ARCHIVES OF ACOUSTICS
12, 2, 89–111 (1987)

SPECTRAL ANALYSIS OF VIBRATIONS IN CONTROL INVESTIGATIONS OF VIBROACOUSTIC HEADS KGS-320

TADEUSZ ZAKRZEWSKI

Mechanization Center of the Mining Industry KOMAG
(44-100 Gliwice, ul. Pszczyńska 37, Poland)

Diagnostic investigations were performed in order to evaluate the usability of the spectral analysis of vibrations in the process of control diagnostics of vibroacoustic arm heads of combined cutter loaders *KGS* 320. Tests were carried out at the acceptance inspection stand during idle running of the head with the consideration of both directions of rotation of the output shaft.

Frequency components corresponding to rotational speeds of some kinematic elements of the system under investigation were isolated on the basis of the spectral analysis of vibrations. A comparison of discrete amplitudes obtained from vibration spectra, allowed to evaluate the range of variability of vibration levels in determined frequency bands, as well as to isolate some kinematic pairs, which are characterised by the maximal vibration intensity. It was found that the rotation direction of the output shaft influences values of amplitudes of some vibration parameters in frequency bands, which contain characteristic frequencies of some elements of the system.

1. Introduction

The question of providing a longlasting reliability of technical objects produced by many branches of industry is one of the important problems of production enterprises. Progress in the domain of construction and technology of production and control of operation processes of determinate machine elements has led to the formation of many various, frequently complex, technical objects. The practical usability of such objects is determined by the type and occurance frequency of failures, and by the repair time. Therefore, in order to improve the quality of technical objects, their dynamic properties have to be investigated during control diagnostics. The physics of processes leading to failures constitute the basis of a scientifically-founded choice of the most effective construction and technologic method, which would increase or the life of basic machine elements [10]. In view of rapid development of technology and high requirements for machinery, concerning their reliability, production precision, life, etc. technical diagnostics should enable to determine the condition of a machine on the basis of its certain parameters, without disassembly or interruption of the technologic cycle. Hence, indirect methods, in which the condition evaluation is performed on the basis of investigations of processes accompanying machine functioning, have to be applied [4]. Such methods are utilized in vibroacoustic diagnostics which infer about the dynamic state of an object from investigations of the vibroacoustic process generated in various points of the body of an investigated object [12, 5]. Vibroacoustic phenomena correspond to most significant physical processes, which take place in the machine and which determine correctness of functioning, such as for example: strain, stress, cooperation of elements etc. Various estimators of a deterministic character, which describe these phenomena, and the dynamic state of dynamic pairs of the machine can be determined by averaging the process in the domain of time, frequency or amplitude.

Performed vibroacoustic investigations were aimed at the relative evaluation of the dynamic state of arm heads in mining combined cutter loaders. The evaluation was carried out on the basis of results of a vibration spectral analysis, which included $n = 10$ heads during idle running with the consideration of both rotation directions of the getter.

2. Applied measuring system

The choice of the research method, which would enable useful signal separation from disturbing signals, is the main problem of vibroacoustic diagnostics. At present, three fundamental separation methods are applied [8]. They concern the domains of: space, time and frequency. The method of signal separation in the domain of time is based on time gating (sampling) of signals with respect to the dynamics of the material system and on obtaining information contained in time intervals of the gate opening. The frequency signal separation method consists in the transformation of the time function into the domain of frequency and then in the analysis of this function in adequate diagnostic bands. The choice of an analysis method depends on the type of machine under investigation and is based on an assumption that the useful signal can be found in certain frequency ranges, while outside these ranges disturbing signals or signals carrying little information, prevail.

At present, the spectral analysis method is measuring method most frequently applied in machine diagnostics [14, 16]. Due to the randomness of vibroacoustic processes related to machine work, the diagnosis of the state of an object is based on signal estimators achieved through analogue or digital signal processing. Analogue signal processing finds wide application in practice, especially when the dynamic state of a machine can be determined from the measurement and analysis of only few estimators of the vibroacoustic signal [15].

Digital processing allows quicker and more complex investigation and analysis of a signal, according to a previously programmed algorithm of digital processing. It should find wider application in vibroacoustic diagnostics of machines in the nearest future.

In order to estimate the dynamic state of arm heads two series of measurements were carried out, and rms and peak values of vibration parameters were recorded respectively for the right and left rotation of the output shaft. Including the assumption that the dynamic state of heads is conditioned by the so-called vibration rate and that the frequency of the primary motion of main drive elements is low (below 50 Hz), it can be approximately accepted that this rate should be:

1) proportional to the displacement amplitude of vibrations, conditioned by the play misalignment of drive shafts,

2) proportional to the velocity amplitude of vibrations, caused by incorrect mating of toothed wheels,

3) proportional to the acceleration amplitude of vibrations, due to manipulative and assembly errors of rolling bearings.

Vibration parameters were measured with a piezoelectric sensor and then the voltage from the transducer was recorded on magnetic tape. The block diagram of the system for direct registration of vibration parameters is presented in Fig. 1. The registered signal was filtered in order to reduce the influence of disturbances with high and very low frequencies. The block diagram of the laboratory system for amplitude – frequency analysis is shown in Fig. 2. Rms and peak values of measured parameters were read off directly from the vibrometer during investigations.

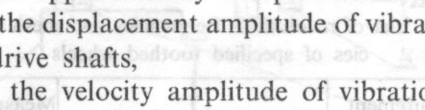


Fig. 1. Block diagram of the system for measurements of vibration parameters: 1) *ACC/U* transducer type 4370, 2) vibrometer type 2511, 3) magnetic recorder "Tandberg" type 115D

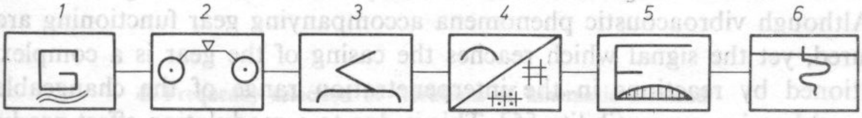


Fig. 2. Block diagram of the system for frequency analysis of vibration parameters: 1) beat frequency generator type 1022 (path calibration), 2) magnetic recorder "Tandberg" type 115*D*, 3) voltage amplifier type 2606, 4) digital recorder type 7502, 5) third-octave analyser type 2113 or heterodyne analyser type 2010, 6) level recorder type 2305

3. Localization of measuring points

The localization of measuring points on the body of a head is a very important problem in vibroacoustic measurements. As the greatest surpluses of dynamic forces in machines with rotational elements are transfered by bearings and transmissions of kinematic elements, thus measuring points were located in the direct nearness of bearings of main drive units of the head, such as idle wheels, main shafts, layshafts, planetary gear. Tab. 1 presents the distribution diagram of measuring points adjoining mentioned kinematic elements of the system. In principle, such measurements should be done in three perpendicular directions, but in our case (head with mainly rotational elements) measurements were limited to the radial and axial direction, what is absolutely sufficient [3].

Table 1. List of revolutions frequencies and mesh frequencies of specified toothed wheels

Measurement point number	Gear wheel	Bearing	Measurement directions
1	Z3	5	radial
2	Z1	—	radial
3	Z2	4; 3	radial
4	Z4	6	radial
5	Z4	6	axial
6	Z5	7	radial
7	Z4; Z5	7	axial
8	Z10; Z11	13	axial
9	Z8	11	radial
10	Z9; Z10; Z11	10	radial
11	Z7	9	axial
12	—	14; 15	radial

A head with an adjustable arm is a toothed gear consisting of several cylindrical gears and a planetery gear, which drives directly the output shaft of the mining organ, as the last transmission. The mining organ is supplied with power from an electric motor according to the kinematic system shown in Fig. 3.

Although vibroacoustic phenomena accompanying gear functioning are easily measured, yet the signal which reaches the casing of the gear is a complex signal conditioned by reactions in the interpenetration range of the changeable (with rotations) bearing susceptibility [5]. This is due to a modulation effect produced by the series connection of dynamic reactions in the range of interpenetration of dynamic reactions of ball bearings. Furthermore, the casing of the gear has a very complex resonance structure and thus various filtration effects will occur in various signal reception points on the casing and a modified signal corresponding to gear functioning will be obtained.

