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INVESTIGATION OF GEARBOX VIBROACTIVITY WITH THE USE OF VIBRATION AND ACOUSTIC PRESSURE START-UP CHARACTERISTICS

This article presents the results of investigating the influence of tooth contact ratio in helical cylindrical gears on vibroactivity of the gearbox. Based on the measurements carried out on a laboratory test stand, time-domain and frequency-domain start-up characteristics of vibrations and acoustic pressure were determined, and vibroactivity was assessed for a gearbox in which 4 pairs of gears were successively mounted with different face contact ratios equal to, respectively, ε_β=1,001; 1,318; 1,574; 2,636.

1. Introduction

Gearboxes are one of main vibration and noise emitters in drive systems. Their operation in the industrial conditions causes, in some cases, significant deterioration of the "vibroacoustic climate" in factory houses. Therefore, it is sought to improve the quality of work in this environment through limiting the vibroactivity of already operating gearboxes, and in the case of new

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drive systems, through the selection of the constructional solutions with low emissions of vibrations and noise.

Investigations conducted in various research centres have been focused on the reduction of vibrations in their source, i.e. in gear meshing and in bearings, as well as on the decrease of vibroactivity of housings, e.g. through the application of additional ribbing.

One of main vibration excitation types in gearboxes is the process of gear meshing. The analysis of dynamic phenomena taking place during this process is shown in papers [1-13]. The influence of wear and failures on dynamic loads acting on gear wheels is presented in [3,6,7]. Papers [1-10] show how the selection of basic meshing characteristics and oil used for lubricating influences the vibroactivity. The phenomenon of vibration transmission from gear meshing to bearing nodes is examined in [1,4,10,12]. The influence of shaft stiffness, as well as constructional and operational characteristics of bearing on the values of forces which induce vibrations of gearbox housings is the scope of research presented in [1,4,11-13].

In the analyses of gearbox housing vibroactivity, both the results of laboratory measurement and simulation results are taken into consideration. The results of laboratory investigations concerning the influence of housing wall thickness, as well as the location and dimensions of ribbing on gearbox vibroactivity are presented in papers [11,14,15,16]. The results of computer analyses of simple plates carried out with the use of FEM and BEM models is shown in papers [13,17-19], while the results of calculations for gearbox housings are shown in papers [20-23].

Some experimental researches [10] indicate significant influence of the face contact ratio of helical gearing on dynamical loads acting on the teeth. With the increase of this value, the loads decrease. Taking this fact into consideration, a thesis is formulated in this paper that the increase of face contact ratio will also cause the decrease in vibroactivity of a gearbox. The determination of a quantitative measure of decrease in vibroactivity of a gearbox housing with the increase of such face contact ratio was the goal of the prepared experiment. In estimations of the vibroactivity of gearboxes, start-up characteristics of vibrations and acoustic pressure acquired during their run-up and run down turned out to be useful.

2. Object and description of realised investigations

In the course of the investigations, 4 pairs of helical gear wheels with identical gear ratio were designed and manufactured, differing in face contact ratio ε_β and total contact ratio ε_C, which is shown in Figure 1 and in Table 1.

Measured and registered manufacturing accuracy of gearing in such wheels matched 6th grade of precision.

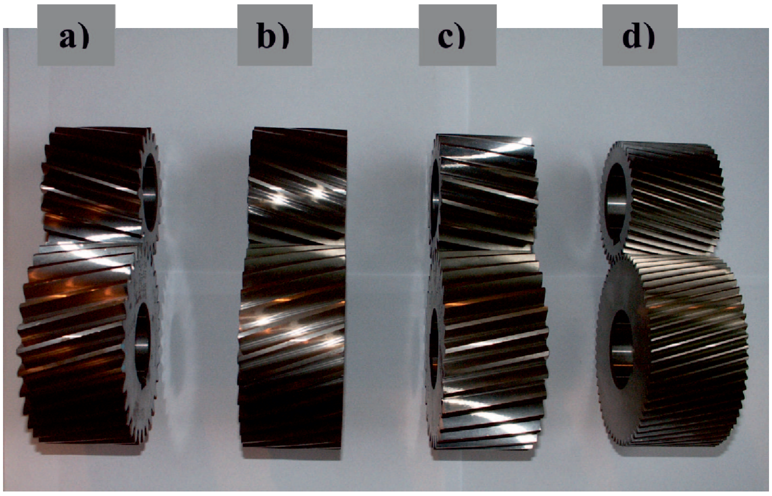

Fig. 1. Investigated gear wheels: a, b, c, d – respectively 1, 2, 3, 4 pair of gear wheels

Table 1.

