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APPLICATION OF WIBRO-ACOUSTICAL STUDIES IN PROCESS OF IMPROVING DEPENDABILITY OF ARM-MOUNTED CUTTER-HEADS OF KGS-320 SHEARERS

Summary. The paper describes an approch to evaluation of dynamic state of arm-mounted cutter-heads used in KGS-320 shearers, based on vibro-acoustical studies made in course of control diagnostic during idle running of the system in question. In general, simple dimensionless estimates have been used, reflecting changes in a vibration level at particular points of a cutter-head frame under test, conditioned by a change of sense of rotation of an output shaft. Relative comparison of new estimates obtained has enabled to trace the cutter-heads characterized by worsened technical conditions. The directional estimates introduced, characterized by nigh diagnostic susceptibility, represent changes of a vibration signal depending not only on the choice of a measuring point location but also on a serial number of the cutter-head under test. The studies described, are aimed to determine the change in vibration signal occurrence rather than to set its cause, the latter to be the subject of further studies because of kinematic complexity of the system tested. The studies shall be undertaken for the system at idle running and during operation under a nominal load.

The analysis of the results obtained suggests that the estimates introduced may constitute a basis for development of detailed classification criteria enabling qualitative evaluation of a cutter-head as a whole and of some of its kinematic elements.

1. INTRODUCTION

Vibro-acoustical processes are basic phenomena accompanying operation of each machine, resulting in wibration of machine surface and generating noise. Apart from close connection with machine operation, vibro-acoustic diagnostic is of constantly growing importance in manufacture, constituting one of product quality control procedures.

Three stages can be distinguished in the development of each mechanical device, i.e. desing, manufacture and operation. At each stage of manufacture of the machine or its elements inevitable errors are introduced resulting in random scatter of dimensions between individual products of the same manufacturing series.

At the stage of manufacture, cooperating elements of the machine are assembled into one functional unit. During manufacture of machines and elements random errors are introduced as compared with a hypothetical ideal standard despite careful selection and mating of rotational and slidable pairs also of permanent connections made to various methods.

Conclusion can hence be drawn that when surveying some machines of the same type, random scatter of geometrical/mechanical properties is observed contained within a certain tolerance range or situated outside this range.

Dynamic processes taking place during operation of machines (even when idling) represented by vibro-acoustic signals are diversified and a diversification degree depends on a randomness value introduced at the manufacturing stage.

This study has been limited to the use of simple dimensionless, directional estimates introduced lately, reflecting overall changes of a vibration level in specified points of cutter-heads under test, resulting from change of sence of rotation of an output shaft. One must also keep in mind that excessive vibration activity of the machine (dynamic strains, stresses) affects dependability and life of the machine. Therefore it is necessary to localize sources of vibrations (stresses) and noises ti introduce changes in the design of the machine and hence to minimize these phenomena to levels permitted by relevant standards and other regulations. Vibro-acoustic studies play therefore an important role in engineering practice contributing to quality of finished products.

2. FACTORS INFLUENCING DYNAMICAL STATE OF MACHINES

It is well-known from practice that even well designed engineering equipment can be incidentally damaged. As a rule, these are random cases where the device is transferred from the state of fitness for use to the nonoperational state.

Fig. 1 shows a graph of failure frequency versus time, characteristic for majority of engineering devices. Three basic periods can be distinguished in the curve in question. The first working period features mainly failures resulting from faulty workmanship, technological faults, operational defects and not to rare designing flaws. During this period a failure frequency is quite high because defects cummulate at this stage, the defects observed being mainly primary defects of the devices.

As the time lapses the failure frequency decreases passing to a second period characterized by a nearly constant failure frequency. In this period failures result from design limitations, fatigue, wear and sometimes from faulty operation.

The third period is characterized by ageing of the device anticipated by a designer and resulted from natural wear of elements, changing of material properties, etc. Conclusion arises that the well planned, designed and manufactured machine should exceed limits of linear variations after the same operation time for each element.

However operational vibro-acoustic diagnostic would be useless then, because restoration of the machine would consist in replacing the whole machine. The question therefore emerges, related to a failure frequency distribution: how do dynamic characteristics change during operational life of the machine?

It is generally known that basic causes of vibrations are as follows:

- unbalanced rotational elements,
- plain motions of masses,
- change of kinematic pair geometry,
- excessive plays at movable and permanent joints.

It si obvious therefore, that almost each cause of vibrations is related to mass decrement process, plastic deformation process or fatigue wear. Hence, a vibration level of a hypothetical machine should be correlated with a failure frequency according to a pattern shown in fig.2.