The evaluation of the technologic state of a gear and its elements with the vibroacoustic method is difficult, because of complicated paths of reactions from the range of interpenetration to individual vibration reception points on the casing of the gear.

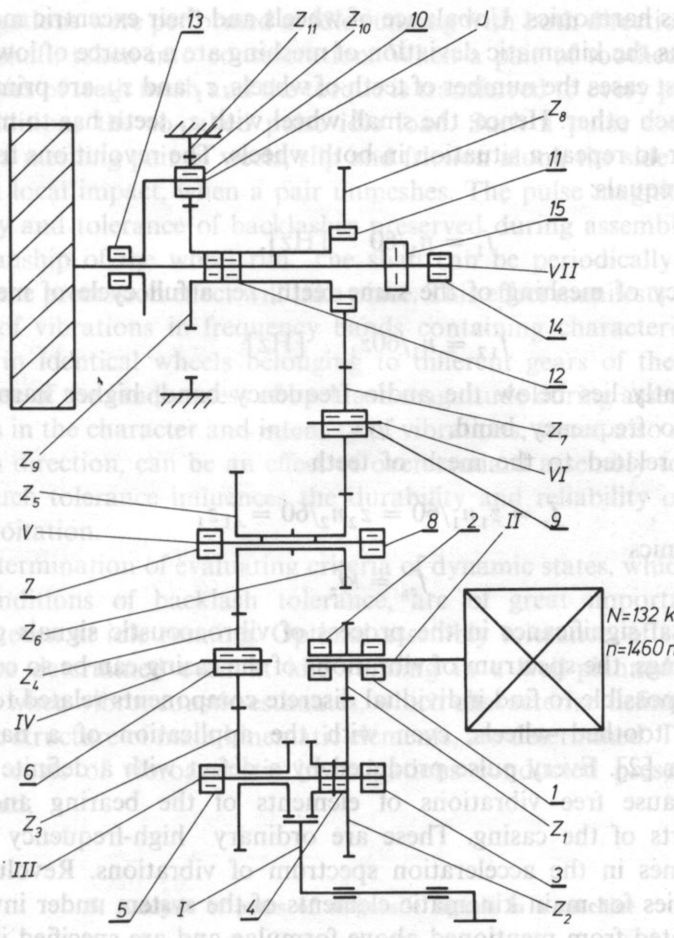


Fig. 3. Kinematic diagram of a head with a specification of shafts (I–VII), toothed wheels ($Z_1 - Z_{11}$) and bearings (1–15)

4. Frequency selection of vibrations of kinematic elements

Toothed gears, ball or barrel bearings are fundamental elements of most machines. In order to compare adequate components of the spectrum of vibrations, frequencies of forced vibrations of definite kinematic elements of the head, such as main shafts, layshafts and toothed wheels, have to be determined.

Every element of a toothed gear is an elastic body and can be excited to vibrate and generates a signal, which has a spectrum of a rather wide frequency range. The level of vibrations is especially high in certain frequency bands of the spectrum, especially for frequencies which are an integral multiple of the excitation frequency [11]. This means that the highest power in toothed wheels is emitted at mesh frequencies and its harmonics. Unbalance of wheels and their excentric mounting on the shaft as well as the kinematic deviation of meshing are a source of low frequency vibrations. In most cases the number of teeth of wheels, z_1 and z_2, are prime numbers with respect to each other. Hence, the small wheel with z_1 teeth has to make z_2 full rotations in order to repeat a situation in both wheels. The revolutions frequency of the small wheel equals:

$$f_1 = n_1/60 \quad [\text{Hz}]. \tag{1}$$

The frequency of meshing of the same teeth, i.e. a full cycle of mesh, equals

$$f_{12} = n_1/60z_1 \quad [\text{Hz}] \tag{2}$$

and most frequently lies below the audio frequency band; higher harmonics can overlap the audio frequency band.

Frequencies related to the mesh of teeth

$$f_z = z_1 n_1/60 = z_2 n_2/60 = f_1 z_1 \tag{3}$$

and their harmonics

$$f_{z_k} = kf_z \tag{3a}$$

are of fundamental significance in the process of vibroacoustic signals generation.

Due to bearings, the spectrum of vibrations of the casing can be so complicated that it may be impossible to find individual discrete components related to rotations and meshing of toothed wheels, even with the application of a narrow-band spectrum analyser [2]. Every pulse produced by a defect with a definite excitation frequency will cause free vibrations of elements of the bearing and then of neighbouring parts of the casing. These are ordinary high-frequency vibrations observed sometimes in the acceleration spectrum of vibrations. Revolutions and meshing frequencies for main kinematic elements of the system under investigation have been calculated from mentioned above formulae and are specified in Table 2.

Table 2. Measuring points assigned to specified kinematic elements

Rotation frequency $[s^{-1}]$ [Hz]	Meshing frequency $[s^{-1}]$ [Hz]
$f_{II} = 24.3$	$F_{Z1} = 464.1$
$f_{III} = 9.06$	$F_{Z2} = 464.1$
$f_{IV} = 6.5$	$F_{Z2'} = 309.4$
$f_V = 4.64$	$F_{Z3} = 182$
$f_{VI} = 2.82$	$F_{Z4} = 182$
$f_{VII} = 0.64$	$F_{Z5} = 182$
	$F_{Z6} = 78.9$
	$F_{Z7} = 78.9$
	$F_{Z8} = 78.9$
	$F_{Z9} = 62.1$
	$F_{Z10} = 30.9$

Investigations were performed at idle running with both directions of rotation of the output shaft, taken into consideration. When a pair of toothed wheels rotates, following pairs of teeth mesh and the torque is transferred to every pair of teeth. This process conditions the so-called pulse idle load. Such a pulse consists of a local collision of a meshing pair of weels, slip and friction along the side contact parts of teeth, and a local impact, when a pair unmeshes. The pulse magnitude depends on the accuracy and tolerance of backlashes preserved during assembly, as well as on the workmanship of the wheel rim. The shaft can be periodically forces by these factors and the precession effect will take place. This effect manifests in the amplitude magnitude of vibrations in frequency bands containing characteristic frequencies. Backlashes in identical wheels belonging to different gears of the same type will differ, because of the randomness of backlash magnitude during assembly. Significant divergencies in the character and intensity of vibrations, stated also after a change of the rotation direction, can be an effect of overstandard assembly tolerance of head elements. Such tolerance influences the durability and reliability of such a system during exploitation.

The determination of evaluating criteria of dynamic states, which correspond to optimal conditions of backlash tolerance, are of great importance in control diagnostics, even at idle running. Optimal assembly tolerance, of some elements at least, can be determined even at idle running in a well-planned and performed experiment, when vibroacoustic estimates, which characterize definite parameters of the dynamic structure of main kinematic elements, are determined. This will facilitate and reduce costs of vibroacoustic investigations conducted presently at variable dynamic load.

5. Analysis of paths of diagnostic signals in a machine

A signal undergoes distorsions and disturbances due to other sources of vibrations on its path from the source to the transducer, in the dynamic system of a machine. The received signal often is a superposition of all generated signals, because of complex paths and size of signals which reach the receiving point on the machine body. It depends on the transmittance structure of the system under investigation. Therefore, the separation of characteristic frequencies in the spectrum of vibrations is a very complex problem. In order to present the complexity of this problem the influence of transmittance on power spectral density functions are analysed in a simple line system during the change of the rotation direction of the output shaft.

Many papers on machine diagnostics neglect a very important problem of the path of a signal $x(t)$ from the source of vibrations to the point of signal reception by a transducer mounted most frequently on the machine body. Instead of the real signal $x(t)$ we obtain signal $y(t)$ on the machine body. If we assume a line dynamic system (shown in Fig. 4), then the relationship between the power spectral density function of input $x(t)$ and output $y(t)$ signals is as follows

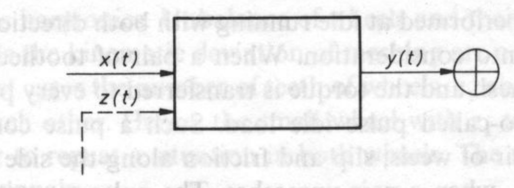


Fig. 4. Simplified diagram of a single input line system

$$G_y(f) = |H(f)|^2 G_x(f), \tag{4}$$

where: $G_x(f)$, $G_y(f)$ – one-sided power spectral density functions of signals $x(t)$ and $y(t)$, $H(f)$ – transmittance of the system, corresponding to the amplitude-phase characteristic of the path of the signal.