Geometrical parameters of gear wheels used in investigations

	Pair 1	Pair 2	Pair 3	Pair 4
Number of pinion teeth z_1 [-]	19	19	19	38
Number of wheel teeth z_2 [-]	30	30	30	60
Normal module m_n [mm]	3,5	3,5	3,5	1,75
Normal pressure angle α_{on} [^{o}]	20	20	20	20
Helix angle β [^{o}]	11,333	15	18	15
Distance between the centers of two gears a_w [mm]	91,5	91,5	91,5	91,5
Transverse contact ratio ε_α	1,239	1,332	1,426	1,4
Face contact ratio ε_β	1,001	1,318	1,574	2,636
Total contact ratio ε_C	2,24	2,65	3	4
Coefficient of pinion addendum modification x_1	0,630	0,5	0,170	0,794
Coefficient of wheel addendum modification x_2	0,633	0,295	0,171	0,795
Face width b_w [mm]	56	56	56	56

It was assumed, according to [1,13,17-19], that the assessment of vibroactivity would be carried out based on the measurement of normal vibration velocity in selected points on the gearbox housing. At the same time, for the comparison purposes, values of acoustic pressure would be measured in the measurement point located above the gearbox housing.

View of the test stand FZG with measurement and recording equipment is shown in Figure 2.

Fig. 2. Test stand FZG with measurement/recording equipment

Measurement system was composed of (Fig. 2):
– laser vibrometer Ometron 300+, used for non-contact normal vibration velocity measurement on the gearbox housing,
– signal analyser Norsonic with a microphone, which was used for measuring acoustic pressure levels above the gearbox,
– optic sensors which measured the reference signal synchronising the location of shafts,
– data acquisition cards National Instruments NI 4472,
– personal computer for signal recording with LabView 8.6 software.

Normal vibration velocity measurements were carried out at 7 points on the top plate of the gearbox housing shown in Fig. 3, while acoustic pressure was determined at point H located at the distance of approx. 0,5 m above the central point of the housing K7.

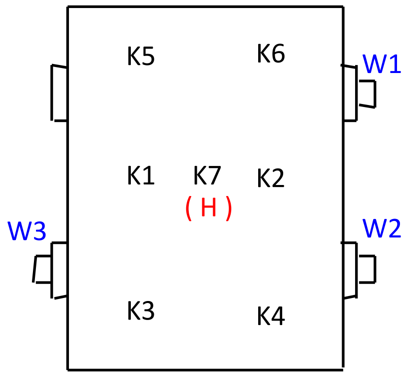

Fig. 3. Upper plate of the gearbox housing with marked vibration velocity measurement points K1-K7, as well as noise measurement point H

Measurements were carried out in the following conditions:

- at variable rotational speed of wheel shaft, assuming linear rotational speed changes in the ranges:
  - run up n_2=300 – 3000 rpm,
  - run down n_2=3000 – 300 rpm,
- at constant rotational speed of the wheel shaft W2 n_2=900 rpm or n_2=1800 rpm,
- unit load of gear wheels in the gearbox was equal Q=2,15 MPa [1],
- sampling frequency $f_{sampl.}$ =25 kHz.

Measurements were carried out in conditions of stabilised temperature of oil inside the housing in the range of 45 ±5°C. Recorded vibroacoustic signals, interpreted as the velocities of normal vibrations on the housing upper plate, and acoustic pressure in the selected measurement point were analysed in MATLAB/Simulink software.