The running-in period (at testing) should be characterized by considerable variance of vibration amplitude, the magnitude of which should decrease to a certain value corresponding with the beginning of the stage II, the latter characterized by almost linear increase in the vibration level. The last stage preceding the breakdown should be characterized by rapid growth of the vibration level. This shows that the vibration process accurately reflects operational condition of machines.

Vibro-acoustic inspection tests are hence very important in the process of improving quality of machine or its operational life and dependability. These tests are, in general, indirect estimations and are based on assumption that the dynamic processes taking place in a machine reflect geometrical, mechanical and functional properties of the machine with sufficient accuracy. This concerns both determined properties given at the stage of designing and random properties obtained at the stage of manufacture. Such vibro-acoustic tests make a part of quality control procedure and are aimed at checking whether machine properties reflected by dynamic processes occuring in the machine are within limits defined by relevent criteria of technical acceptance and commissioning.

3. DIMENSIONLESS AMPLITUDE DISCRIMINANTS

Many of studies concerning vibro-acoustic diagnostic of machines relate to the use of basic functional discriminants as power spectral density function, amplitude probability density function and correlation function for evaluation of dynamical status of various mechanical devices [1, 2, 3]. In most of these studies it has been shown that specified functional discriminants of vibro-acoustic processes for efficient and inefficient machines greatly differ both quantitatively and qualitatively and hence can be successfully utilized in vibro-acoustic diagnostic of machines [4].

Intensive studies are made concurrently to find new, efficient methods of vibro-acoustic process data processing thus enabling utilization of other properties of the processes in question, reflecting more comprehensively the dynamic status of the system under test [5].

Susceptibility of dimensional discriminants to changes of machine working conditions has encouraged studies on quotient of these quantities i.e. on dimensionless discriminants characterized by lack of this unfavourable attribute [6].

In diagnostic of transmissions and bearings being the main elements of arm-mounted cutter-heads some simple characteristics of vibro-acoustic state are utilized enabling relative, comparative estimation of the overall dynamical condition of the cutter-heads under test.

A general definition of amplitude dimensionless discriminants of vibro-acoustic process x(t) is expresses by the formula:

$$X_x = \frac{\left[\int\limits_{-\infty}^{\infty} |x|^l \cdot p(x)dx\right]^{\frac{1}{l}}}{\left[\int\limits_{-\infty}^{\infty} |x|^m \cdot p(x)dx\right]^{\frac{1}{m}}}$$

where:

$p(x)$ - probability density function of amplitudes,

l, m - moments of suitable orders

Assuming $l \to \infty$ and $m = 2$ an expression defining a so called peak factor is obtained

$$C_x = \frac{\hat{x}}{x_{RMS}}$$

where:

$\hat{x}$ - peak amplitude being mean measure of maximum values

x_{RMS} - rms amplitude

Our study is limited mainly to illustrating effect of output shaft sense of rotation on relative measures of peak factor of vibration displacement and velocity for r.h. and l.h. rotation. The following factors are introduced:

$$C_{xP} = \frac{\hat{x}_p}{(x_{RMS})P}, \qquad C_{xL} = \frac{\hat{x}_l}{(x_{RMS})L} \tag{1}$$

where:

index P describes r.h. rotation while index L l.h. rotation.

To determine effect of output shaft sense of rotation on generation of vibration signals at individual points of the frame, the dimensionless peak directivity coefficients for vibration displacement and velocity expressing relative changes in levels of suitable peak factors for r.h. and l.h. rotation have been introduced basing on a definition:

$$\beta_{PL} = \frac{(Cz)P}{(Cz)L}, \qquad \gamma_{PL} = \frac{(Cv)P}{(Cv)L} \tag{2}$$

In case of correct cooperation of main kinematic elements, the change in sense of rotation of the system should not influence significantly values of peak directivity coefficients introduced. One can assume then that correct cooperation status of kinematic elements should correspond to limit cases.

$$\beta_{PL} \longrightarrow 1, \qquad \gamma_{PL} \longrightarrow 1 \tag{3}$$

Values of deviation from correct dynamical cooperation status for the system in question are represented by expressions

$$\Delta\beta_{PL} = \beta_{PL}^{-1} \tag{4}$$

$$\Delta\gamma_{PL} = \gamma_{PL}^{-1} \tag{5}$$

Then, $(\Delta\beta_{PL}, \Delta\gamma_{PL}) > 0$ determines a certain class of positive errors, while $(\Delta\beta_{PL}, \Delta\gamma_{PL}) < 0$ is a negative error class.