Having the power spectral density $G_x(f)$ of the input signal and the gain coefficient $H(f)$ of the system, we can determine the power spectral density $G_y(f)$ of the output signal.

Let us consider now a superposition of n line systems (Fig. 5) with constant parameters, with q strictly determined output signals $x_i(t)$, $i = 1, 2, 3, \ldots, q$, measu-

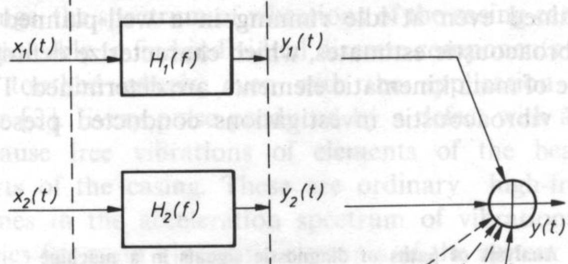


Fig. 5. Diagram of a multi input line system

red at the output through the global output signal $y(t)$. This signal will be a sum of n components of output signals

$$y(t) = \sum_{i=1} y_i(t). \tag{5}$$

Several typical transformations based on the Fourier transform lead to the following form of the expression for the power spectral density of the output signal:

$$G_y(f) = \sum_{i=1}^{q} \sum_{j=1}^{q} H_i^*(f) H_j(f) G_{ij}(f), \tag{6}$$

where: $G_{ij}(f)$ – mutual power spectral density of signals $x_i(t)$ and $x_j(t)$, $H^*(f)$ – complex function conjugated with function $H(f)$.

For uncorrelated processes we have:

$$G_y(f) = \sum_{i=1}^{q} |H_i(f)|^2 G_i(f). \tag{6a}$$

A change of the rotation direction of the output shaft should not lead to transmittance changes of the system. Hence, we can accept that the transmittance of point n on the head casing is constant for a right (P) and left (L) rotation

$$\{|H_i(f)|2\}_n^{P,L} = \text{const}. \tag{7}$$

In such a case power spectral densities of signals registered in a given point can be expressed as

$$G_y^P(f) = \sum_{i=1}^{q} |H_i(f)|^2 G_i^P(f), \quad G_y^L(f) = \sum_{i=1}^{q} |H_i(f)|^2 G_i^L(f). \tag{8}$$

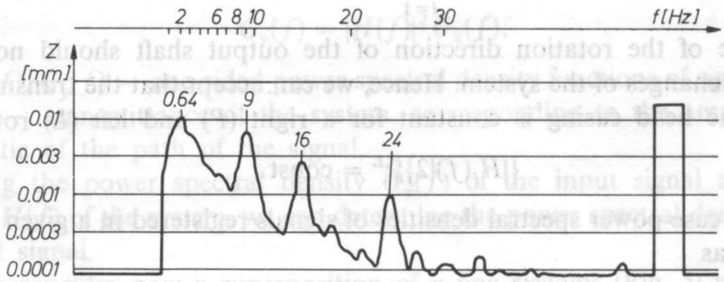
This means that the power spectral density in a given measuring point will not have the error of transmittance estimation when the direction of rotation is changed. Differences in the power spectral density structure occuring in various points after a change of the rotation direction should be conditioned mainly by the assymetry of precession, which takes place in motion dynamics of definite kinematic pairs.

6. Results and discussion

Such fundamental parameters of vibration as displacement, velocity and acceleration, have been measured and analysed. Their comparison in certain frequency intervals has lead to a classification of heads under investigation with respect to intensity of emitted vibration signals. The spectral analysis of vibration parameters, presented in the form of amplitude-frequency characteristics, enabled us to determine dominating frequencies and to assign them adequate revolutions frequencies of certain kinematic elements of the investigated system.

6.1. *Line spectrum of vibrations displacement*

A narrow-band analysis in the frequency range 0–50 Hz has been performed at both rotation directions of the output shaft, in order to determine amplitude-frequency distribution of vibrations displacement in individual measuring points on heads under investigation. Displacement spectra of vibrations for right rotations, recorded in point $n = 7$ of casings of heads numbered $N = 27$ and $N = 28$, respectively, are shown in Fig. 6. These heads have significantly diversified displacement amplitudes in certain frequency bands, especially in the sub-audio range. Rms values of amplitudes were determined from obtained spectrograms of displacement of vibrations. Considerations were limited to a frequency range 0–30 Hz due to a lack of sufficient amplitude resolution for frequencies exceeding 30 Hz. Figs. 7 and 8 present graphically the displacement spectra of



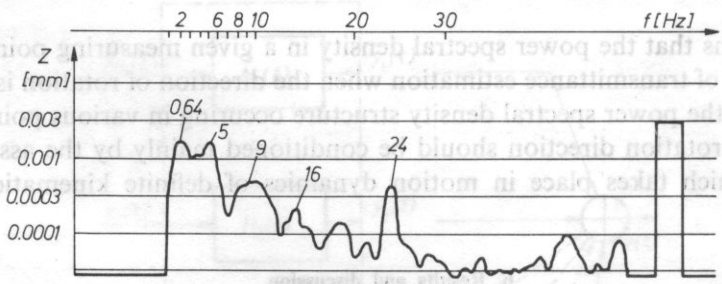


Fig. 6. Line spectra of vibrations displacement: a) Head no 27, point 7, right rotation of the output shaft, b) Head no 28, point 7, right rotation of the output shaft

vibrations for heads $N = 27$, 28, determined respectively for points $n = 2$ and $n = 12$, for a right (P) and left (L) rotation of the system. A considerable amplitude-frequency variability has been observed in registered spectra of displacements in most measuring points of the casing. It results from the spectral analysis, presented in the form of a frequency distribution of amplitudes for all measuring points, that displacement amplitude maxima of vibrations occur mainly in bands with mid-band frequencies: $f = 1$, 2, 4, 6, 9, 16 and 25 Hz.

A comparison of calculated characteristic frequencies corresponding to rotational speeds of shafts and certain toothed wheels (Table 1), with the graphic frequency distribution of displacement of vibrations proves that frequencies dominating in the frequency spectrum determine bands which contain calculated characteristic frequencies. Global changes of displacement of vibrations averaged with respect to all

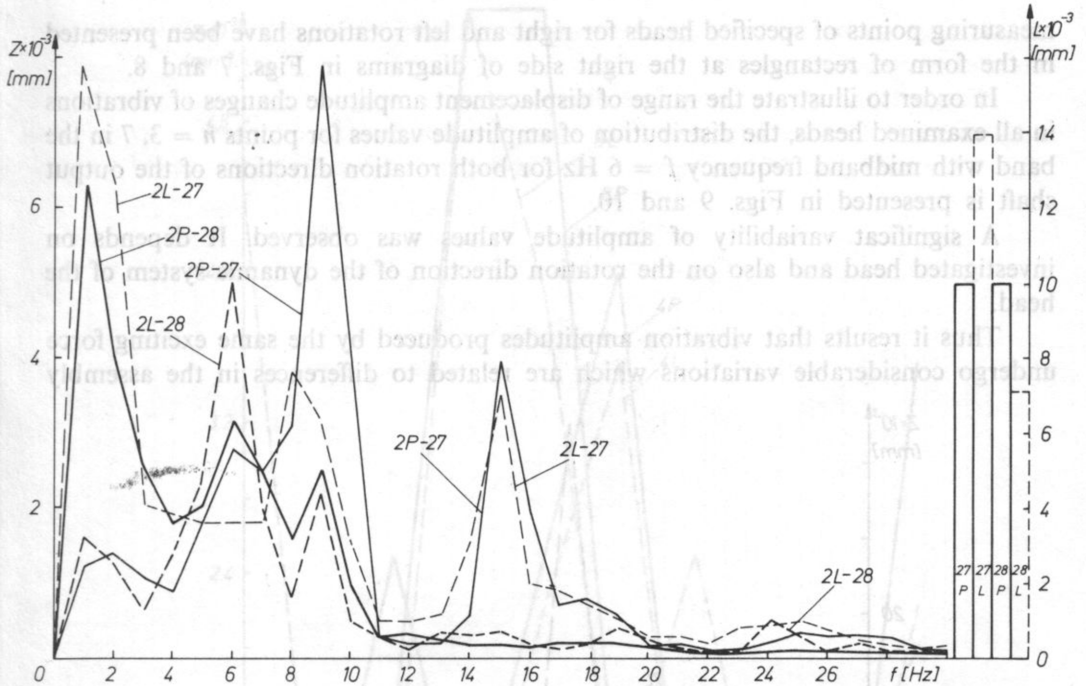


Fig. 7. Frequency analysis of vibration displacement for heads $N = 27$ and 28 in measuring point 2; ——— – right rotation; – — – — – left rotation