Based on the recorded vibration signals, a quantity of vibroactivity v^2_{avg} defined in papers [1, 20-24] was determined for the upper housing plate, interpreted as a average value of normal vibration velocities for measurement points K1-K6 and separately for a centre point K7. This value is calculated from the following formula:

$$v^2_{avg} = \frac{1}{n} \sum_{i=1}^{n} \left(\frac{1}{k} \sum_{j=1}^{k} (v_{ij}(t_i))^2\right) \tag{1}$$

when:

v_{ij} (t) – vibration velocity in time t_i, j- measurement point,
n – number of samples of time analysis,
k – number of measurement points.

The RMS value of the acoustic pressure at the selected point above the housing plate was also determined.

3. Analysis of measurement results

Through the analyses of the influence of gearbox rotational speed on its vibroactivity, time-frequency distributions of normal vibration velocities were determined for point 7 of the housing and of acoustic pressure signals above this point on the housing, all recorded during running-up and running-down of the gearbox. Figure 4 and 5 show sample characteristics of vibrations and acoustic pressure in the case of operation of 1st and 4th wheel pair. The number of teeth in the 4th pair was two times greater than the corresponding amount of teeth from in 1st pair.

The frequency bands for which the increase of vibration and acoustic pressure amplitude takes place during the run up were determined based on the results of calculations of the average value of short time spectra of time-frequency distributions presented in Figures 3 and 4. In the figures, the spectra amplitude values are shown in a scale magnified by a factor of ten.

The frequency ranges, for which an increase in local vibration and acoustic pressure amplitude took place, were identified by carrying out modal analysis which, with the use of a modal hammer, determined the eigenfrequencies of the housing. The values of these frequencies are also shown in the considered figures.

One can notice that the increase in vibration and acoustic pressure amplitude takes place in the frequency bands corresponding to the selected eigenfrequencies of the gearbox housing.

Analysing the results of measurements, we could notice that during the run up of the gearbox with 1st pair of gear wheels installed a distinctive increase in vibration and acoustic pressure appeared at the rotational speed close to 1800 rpm, which corresponded to a meshing frequency of 900 Hz. We could also observe a small increase in these values at the rotational speed of 900 rpm (Fig. 4), at which meshing frequency was equal to 450 Hz. Similar changes were noticed for 2nd and 3rd pairs of wheels. The increase in vibration velocity and acoustic pressure amplitude at these rotational speeds was due to the fact that, correspondingly, the first or the second meshing frequency (vibration induction) was in these cases close to the dominant resonance frequency of the housing equal to 934 Hz.