The class of positive errors reflects the predominance of vibrations conditioned by r.h. rotation of the system while the negative error class characterizes predominance of vibrations accompanying l.h. rotation of the system.

4. CHARACTERISTICS OF KINEMATIC SYSTEM OF ARM-MOUNTED CUTTER-HEAD KGS-320

Coal shearers as main coal winning machines are manufactured in various models characterized by varying ranges of cutting height, installed power, traction power and travelling speed determining the output of the shearers. KWB type shearers are provided with electric drive motors mounted in the body of the machine; one of the motors drives the winning unit and travelling unit while the second motor drives the winning unit only. Such a system of motor loading generates disadvantageous power distribution between the winning unit and the trawelling unit and this affects the travelling speed and shearer output [7]. To overcome this disadvantages shearer of KGS type has been designed in KOMAG where cutting drum drive motors are situated in arms while 60 kW drive motor for the travelling mechanism is built in the shearer body.
The shearer comprises two cutter-heads one mounted for r.h. jud and the second one for l.h. jud.

A kinematic diagram of KGS-320 shearer cutter-head together with specification of shafts (II-VII), gears (Z_1 - Z_{11}) and rolling bearings (1-15) is shown in fig. 3.

Transmission of power from the electric motor to the winning unit is realized via a kinematic system terminated with a planetary gear. A shearer arm is made as a uniform chamber where combined splash and oil bath lubrication is adopted for gears and bearings.

The choice of measuring points suitably distributed on a cutter-head frome is critical for vibro-acoustic measurements. Because in machines having rotative elements the highest excess of dynamic forces is transmitted by bearing units and kinematic transmissions, the measuring points chould be located at the close proximity of supporting bearings and gears.

Measurements has included direct recording of rms and peak values of amplitudes for vibration displacement and velocity during idle running for both senses of rotation of the output shaft.

For dynamic evaluation of main kinematic elements of the cutter-head the discriminants have been used reflecting deviations of dimensionless peak directivity coefficients for vibration displacement and velocity from a level assumed as a zero level. Measurements have been made in actual conditions at commisioning of manufactured prototype cutter heads. The block diagram of a measuring system is described in details in previous papers [8, 9].

5. ANALYSIS OF MEASUREMENT RESULTS

Measurement and analysis have been made for defined directional point discriminants obtained in form of a quotient of peak factor values of vibration displacement and velocity for both r.h. rotation and l.h. rotation of the system. The analysis of the results has enabled to find points on the frame characterized by occurrence of maximum shock vibrations; by relative comparision of the results obtained, the overall dynamic state of the cutter-heads under test could be approximately evaluated.

Taking into account that the total dynamic state of heads is determined by so called vibration intensity and that frequency of rotation for main elements is relatively small, one can assume that, approximately, this intensity, as per requirements of ISO [10], should be:

1°) proportional to vibration displacement amplitude for vibrations resulted from plays or from nonaxiallity of driving shafts alone

2°) proportional to vibration velocity amplitude for vibrations resulted from inaccuracy in gear meshing.

It can be assumed thus that a dimensionless peak directivity coefficient of vibration displacement is a measure characterizing a dynamic state of amin and intermediate shafts while a dimensionless peak directivity coefficient of vibration velocity can be taken as a measure of dynamic state of meshing for all gears of the cutter-head; this seems to support previous conclusions of the author [11, 12]. Deviations of values obtained for these coefficients from a dynamic state assumed by convention as a zero state, determine the total degree of dynamic diversification of specified kinematic elements.

To illustrate the distribution of deviation values obtained for peak directivity coefficient for vibration displacement in all cutter-heads under test, a graph has been given in fig. 4 representing these changes versus an order number N of the cutter-head at specified measurement points $n = 5, 6, 7, 8$ characterized by various degree of vibro-acoustic sensitivity. Maximum deviations of this coefficient has been observed for the head $N = 30$ at measurement points $n = 5, 6, 7$ that can reflect a worsened technical state when comparing with other cutter-heads.

Fig. 5 contains an exemplary graph of changes in deviations of the peak directivity coefficient for vibration displacement in particular measurement points n, for the cutter-heads characterized by maximum (heads $N = 22, 30$) and minimum changes ($N = 27, 28$) of this coefficient.