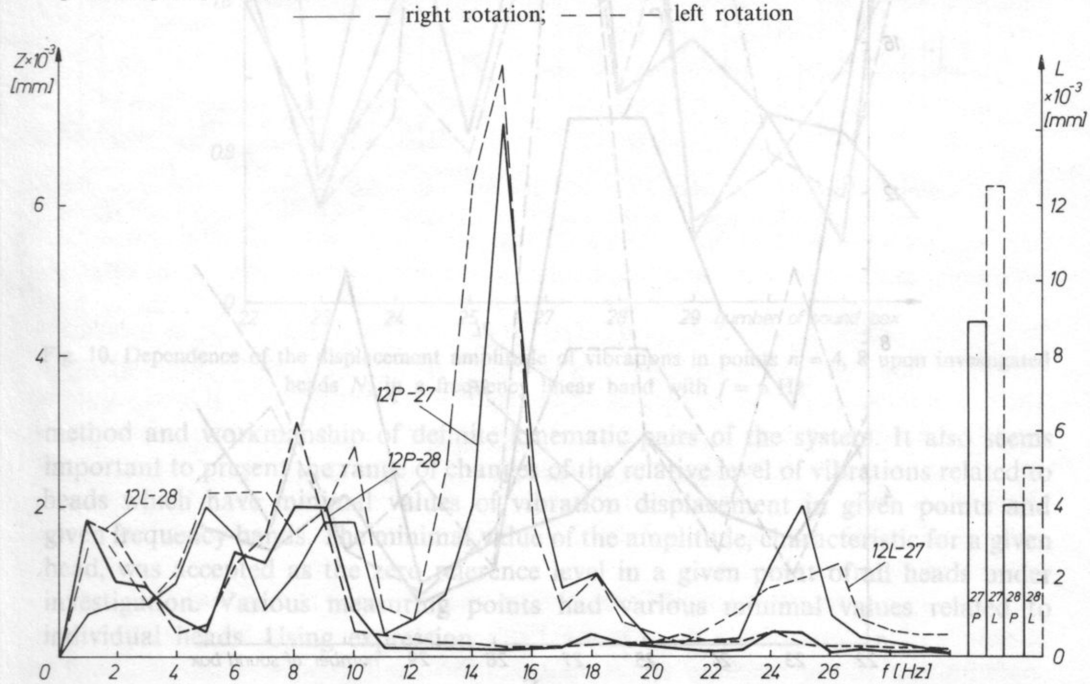


Fig. 8. Frequency analysis of vibrations displacement for heads $N = 27$ and 28 in point $n = 12$ for both directions of rotation (P – right rotations, L – left rotations)

measuring points of specified heads for right and left rotations have been presented in the form of rectangles at the right side of diagrams in Figs. 7 and 8.

In order to illustrate the range of displacement amplitude changes of vibrations in all examined heads, the distribution of amplitude values for points $n = 3, 7$ in the band with midband frequency $f = 6$ Hz for both rotation directions of the output shaft is presented in Figs. 9 and 10.

A significat variability of amplitude values was observed. It depends on investigated head and also on the rotation direction of the dynamic system of the head.

Thus it results that vibration amplitudes produced by the same exciting force undergo considerable variations which are related to differences in the assembly

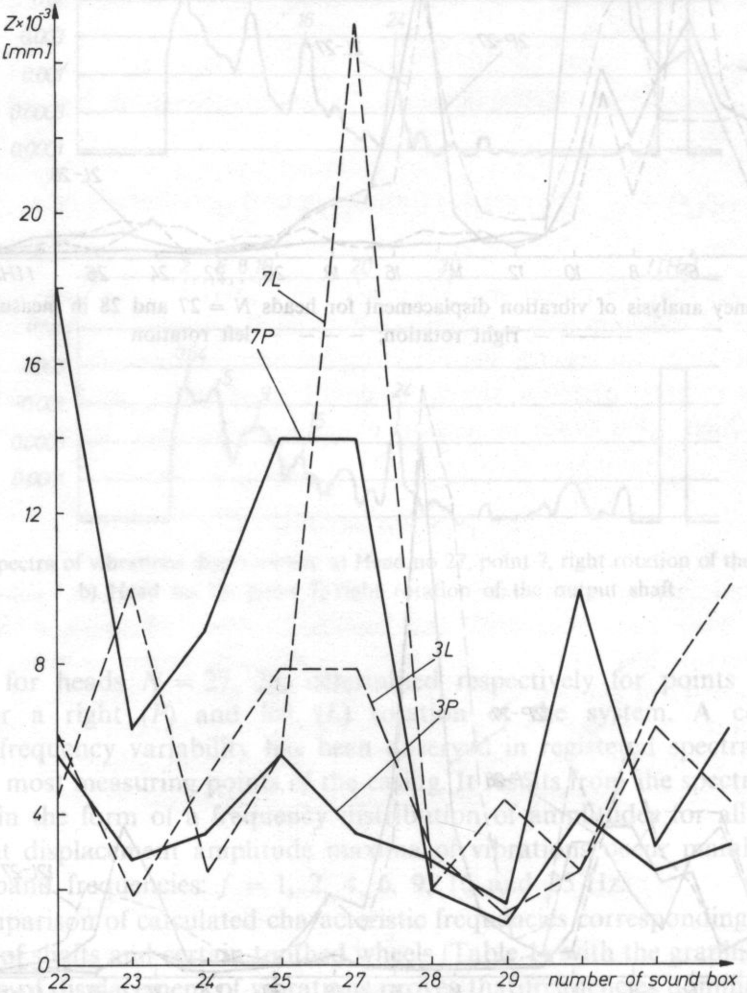


Fig. 9. Dependence of the displacement amplitude of vibrations in points $n = 3, 7$ upon investigated heads N, in a frequency linear band with $f = 2$ Hz