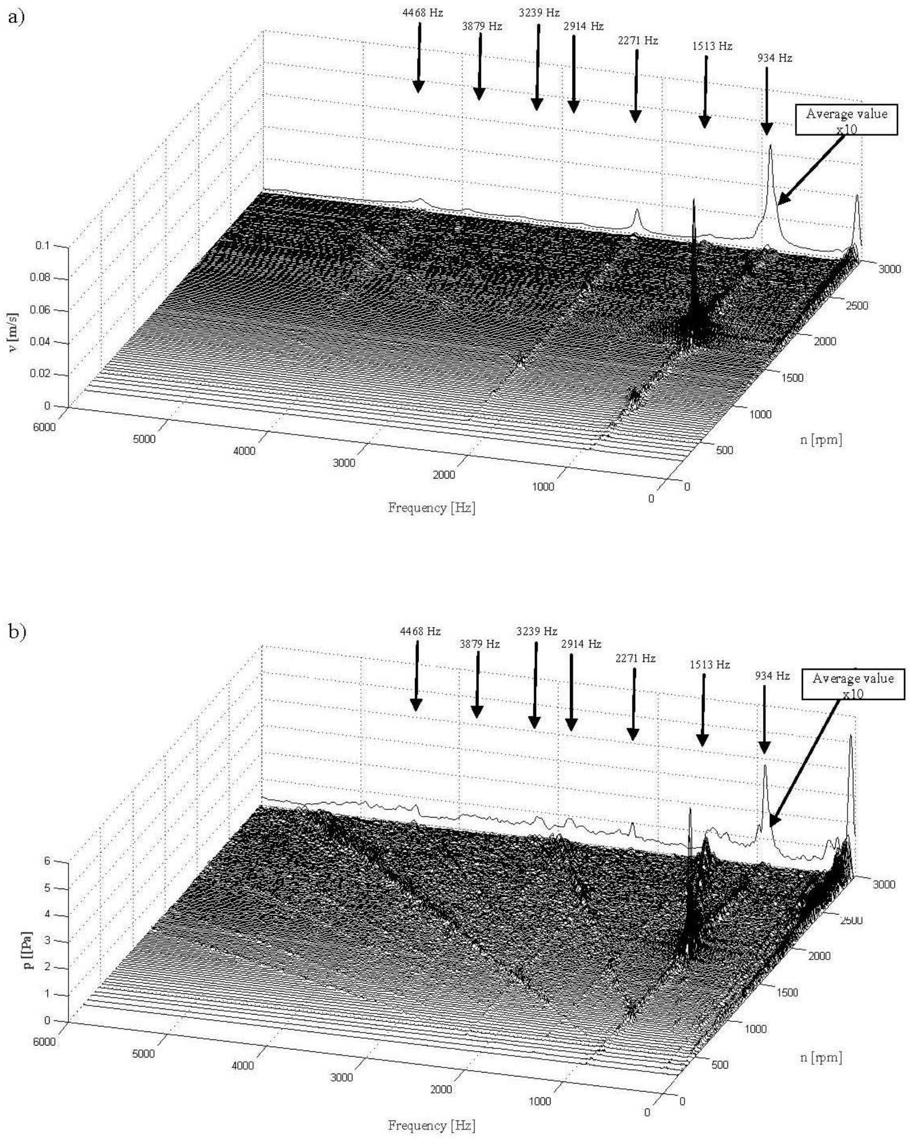

Fig. 4. Time-frequency distributions (pair 1): a) vibration velocities (K7), b) acoustic pressure (H)

In the case of the 4th pair of gear wheels, which was characterised by the same gear ratio but a twice greater number of teeth on both wheels, a rise in vibroactivity was observed within the rotational speed range around 900 rpm (Fig. 5), which corresponded to the meshing frequency of 900 Hz, close to the resonant frequency of the housing (934 Hz).