On the right side, the values of this coefficient averaged in relation to all measurement points are shown in form of rectangulers. Maximum deviation values of peak direstivity coefficient for vibration displacement are observed in measuring points $n = 5, 7$ reflecting extremal differentiation corresponding to predominance of vibrations accompanying r.h.

rotation (n = 5) for positive values and to predominance of vibrations accompanying l.h. rotation for negative values. In the same manner, deviation distribution for peak directivity coefficient of vibration velocity is shown at maximum sensitivity points n = 1, 2, 3, 4 for all cutter-heads under test. The graph of this distribution is shown in fig. 6. When analyzing this graph, it is evident that cutter-heads N = 29, 30, 31 should be classified as systems characterized mainly by inefficient meshing of gears.

Fig. 7 shows deviation distribution for peak directivity coefficient values of velocity at particular measurement points on a cutter-head frame for cutter-heads characterized by maximum (N = 30, 31) and minimum (N = 27, 32) changes of deviations for this coefficients. Diametrally different changes of this coefficient should be noted for cutter-heads N = 30, 31; for the cutter-head N = 30 almost all deviation values are positive while for the cutter-head N = 31 deviation values are negative that evidences, like previously, the domination of vibrations accompanying r.h. rotation in the first instance and the domination of vibrations accompanying l.h. rotation of the system in the second instance.

6. CONCLUSIONS

Dimensionless directional discriminants used have enabled to carry out overall relative evaluation of technical state of cutter-head main elements. Basing on deviation distribution for values of peak directivity coefficients of vibration displacement and velocity versus location of measurement point, the points characterized by maximum changes of deviation for these coefficient values comparing with zero reference values have been determined.

Directional estimates introduced have enabled detection of cutter-heads characterized by increased operational/technical tolerance conditioned mainly by dynamic state of main and intermediate shafts and correctness of gear meshing.

No determined classification criteria can be derived in this stage of study because of no-load condition of a cutter-head driving system when under test and a limited number of cutter-heads used in the study. These criteria when finally set would be used directly in vibro-acoustic diagnostic of cutter-heads. We would like emphasise in the last place, that results of vibro-acoustic diagnostic studies carried out for coal shearers KGS-320 give a substantial basis for conclusion that the method described will permit in near future the development of not only classification criteria (after suitable experimental conditions are provided) but also design directives (when used in parallel with simulated dynamic studies) to enable the manufacture of products characterized by dependability, durability and silent-running.

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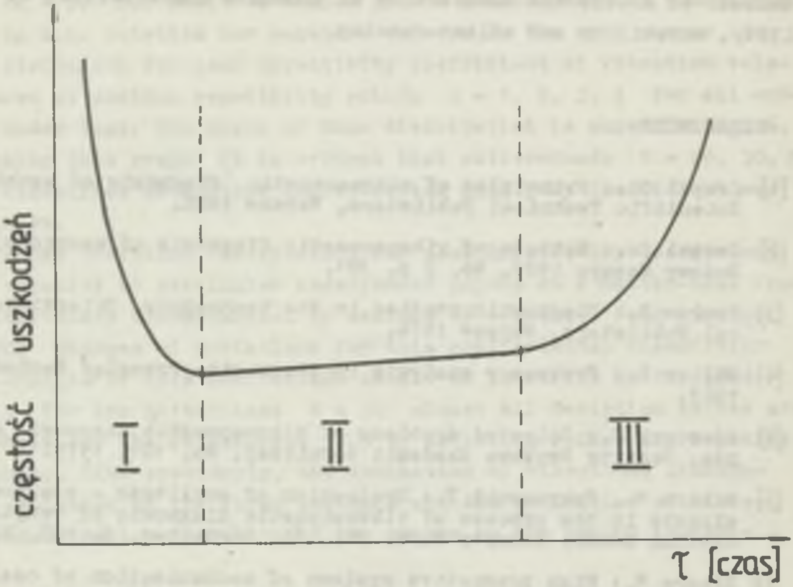

Fig. 1. Failure frequency distribution versus operation time
Rys. 1. Rozkład występowania częstości uszkodzeń w funkcji czasu eksploatacji