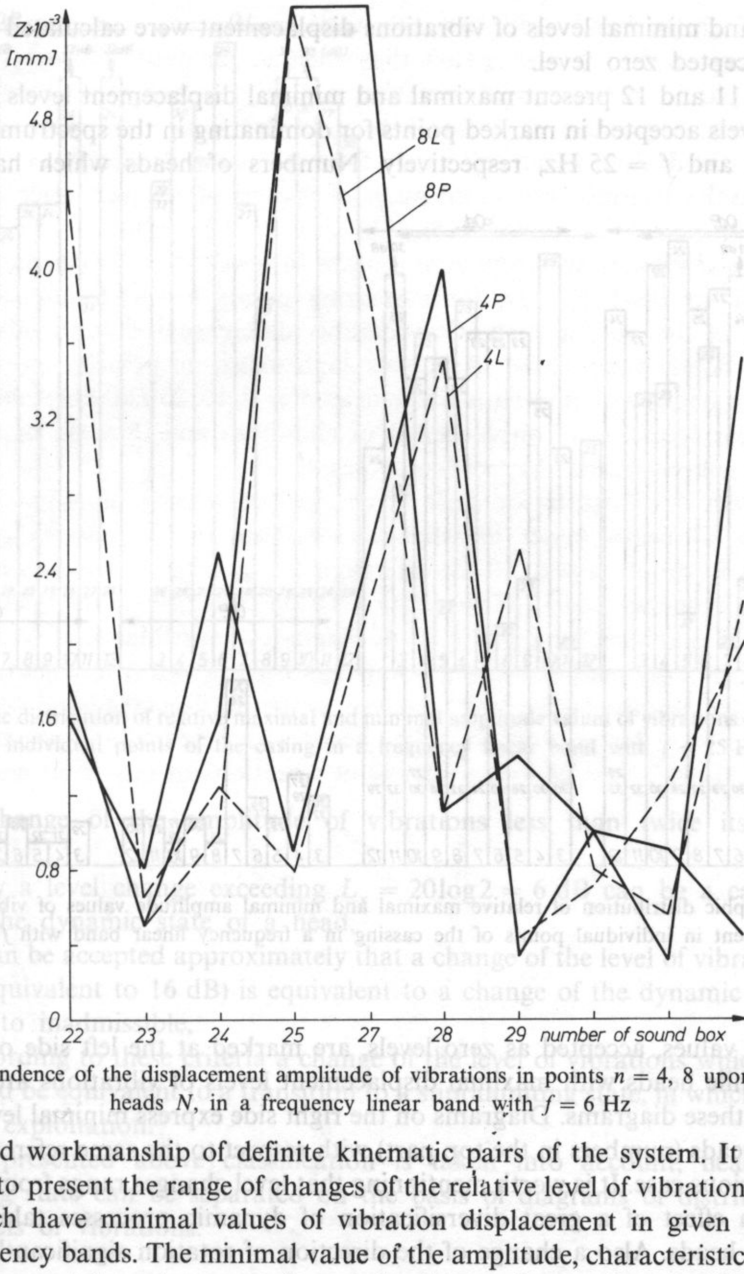


Fig. 10. Dependence of the displacement amplitude of vibrations in points $n = 4, 8$ upon investigated heads N, in a frequency linear band with $f = 6$ Hz

method and workmanship of definite kinematic pairs of the system. It also seems important to present the range of changes of the relative level of vibrations related to heads which have minimal values of vibration displacement in given points and given frequency bands. The minimal value of the amplitude, characteristic for a given head, was accepted as the zero reference level in a given point of all heads under investigation. Various measuring points had various minimal values related to individual heads. Using expression

$$L_z = 20\log\frac{Z_{sk(\max)}}{Z_{sk(\min)}} \quad [\text{dB}],$$

maximal and minimal levels of vibrations displacement were calculated with respect to the accepted zero level.

Figs. 11 and 12 present maximal and minimal displacement levels with respect to zero levels accepted in marked points for dominating in the spectrum frequencies: $f = 9$ Hz and $f = 25$ Hz, respectively. Numbers of heads which have minimal

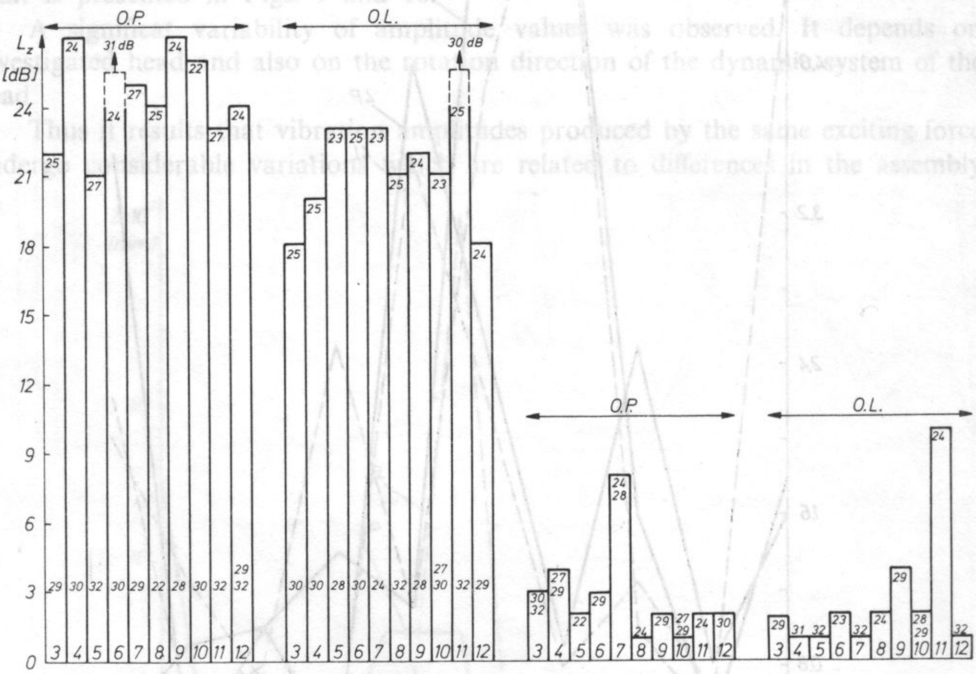


Fig. 11. Graphic distribution of relative maximal and minimal amplitude values of vibrations displacement in individual points of the cassing in a frequency linear band with $f = 9$ Hz

amplitude values, accepted as zero levels, are marked at the left side of measuring points. While heads with maximal displacement levels of vibrations are marked at the top of these diagrams. Diagrams on the right side express minimal level values of specified heads (numbers in the top part) with respect to the same reference levels as in the previous case. It is worth mentioning that level changes range from 0 to 33 dB. This is an effect of a great diversification of dynamic processes taking place in individual heads. Also a change of the direction of rotation significantly influences generated dynamic processes.

A very important conclusion results from criteria stated in standard *ISO* 3945 – a change of the level of vibrations in a 6:1 ratio is always a cause for a change of the classification of the machine vibration condition from good to inadmissible. The following classification of the dynamic state can be accepted on the basis of this standard:

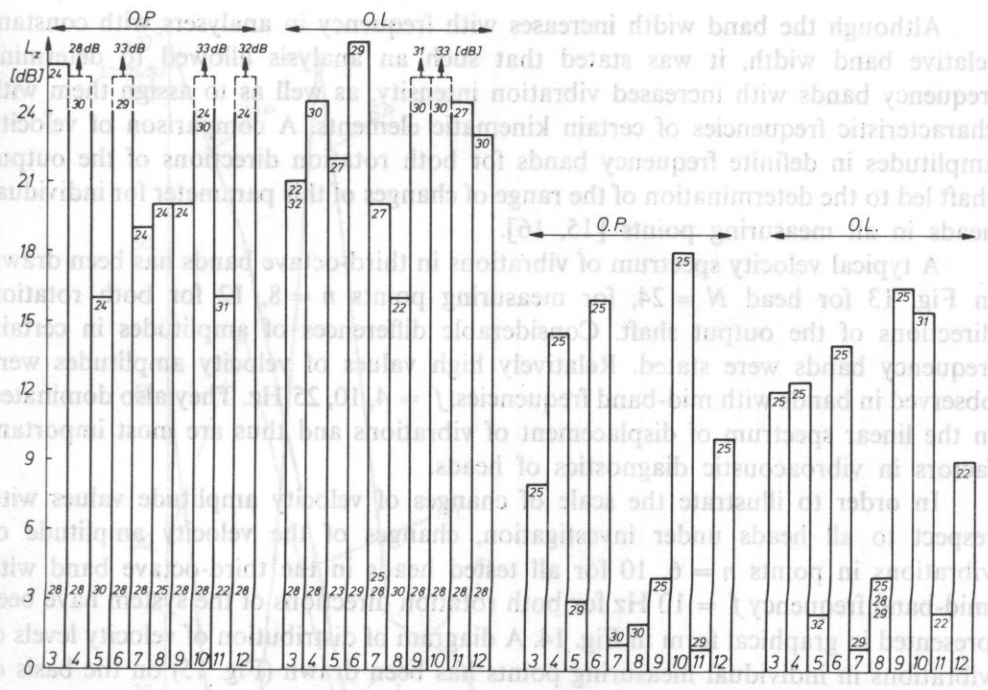


Fig. 12. Graphic districution of relative maximal and minimal amplitude values of vibrations displacement in individual points of the casing in a frequency linear band with $f = 25$ Hz

– a change of the amplitude of vibrations less than twice its value is insignificant,

– only a level change exceeding $L_x = 20\log 2 = 6$ dB can be a cause for a change of the dynamic state of a head,

– it can be accepted approximately that a change of the level of vibrations in a 6:1 ratio (equivalent to 16 dB) is equivalent to a change of the dynamic condition from good to inadmissible,

– according to these criteria a change of the level of vibrations which exceeds 16 dB should be equivalent to a transition to a supralimiting state, in which the head is unfit for exploitation.

If the presented above classification is taken into account, heads in the supralimiting state can be separated on the basis of diagrams of distributions of limiting levels of vibrations.