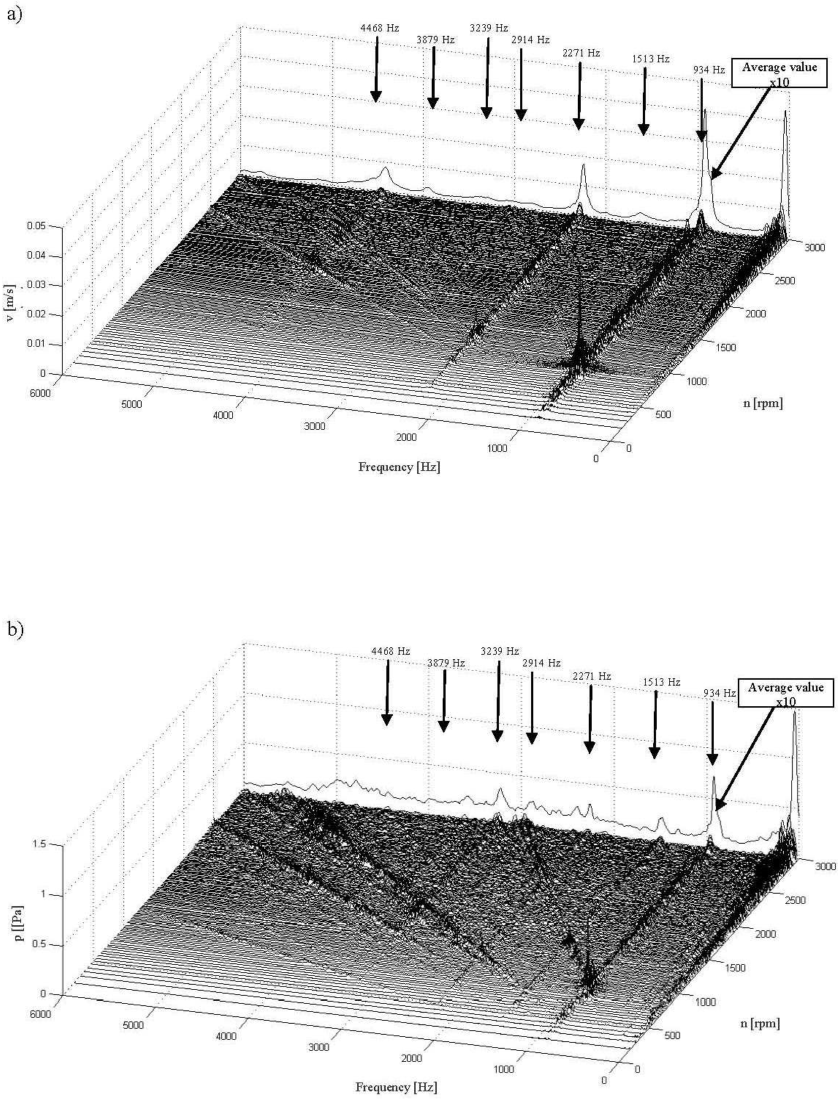

Fig. 5. Time-frequency distributions (pair 4): a) vibration velocities (K7), b) acoustic pressure (H)

Based on the measurements of normal vibrations at point 7, we determined the value of vibroactivity v_7^2 of the housing (calculated in the same way as v_{avg}^2, only in steps window) during gearbox run up and run down. At the same time, the values of acoustic pressure above this point were identified. Reference values were assumed equal to, respectively, $v_{ref}=5\cdot10^{-8}$ m/s and $p_{ref.}=2\cdot10^{-5}$ Pa. The results of these calculations for 4 pairs of wheels are presented in Fig. 6 and 7.

It can be noticed that the acoustic pressure and average vibration energy at the selected housing point (K7) vary in a similar way in the function of rotational speed of the gearbox. Analysing the nature of changes in these parameters, one can observe that their significant increase takes place in the cases when meshing frequency or its harmonics are close to the resonant frequency (934 Hz). This notion is confirmed by the results of time-frequency analysis of these signals.

a)

b)

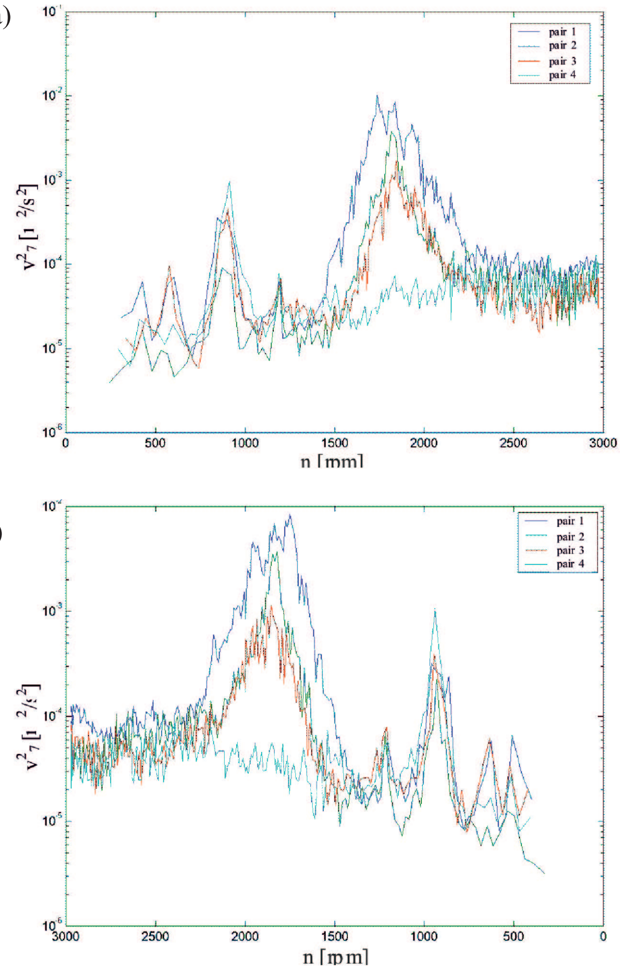

Fig. 6. The values of gearbox housing vibroactivity measure v_7^2 in function of wheel shaft rotational speed: a) run up, b) run down