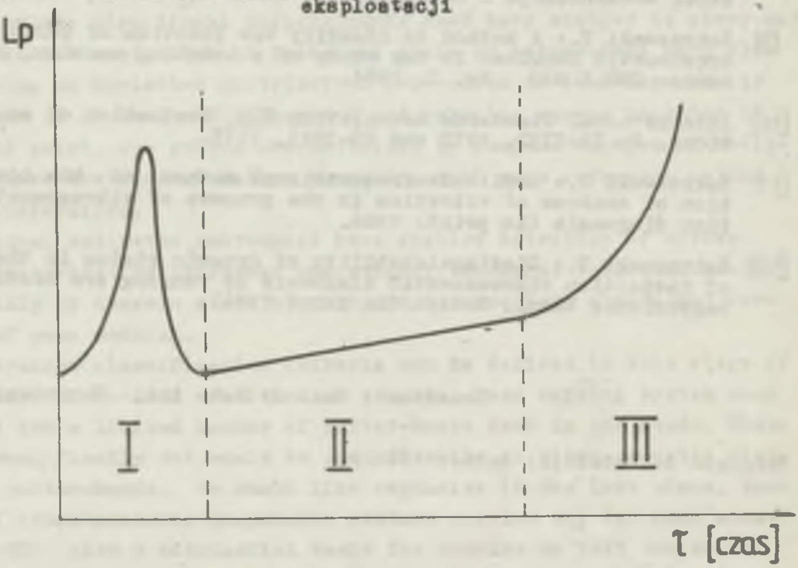

Fig. 2. Machine vibration level versus operation time; I - running-in stage, II - normal operation stage and III - breakdown vibration stage

Rys. 2. Poziom drgań maszyny w funkcji czasu eksploatacji z zaznaczeniem obszaru docierania I, obszaru normalnej eksploatacji II oraz obszaru drgań awaryjnych III

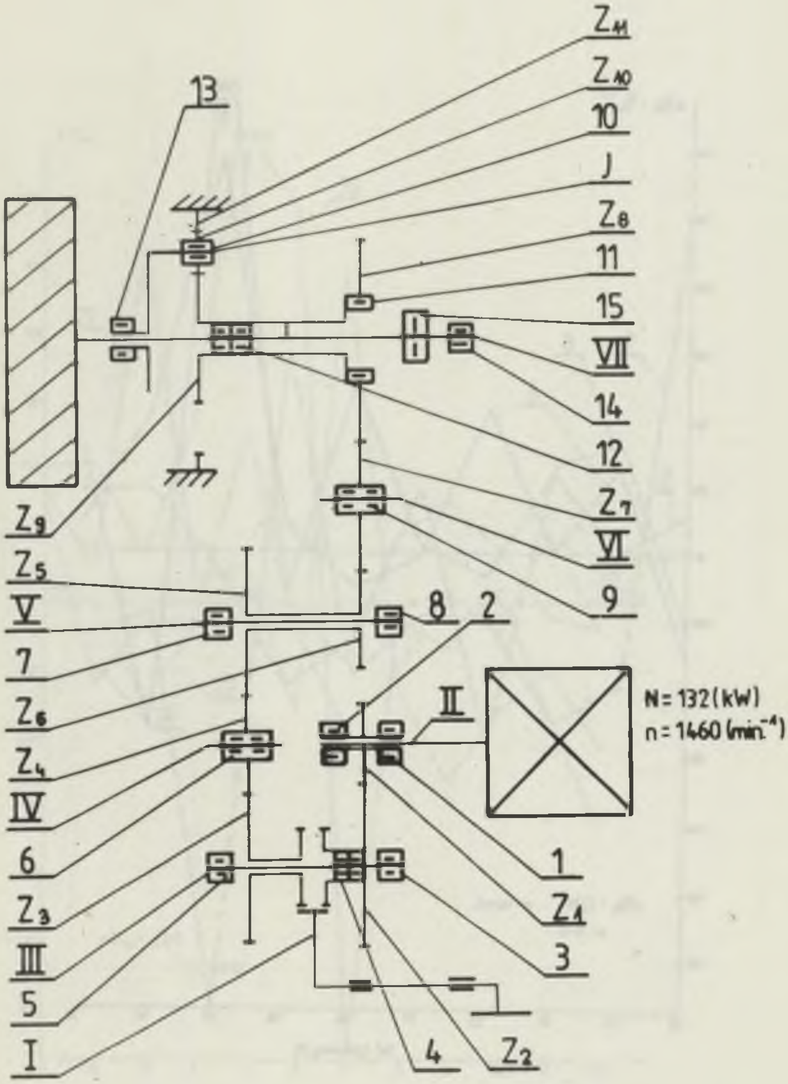

Fig. 3. Kinematic diagram of arm-mounted cutter-head of KGS-320 shearer
Rys. 3. Schemat kinematyczny głowicy ramieniowej kombajnu ścianowego KGS-320