6.2. *Third-octave spectrum of velocity of vibrations*

On the basis of a third-octave spectrum analysis of the vibration velocity in a frequency range 1–200 Hz and in measuring points of the same heads, frequency bands with maximal velocity amplitude values of vibrations were determined.

Although the band width increases with frequency in analysers with constant relative band width, it was stated that such an analysis allowed to determine frequency bands with increased vibration intensity, as well as to assign them with characteristic frequencies of certain kinematic elements. A comparison of velocity amplitudes in definite frequency bands for both rotation directions of the output shaft led to the determination of the range of changes of this parameter for individual heads in all measuring points [15, 16].

A typical velocity spectrum of vibrations in third-octave bands has been drawn in Fig. 13 for head $N = 24$, for measuring points $n = 8$, 12 for both rotation directions of the output shaft. Considerable differences of amplitudes in certain frequency bands were stated. Relatively high values of velocity amplitudes were observed in bands with mid-band frequencies: $f = 4, 10, 25$ Hz. They also dominated in the linear spectrum of displacement of vibrations and thus are most important factors in vibroacoustic diagnostics of heads.

In order to illustrate the scale of changes of velocity amplitude values with respect to all heads under investigation, changes of the velocity amplitude of vibrations in points $n = 6$, 10 for all tested heads in the third-octave band with mid-band frequency $f = 10$ Hz for both rotation directions of the system have been presented in graphical form in Fig. 14. A diagram of distribution of velocity levels of vibrations in individual measuring points has been drawn (Fig. 15) on the basis of results of the amplitude-frequency analysis in the third-octave band with mid-band frequency $f = 16$ Hz in order to isolate heads with maximal and minimal values of levels. Maximal values of velocity levels of vibrations in this band were stated for

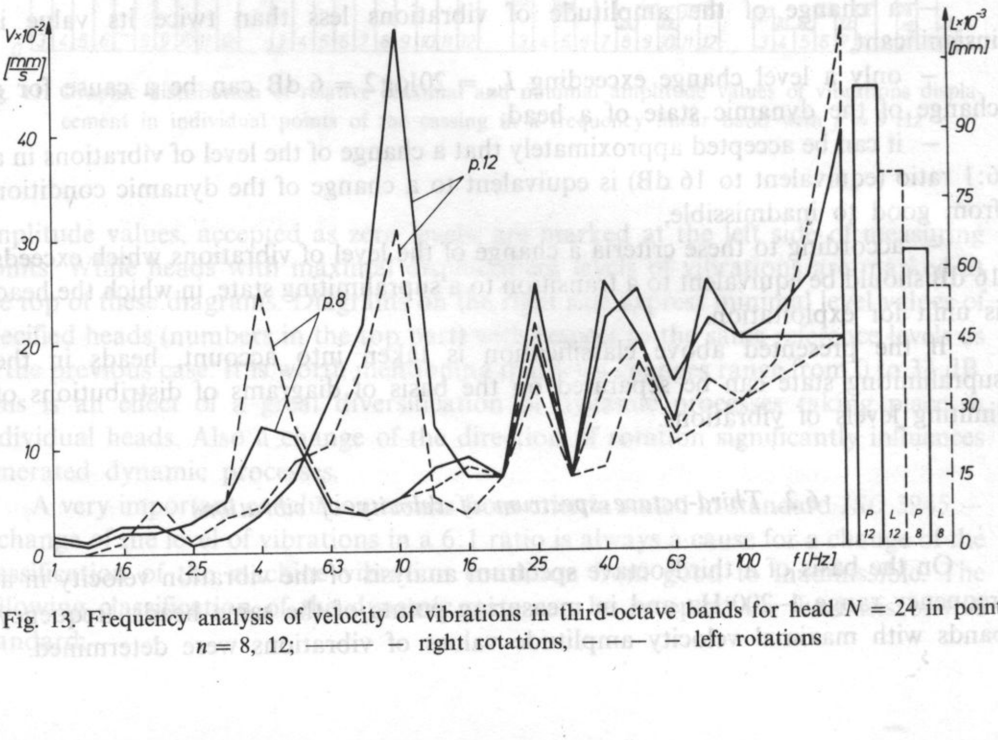


Fig. 13. Frequency analysis of velocity of vibrations in third-octave bands for head $N = 24$ in points $n = 8$, 12; — right rotations, — — left rotations

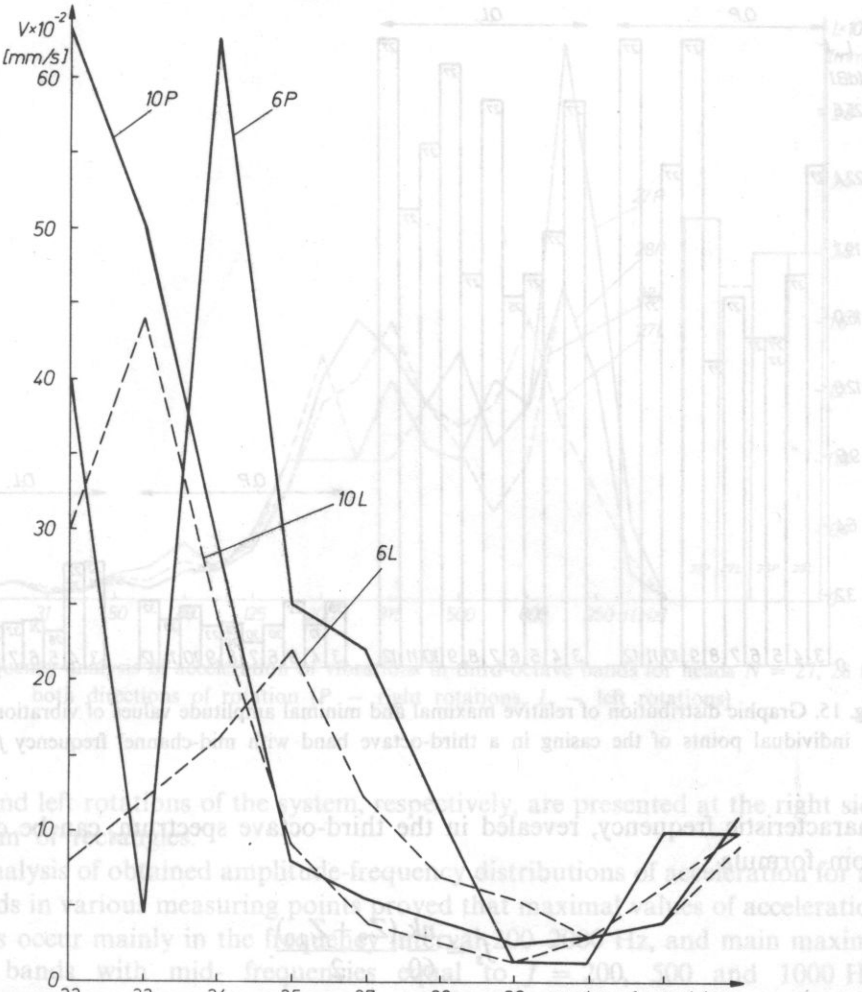


Fig. 14. Dependence of the velocity amplitude of vibrations in points $n = 6, 10$ upon investigated heads N in a third-octave band with mid-channel frequency $f = 10$ Hz

head $N = 27$ in nearly all measuring points for right and left rotations of the system. Hence, this frequency band dominates for this head. Highest values of the maximal velocity level for this band occured in points $n = 9, 12$, situated radially with respect to the main shaft of the planetary gear. Mid-band frequency $f = 16$ Hz, which dominates in the third-octave spectrum, corresponds to the characteristic frequency dependent on the superposition of vibrations generated by a series connection of a system consisting of a toothed planet wheel $Z_{10} = 19$, $m = 10$ and a toothed sleeve $Z_9 = 29$, $m = 10$ (Table 1). Taking mentioned above data into consideration, the