Additionally, in order to analyse the influence of face contact ratio ε_β of helical gearing on gearbox vibroactivity, we conducted measurements of

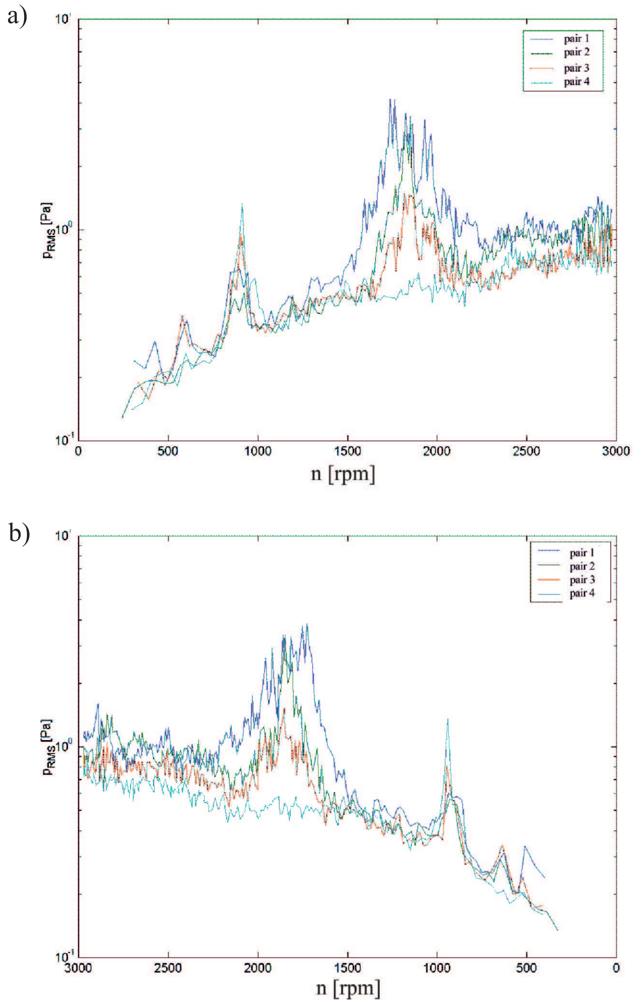

Fig. 7. Acoustic pressure in function of wheel shaft rotational speed: a) run up, b) run down

vibrations at selected housing points and acoustic pressure at constant speed of the gearbox equal to, successively, n_2=900 rpm and n_2=1800 rpm. The results of this analysis expressed in a dB scale are presented in Figure 8.

As it results from the measurements, the levels of average vibration velocity defined based on 6 measurement points v^2_{avg}, these levels determined for the central point of upper housing plate v^2_7, as well as the RMS value of acoustic pressure p_{RMS}, show similar relationships with the tooth contact ratio of helical gear wheels.

In the case when rotational speed was equal to 1800 rpm (fig. 8 a) a decrease in gearbox housing vibroactivity was observed together with an

