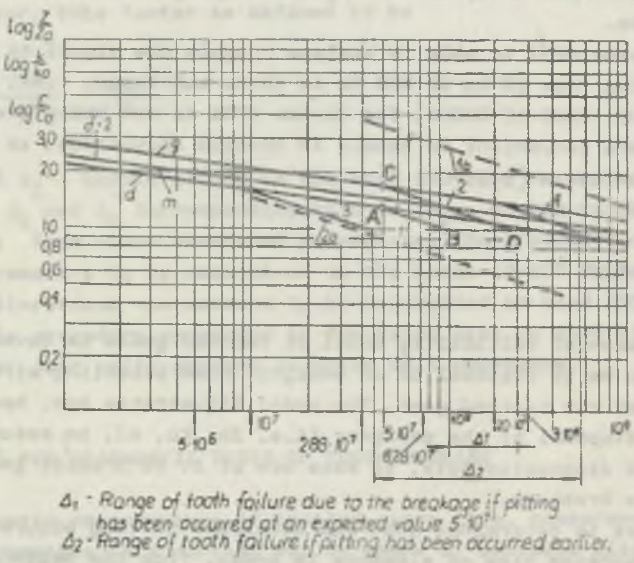

Fig. 1. Fundamentals for prediction of mesh failure due to the breakage when pitting has been occurred earlier

Rys. 1. Podstawy szacowania uszkodzenia zazębienia w wyniku złamania, w przypadku kiedy wcześniej wystąpiło zjawisko pittigu

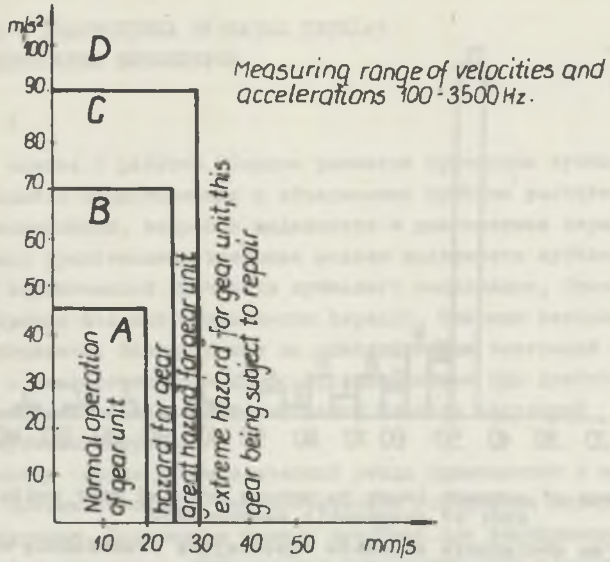

Fig. 2. Classification of the states of hazard for gear units, interpretation sea text of the paper

Rys. 2. Klasyfikacja stanu zagrożenia przekładni

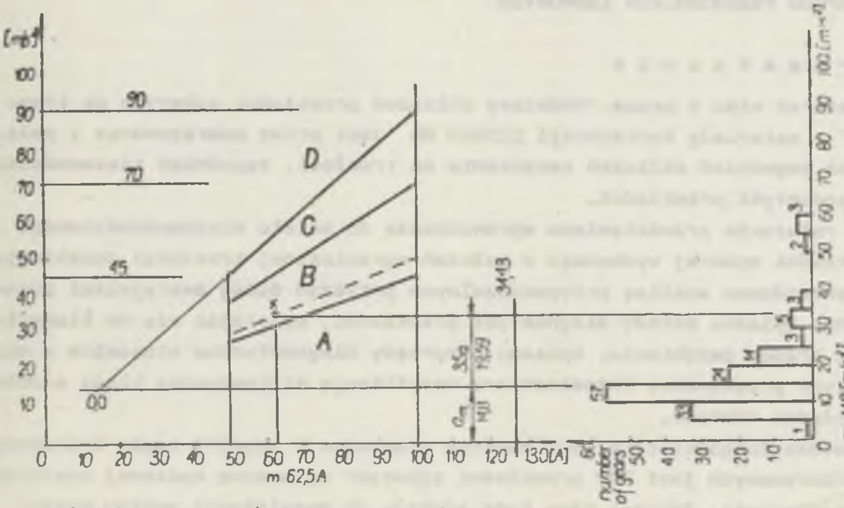

Fig 3 Effect of load on vibration parameters of the gear; statistical verification of the state limit of a gear in suitable quality