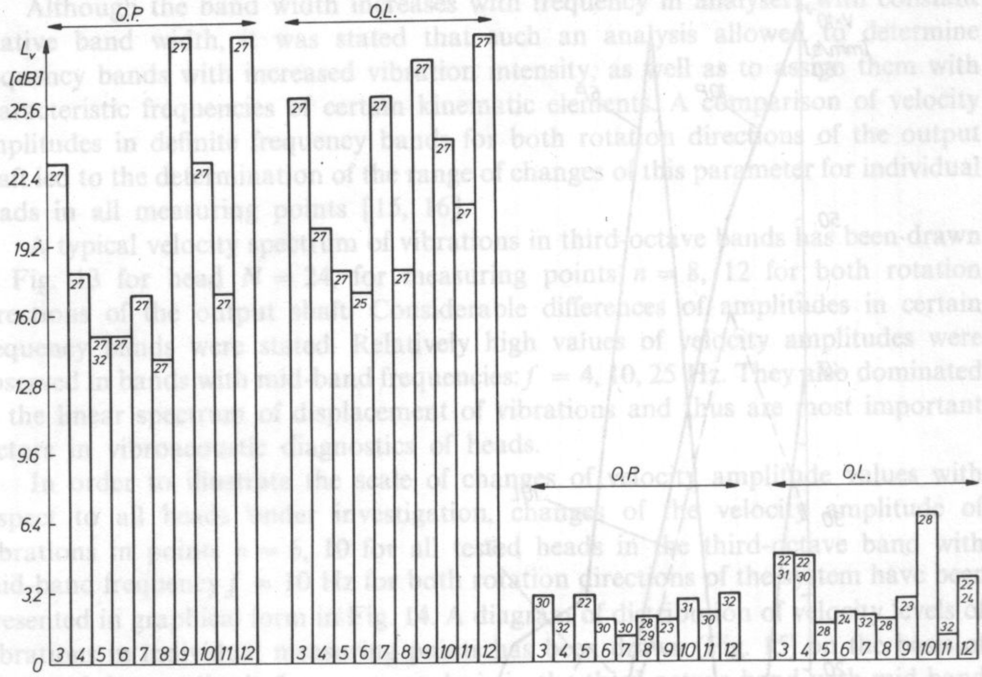


Fig. 15. Graphic distribution of relative maximal and minimal amplitude values of vibrations velocity in individual points of the casing in a third-octave band with mid-channel frequency $f = 16$ Hz

characteristic frequency, revealed in the third-octave spectrum, can be determined from formula

$$f_s = \frac{nk}{60} \cdot \frac{(Z_9 + Z_{10})}{2}.$$

For $k = 1$, we have

$$f_s = 16 \text{ Hz}.$$

This characteristic frequency was also found in the line displacement spectrum of vibrations (Fig. 7) in points $n = 9$, 12 of some of the investigated heads.

6.3. *Third-octave spectrum of acceleration of vibrations*

Also a spectral analysis of vibrations acceleration in third-octave bands was performed in frequency range 20–2000 Hz for all measuring points on bodies of investigated heads. A typical spectrum is shown in Fig. 16. It is a spectrum registered in measuring point $n = 12$ for heads $N = 27$, 28 in bands with marked mid-band frequencies. Rms acceleration values averaged with respect to all measuring points

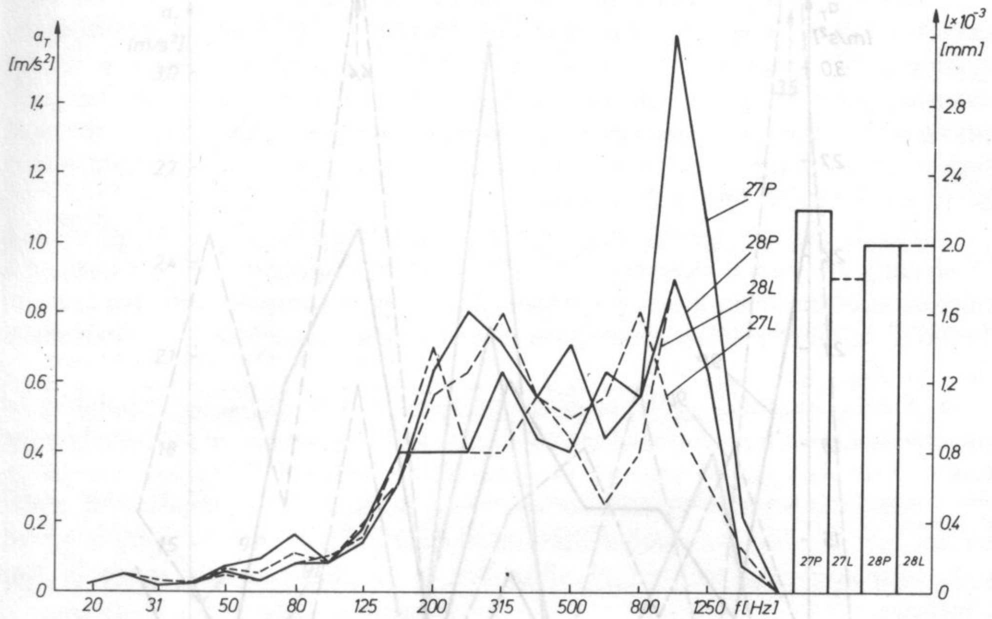


Fig. 16. Frequency analysis of acceleration of vibrations in third-octave bands for heads $N = 27, 28$ for both directions of rotation (P — right rotations, L — left rotations)

for right and left rotations of the system, respectively, are presented at the right side in the form of rectangles.

An analysis of obtained amplitude-frequency distributions of acceleration for all tested heads in various measuring points proved that maximal values of acceleration amplitudes occur mainly in the frequency interval 200–2000 Hz, and main maxima occur in bands with mid- frequencies equal to $f = 200$, 500 and 1000 Hz. Taking into consideration that the technologic state of bearings is represented relatively best by acceleration [2], it can be accepted that these maxima can reflected principally by peripheral wavines or bearing cage backlash and run-out, which occur in some rolling bearings. It results from calculations of characteristic frequencies of vibrations related to mentioned above defects, that acceleration spectra of vibrations in bands with specified above midband frequencies contain the following characteristic frequencies: $f_w = 194$ Hz (bearing 4), $f_w = 173$ Hz (bearing 1) and $f_w = 501$ Hz (bearing 2) (Fig. 3). Various values of acceleration amplitudes of vibrations in these frequency bands can reflect the dynamic state of specified bearings.

Figs. 17 and 18 present changes of acceleration amplitudes of vibrations in points $n = 9$, 10 of investigated heads in bands with mid- frequencies equal to $f = 800$ Hz and 1000 Hz. In these bands these amplitudes dominate in mentioned measuring points. Various values of acceleration amplitudes, which occur here, depend mainly on the investigated head (its dynamic state) and also on the rotation

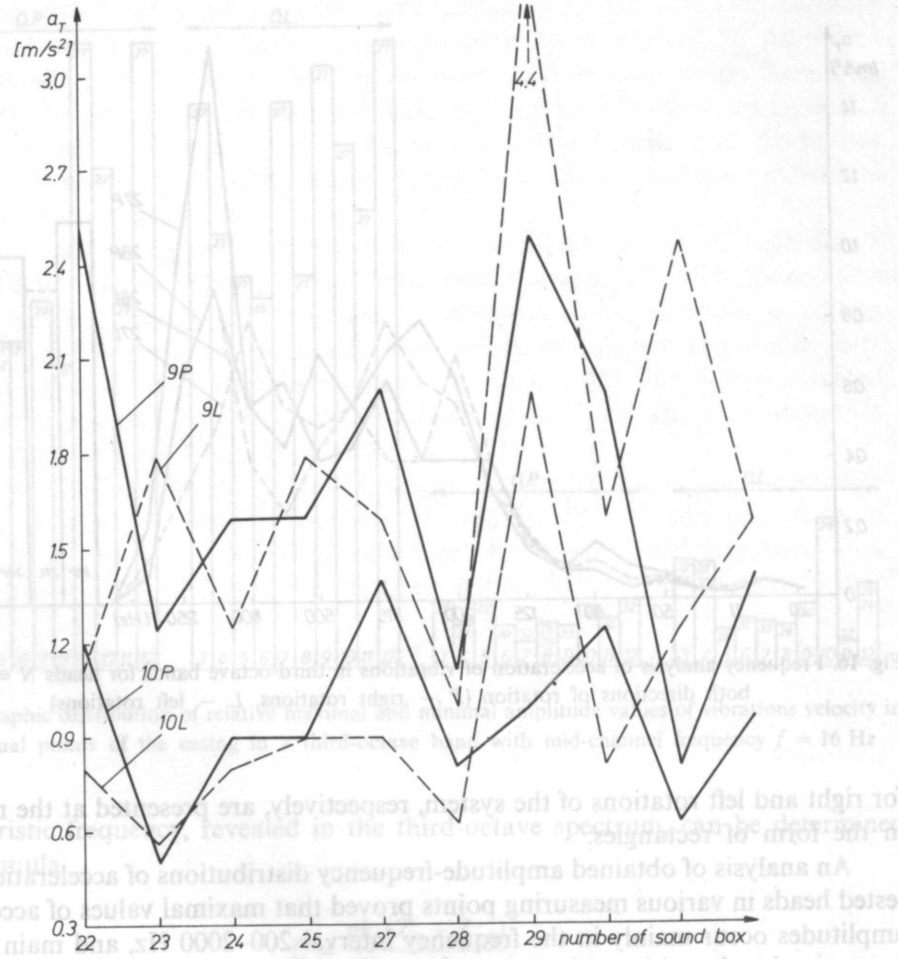


Fig. 17. Dependence of the acceleration amplitude of vibrations in points $n = 9$, 10 upon investigated heads N in a third-octave band with mid- frequency $f = 800$ Hz