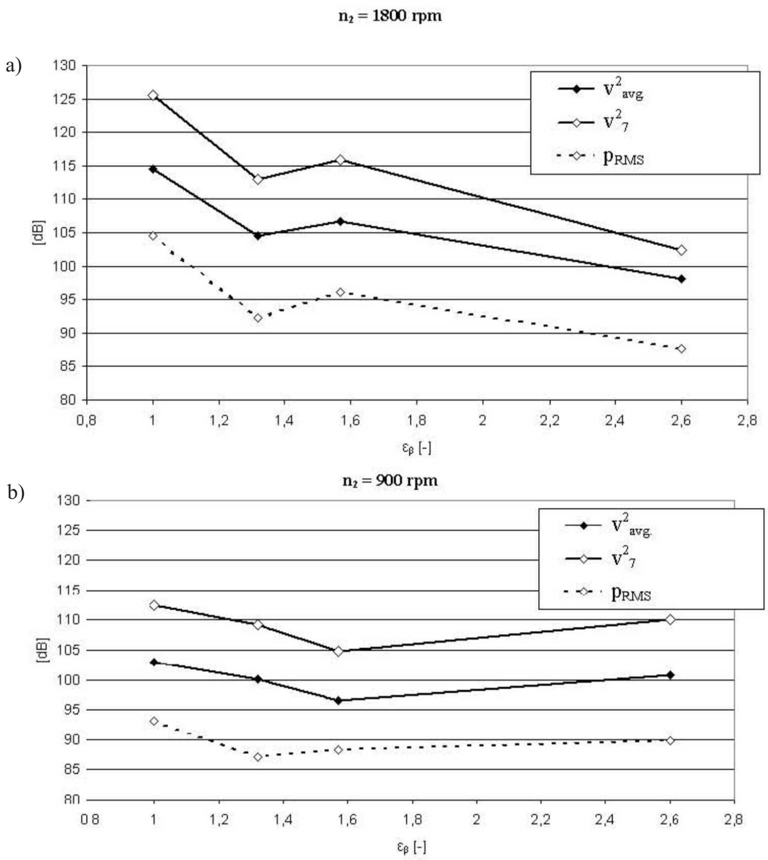

Fig. 8. Values of vibroactivity measure v^2_{avg}, v^2_7 and acoustic pressure p_{RMS} of gearbox housing in dependence on face contact ratio a) rotational speed n_2 =1800 rpm, b) rotational speed n_2 =900 rpm

increase in tooth contact ratio ε_β, which was caused by simultaneous decrease in dynamic forces generated in the meshing. However, in the case when rotational speed of the gearbox was equal to 900 rpm (fig. 8 b), a distinctive increase in vibroactivity with 4th pair of wheels was observed. In this case, meshing frequency was equal to 900 Hz, therefore it was close to the eigenfrequency of 934 Hz, which resulted in a significant increase in gearbox vibroactivity.

4. Conclusion

The results of research presented in this paper point to the purposefulness of using start-up characteristics of housing vibrations and acoustic pressure for assessing the vibroactivity of a gearbox. The analysis of such characteristics makes it possible to define ranges of gearbox rotational speeds in which significant increases of these values may take place. An increase in gearbox vibroactivity in such rotational speed ranges is due to the fact that vibration inducting frequencies, i.e. meshing frequency or its harmonic, are close to eigenfrequencies of the housing.

In the design process, it is possible to select meshing frequency (vibration induction) by changing the number of teeth on gear wheels, as well as eigenfrequencies of the housing by applying ribbing in such a way as to make these frequencies as far apart as possible. Such measures will exert a beneficial influence on vibroactivity of the gearbox.

The research conducted by the authors confirmed the thesis about a significant influence of tooth contact ratio in helical gearing on gearbox vibroactivity. With an increase in this parameter, as it is pointed out by our research, dynamic loads on gear wheels and vibration inducting forces in bearing nodes decrease significantly, which also causes a decrease in levels of vibrations and acoustic pressure emitted by the gearbox housing.

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Badania wibroaktywności przekładni zębatej z wykorzystaniem rozruchowych charakterystyk drgań i ciśnienia akustycznego

Streszczenie

Praca przedstawia wyniki badań wpływu liczby przyporu zazębienia skośnego kół walcowych na wibroaktywność przekładni. Na podstawie pomiarów prowadzonych na stanowisku wyznaczono rozruchowe charakterystyki czasowo-częstotliwościowe drgań i ciśnienia akustycznego oraz oceniano wibroaktywność przekładni, w której montowano kolejno 4 pary kół o różnej poskokowej liczbie przyporu wynoszącej odpowiednio ε_{β}=1,001; 1,318; 1,574; 2,636.