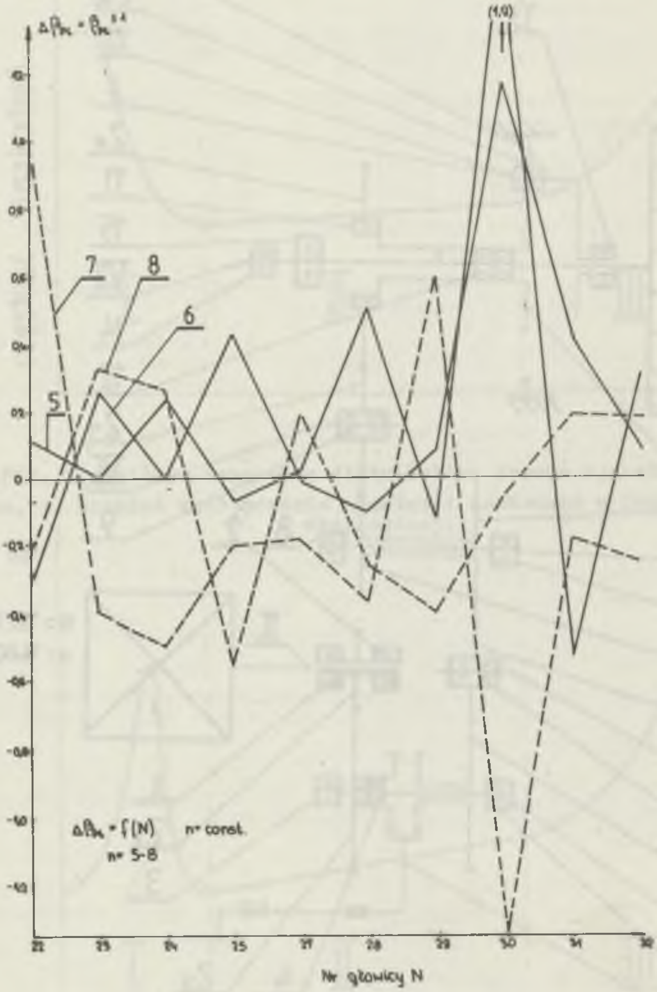

Fig. 4. Distribution of changes in peak directivity coefficient deviation value for vibration displacement versus order number N of cutter-head

Rys. 4. Rozkład zmian wartości odchyleń współczynnika kierunkowości szczytowej przemieszczenia drgań w funkcji numeru porządkowego N głowicy

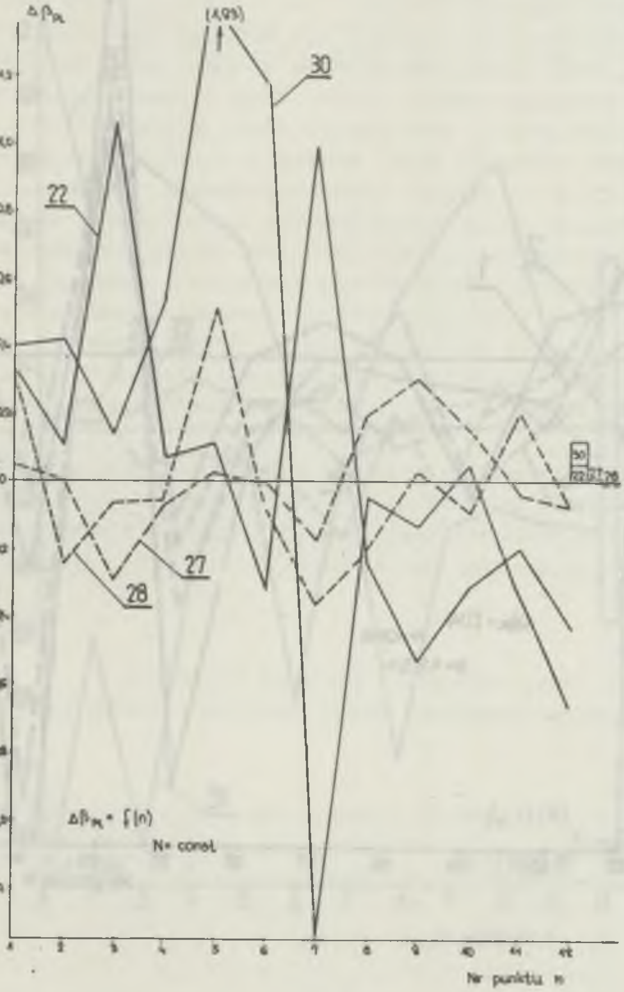

Fig. 5. Distribution of changes in peak directivity coefficient deviation value for vibration displacement versus location of measurement point n