direction of the output shaft. It is impossible to evaluate the importance of accelerations of dominating frequencies revealed in the third-octave spectrum at this stage of research. However, they may be of great importance in the process of vibraacoustic diagnostics.

7. Conclusions

Results of the amplitude-frequency analysis have indicated that the structure of vibrations received in individual points on the body of investigated heads is distinctly diversified. Frequency intervals with characteristic frequencies of certain kinematic

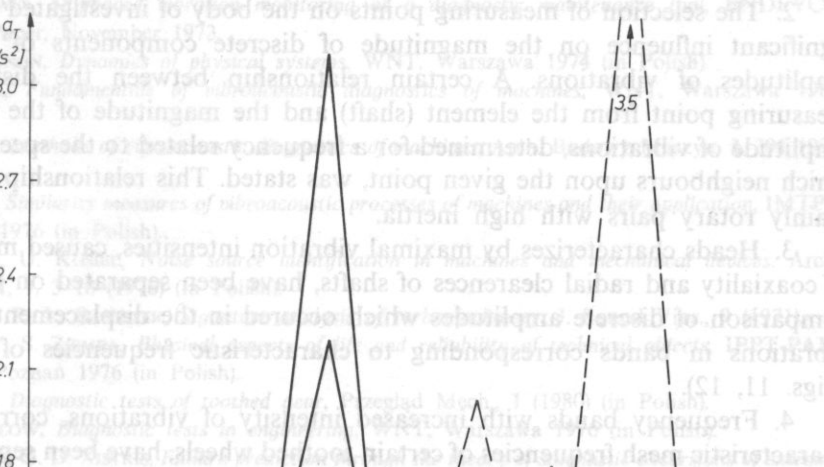


Fig. 18. Dependence of the acceleration amplitude of vibrations in points $n = 9, 10$ upon investigated heads N in a third-octave band with mid-channel frequency $f = 1$ kHz

elements of the system have been determined on their basis. A comparison of amplitudes in adequate frequency bands led to the determination of the range of maximal variations of measured parameters. A significant influence of the rotation direction on the structure of generated vibroacoustic signals was indicated. An analysis of measurement results has led to the following conclusions:

1. A spectral analysis of vibration displacements has disclosed discrete components related to revolutions frequencies of all main shafts and layshafts (Table 2), which testifies to their good frequency correlation. Considerable relative displacement amplitude level changes of vibrations, which were determined for revealed discrete components in definite points on the body of investigated heads, are due to unbalance or backlash in adequate rotary pairs or lack of their coaxiality.

2. The selection of measuring points on the body of investigated objects has a significant influence on the magnitude of discrete components of displacement amplitudes of vibrations. A certain relationship between the distance of the measuring point from the element (shaft) and the magnitude of the displacement amplitude of vibrations, determined for a frequency related to the speed of the shaft which neighbours upon the given point, was stated. This relationship characterizes mainly rotary pairs with high inertia.

3. Heads characterizes by maximal vibration intensities, caused mainly by lack of coaxiality and radial clearences of shafts, have been separated on the basis of a comparison of discrete amplitudes which occured in the displacement spectrum of vibrations in bands corresponding to characteristic frequencies of these shafts (Figs. 11, 12).

4. Frequency bands with increased intensity of vibrations, corresponding to characteristic mesh frequencies of certain toothed wheels, have been separated on the basis of velocity spectral analysis in third-octave bands.

5. Maximal velocity amplitudes contained in third-octave bands with center frequencies coinciding with frequencies for which amplitude maxima were observed in the displacement spectrum of vibrations, were stated in the range of low frequencies.

6. Heads with maximal and minimal velocity levels of vibrations were found from comparison of certain components of the third-octave spectrum of the velocity of vibrations. This means that the assembly and production of the kinematic system of investigated heads varies greatly.

7. Frequency intervals in which components with considerable amplitudes occured, were found on the basis of a spectral analysis of acceleration of vibrations in third-octave bands. These intervals are related to the dynamic state of rolling bearings and the precision of their mounting on shafts. Considerable variations of the level of vibrations were observed in investigated points during a comparison of maximal amplitude values, especially in the band of medium frequencies. This gives evidence of dynamic cooperation non-homogeneities of definite elements of tested heads.

8. The rotation direction of the output shaft influences significantly the value of discrete components in described spectra of vibrations. It is not unlikely that differences of values of definite spectral estimates, conditioned by the rotation direction of the output shaft, can carry important information about the assembly correctness and cooperation of certain elements of the system.

If detailed classification criteria would be worked out on the basis of many investigated objects of the same type, then the described method could become one of more effective quality inspection methods of produced heads.

References

[1] J. S. Bendat, A. G. Piersol, *Methods of analysis and measurement of random signals*, PWN, Warszawa 1976 (in Polish).

[2] M. P. Blake, W. S. Mitchell, *Vibration and acoustic measurement handbook*, New York–Washington, Spartan Books 1972.

[3] Ch. R. Bowes, *Shipboard vibration monitoring as a diagnostic maintenance tool*, ENDEVCO Technical Paper, November 1973.
[4] R. H. Cannon, *Dynamics of physical systems*, WNT, Warszawa 1974 (in Polish).
[5] Cz. Cempel, *Fundamentals of vibroacoustic diagnostics of machines*, WNT, Warszawa 1982 (in Polish).
[6] Cz. Cempel, *Methods of vibroacoustic diagnostics of machines*, Arch. Budowy Maszyn, **2**, 391 1978 (in Polish).
[7] Cz. Cempel, *Similarity measures of vibroacoustic processes of machines and their application*, IMTPP Report 23, 1976 (in Polish).
[8] Cz. Cempel, U. Kosiel, *Noise source identification in machines and mechanical devices*, Arch. Akustyki, **11**, 1, 3–18 (1976) (in Polish).
[9] V. Donato, R. L. Banister, *Signature analysis of turbomachinery*, J. Sound Vibr., **9** (1971).
[10] Z. Handzel, S. Ziemba, *Physical aspects of life and reliability of technical objects*, IPPT-PAN, Warszawa–Poznań 1976 (in Polish).
[11] L. Müller, *Diagnostic tests of toothed gear*, Przegląd Mech., **1** (1980) (in Polish).
[12] B. W. Pawłow, *Diagnostic tests in engineering*, WNT, Warszawa 1976 (in Polish).
[13] T. S. Sankar, G. D. Xistris, *Failure prediction through the theory of stochastic excursions of extreme vibration amplitudes*, J. Eng. Ind. (1972).
[14] W. Sikora, T. Zakrzewski, *Usability evaluation of amplitude-spectral discriminators in vibroacoustic diagnostics of arm heads of combined cutter loaders*, Zeszyty Naukowe Pol. Śl. (Mining), **137**, 99–139 (1985) (in Polish).
[15] T. Zakrzewski, *Vibration diagnostic method in the extractive industry*, Mechanizacja i Automatyzacja Górnictwa, **12** (1982) (in Polish).
[16] T. Zakrzewski, *Evaluation of the vibroacoustic state of arm heads*, Mechanizacja i Automatyzacja Górnictwa, 4/180 (1984) (in Polish).

Received on June 24, 1985; revised version on May 26, 1986.