Fig. 3. Effect of load on vibration parametes of the gear; statistical verification of the state limit of the gear in suitable quality

Rys. 3. Wpływ obciążenia na parametry drgań przekładni, statystyczna weryfikacja stanu granicznego przekładni o określonej jakości

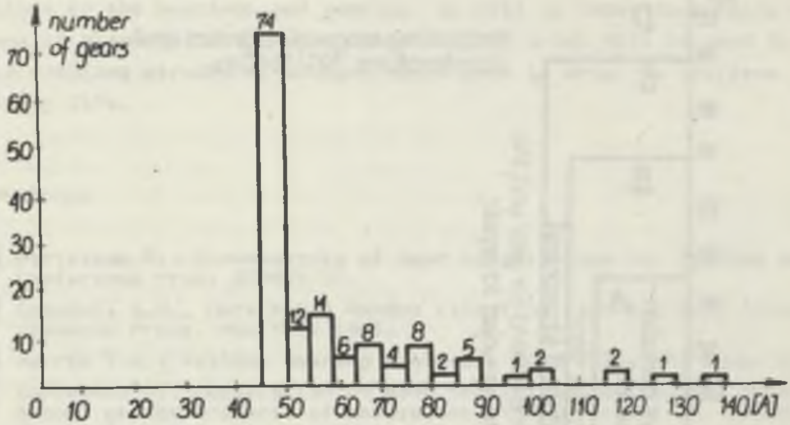

Fig. 4. Histogram of current imput by motors driving gear units at a moment of diagnostic measurement

Rys. 4. Histogram obciążenia silników napędzającą przekładnię w czasie pomiarów diagnostycznych

NIEZAWODNOŚĆ I DIAGNOSTYKA PRZEKŁADNI ZĘBATYCH DO NAPĘDU PRZENOŚNIKÓW TAŚMOWYCH

S t r e s z c z e n i e

Referat wraz z pracą "Podstawy obliczeń przekładni zębatych na trwałość" - materiały konferencji ICOREM 86 jest próbą zobrazowania i połączenia zagadnień obliczeń zazębienia na trwałość, zagadnień niezawodności i diagnostyki przekładni.

W referacie przedstawiono wprowadzenie do modelu niezawodnościowego przekładni zębatej wychodząc z założeń ograniczonej trwałości zazębienia. Przeprowadzono analizę przypuszczalnych przyczyn dużej awaryjności przekładni. Opisano metodę diagnostyki przekładni, opierając się na klasyfikacji stanów zazębiania. Opisano przyrządy diagnostyczne stosowane w diagnostyce przekładni. Przedstawiono weryfikację statystyczną klasy stanów przekładni dobrych.

Metoda diagnostyczna jest obecnie stosowana w kopalni węgla brunatnego i nadzorowanych jest 200 przekładni zębatych za pomocą opisanej aparatury diagnostycznej. Zebrane dane będą służyły do weryfikacji modelu niezawodnościowego i dadzą podstawę kierunku zmian konstrukcyjnych.

НАДЕЖНОСТЬ И ДИАГНОСТИКА ЗУБЧАТЫХ ПЕРЕДАЧ
ПРИВОДОВ ЛЕНТОЧНЫХ КОНВЕЙЕРОВ

Р е з ю м е

Доклад вместе с работой "Основы расчетов прочности зубчатых передач" является опытом представления и объединения проблем расчетов прочности зубчатых зацеплений, вопрооов надежности и диагностики передачи.

В докладе представлено введение модели надежности зубчатой передачи, исходя из ограниченной прочности зубчатого зацепления, Проведен анализ допустимых причин большей аварийности передач. Описаны методика диагностики зубчатой передачи, базирующаяся на классификации состояний зубчатого зацепления, и диагностические приборы, применяемые при диагностике передачи. Представлена статистическая верификация классов состояний действующих исправно зубчатых передач.

В настоящее время диагностический метод применяется в шахтах бурого угля; при помощи описанной диагностической аппаратуры контролируется 200 зубчатых передая. Полученные данные послужат для верификации модели надежности и дадут основание направления конструкционных изменений.