Rys. 5. Rozkład zmian wartości odchyleń współczynnika kierunkowości szczytowej przemieszczenia drgań w funkcji położenia punktu pomiarowego n

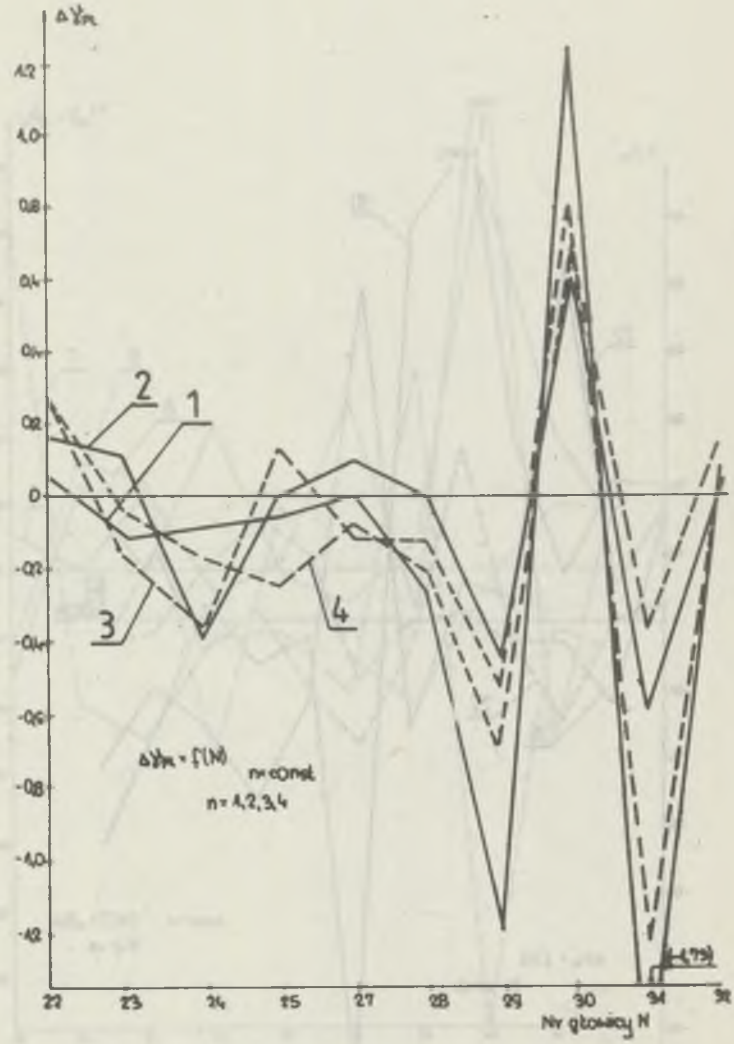

Fig. 6. Distribution of changes in peak directivity coefficient deviation value for vibration velocity versus order number N of cutter-head

Rys. 6. Rozkład zmian wartości odhyleń współczynnika kierunkowości szczytowej prędkości drgań w funkcji numeru porządkowego N głowicy

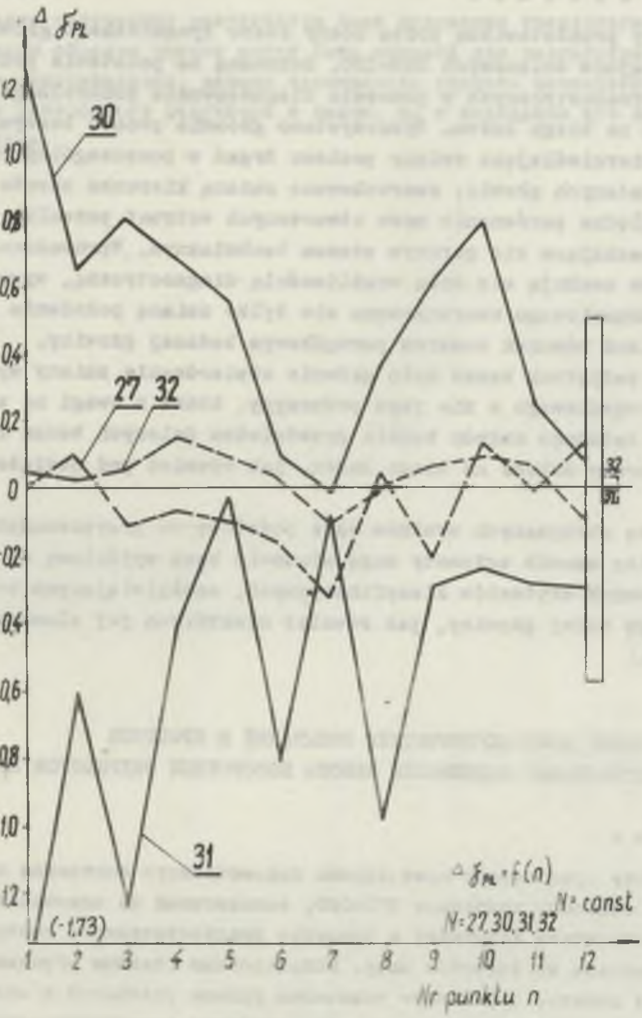

Fig. 7. Distribution of changes in peak directivity coefficient value for vibration velocity versus location of measurement point n

Rys. 7. Rozkład zmian wartości odchyleń współczynnika kierunkowości szczytowej prędkości drgań w funkcji położenia punktu pomiarowego

WYKORZYSTANIE BADAŃ WIBROAKUSTYCZNYCH W PROCESIE DOSKONALENIA NIEZAWODNOŚCI DZIAŁANIA GŁOWIC RAMIENIOWYCH KGS-320

Streszczenie

W pracy przedstawiono próbę oceny stanu dynamicznego głowic ramieniowych kombajnów ścianowych KGS-320, dokonaną na podstawie przeprowadzonych badań wibroakustycznych w procesie diagnozowania kontrolnego podczas pracy układu na biegu luzem. Wykorzystano głównie proste bezwymiarowe estymaty odzwierciedlające zmiany poziomu drgań w poszczególnych punktach korpusu badanych głowic, uwarunkowane zmianą kierunku obrotu wału wyjściowego. Względne porównanie nowo utworzonych estymat pozwolił wyodrębnić głowice cechujące się gorszym stanem technicznym. Wprowadzone estymaty kierunkowe cechują się dużą wrażliwością diagnostyczną, wyrażając zmiany sygnału drganiowego uwarunkowane nie tylko zmianą położenia punktu pomiarowego, lecz również numerem porządkowym badanej głowicy.

Celem podjętych badań było głównie stwierdzenie zmiany występowania sygnału drganiowego a nie jego przyczyny, która z uwagi na złożoność kinematyczną badanego układu będzie przedmiotem dalszych badań zarówno w warunkach pracy układu na biegu luzem, jak również pod obciążeniem nominalnym.

Analiza otrzymanych wyników daje podstawę do przypuszczenia, że wprowadzone w ten sposób estymaty mogą stanowić bazę wyjściową do opracowania szczegółowych kryteriów klasyfikacyjnych, umożliwiających ocenę stanu jakościowego całej głowicy, jak również niektórych jej elementów kinematycznych.

ИСПОЛЬЗОВАНИЕ ВИБРОАКУСТИЧЕСКИХ ИСПЫТАНИЙ В ПРОЦЕССЕ УСОВЕРШЕНСТВОВАНИЯ НАДЕЖНОСТИ РАБОТЫ ПОВОРОТНЫХ РЕДУКТОРОВ КГС-320

Резюме

В работе представлен опыт оценки динамического состояния поворотных редукторов очистных комбайнов КГС-320, выполненный на основании проведенных виброакустических испытаний в процессе диагностического контроля во время работы системы на холостом ходу. Использованы главным образом простые безразмерные оценки, отражающие изменения уровня колебаний в отдельных пунктах корпуса исследуемых редукторов, обусловленные изменением направления оборота ведомого вала. Относительное сравнение полученных новых оценок позволило выделить редукторы, характеризующиеся худшим техническим состоянием.

Введенные оценки направления характеризуются большой диагностической восприимчивостью, выражая изменения сигнала колебания, обусловленного не только изменением положения точки замера, но и порядковым номером исследуемого поворотного редуктора.

Целью предпринятых испытаний была, главным образом, констатация изменения появления сигнала колебания, а не его причина (которая из-за кинематической сложности исследуемой системы будет предметом дальнейших исследований) в условиях работы системы как на холостом ходу, так и при номинальной нагрузке.

Анализ полученных результатов дает основание предполагать, что введенные таким образом оценки могут быть основой для разработки подробных критериев классификации, дающих возможность оценить качественное состояние не только поворотного редуктора в целом, но и отдельных его кинематических элементов.