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DYNAMIC LOAD IDENTIFICATION IN GEAR WHEELS ON THE BASIS OF VIBROACOUSTIC MEASUREMENTS

Wilk Andrzej Professor, Ph.D., D.Sc., Eng.

The Silesian University of Technology - Gliwice Poland

ABSTRACT

Tooth working surfaces wear in wheels and other transmission elements at work increases dynamic loads. Optimum symptoms of dynamic load changes were sought for diagnosing in the elaboration. Simulating examinations of gear wheels with spur or helical teeth were carried out. Non-linear dynamic models of a pair of gear wheels were chosen as the basis of examinations. Their changing stiffness meshing, interteeth backlash and teeth errors were taken under consideration. It has been established that there is a linear dependence between $(K_\upsilon - 1 = P_d)$ additional dynamic load and the effective value of circumferential vibration accelerations of wheels.

Vibration measurements, most frequently for selected points of housing, are carried out in industrial gear weels. The examination of a pair of gear wheels working at circulating power system of the test stand were carried out. Circuit vibrations of rotating gear wheels and those of selected points of the housing were measured and analyzed simultaneously at the test stand by piezoelectric transducer.

Marginal correlation between non-filtrated effective values of vibration in transmission housing and circumferential vibrations of gear wheels has been observed. The correlation has increased significantly after selecting the housing vibration signal. This selection eliminated the dominating frequencies of free vibrations in the measured points.

1. SIMULATING EXAMINATIONS OF SPUR AND HELICAL GEARS

Dynamic models designed by L.Müller [1,2,4] are the basis of examinations. These models are presented in Fig.1,2 and they have been developed in order to make wear simulation of working surfaces of teeth possible (according to the assumed wear hypothesis).

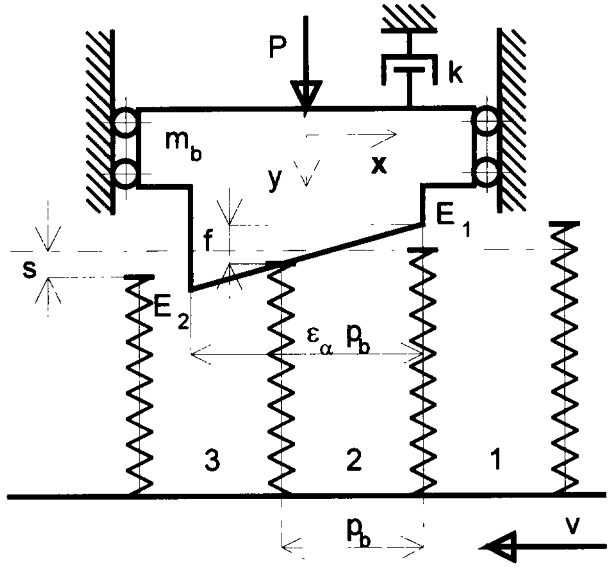


Fig. 1. Dynamic model of a pair of toothed spur wheels by L. Müller

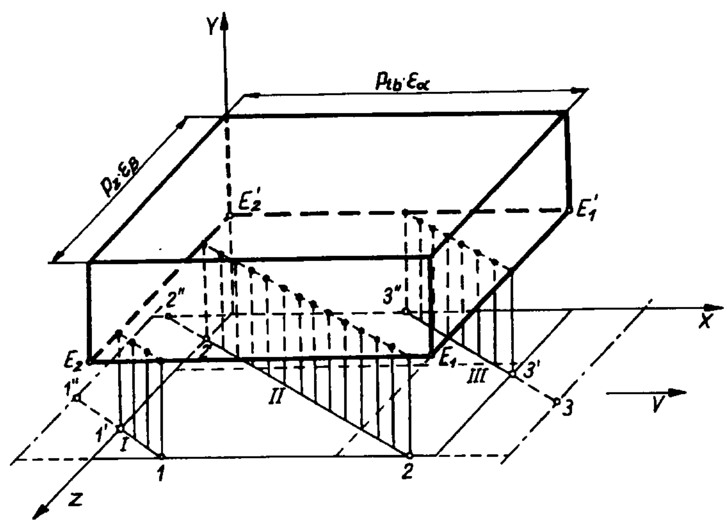


Fig. 2. Spacial model of a pair of toothed helical wheels

In the case of helical meshing, the model is a set of narrow spur wheels being shifted respectively in the phase of meshing but not being related to one another (in the case of independent base). Differential equation of model's movement has the following form:

$$a = \frac{d^2 y}{dt^2} = 1 - 2\varphi \frac{dy}{d\tau} - \frac{1}{\varepsilon_\beta\, t} \sum_{i=1}^{t\varepsilon} (c_{1i} u_{1i} + c_{2i} u_{2i}), \tag{1}$$

where:

τ – time,

t – number of replacing spur gear wheel corresponding to a pitch scale,

a – vibration acceleration,

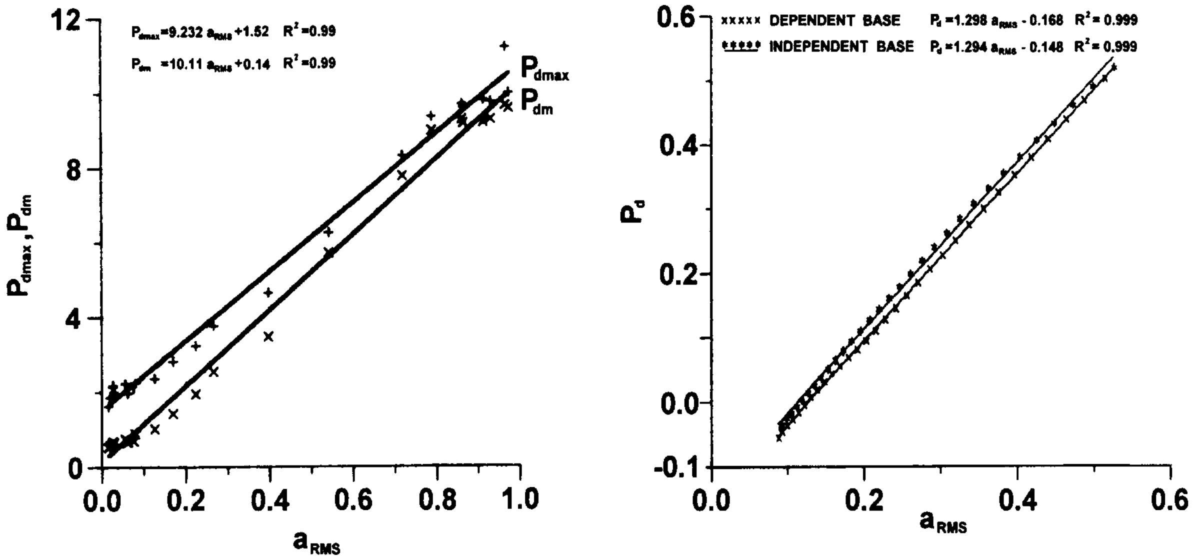
φ – damping coefficient,

ε_β – pitch coefficient number,
c_{1i} – rigidity of the first pair of teeth in i replacing spur wheel,
c_{2i} – rigidity of the second pair of teeth in i wheel,
u_{1i} – deflection of the first pair of teeth in i wheel,
u_{2i} – deflection of the second pair of teeth in i wheel.

Deflections of teeth can be calculated if the displacement of solid in a model is known. They represent relative circumferential displacement of a toothed wheel pair and the working surface wear of $g_{1,2i}$ teeth in presented replacing wheel.
Thus:

$$
\begin{aligned}
u_{1i} &= y - g_{1i} \\
u_{2i} &= y - g_{2i}
\end{aligned}
\qquad (2)
$$

The examined models represent a pair of toothed wheels which have been isolated from the other elements of the drive. Changing rigidity of meshing, interteeth backlash, transmission errors are considered in the model. Numerical method has been applied for solving differential equations of motion because of their nonlinear nature. Solving these equations enables to determine instantaneous value distribution of dynamic load in contact area.



a b

Fig. 3. P_{dmax} maximum value diagram and P_{dm} average dynamic surplus in dependence of the effective value of circumferential vibration acceleration in a_{RMS} wheels in case of: **a)** spur wheels which have the following parametres - relative motion speed of $\upsilon/\upsilon_{res} = 0.35$, transverse contact ratio of $\varepsilon_\alpha = 1.6$, random transmission errors related to tooth static deflection $s = 1$, **b)** wheels with helical teeth which have the following parametres - $\varepsilon_\alpha = 1.4$, $\upsilon/\upsilon_{res} = 0.8$, overlap ratio $\varepsilon_\beta = 0.8$

Fig. 3a presents an example of dynamic surplus diagram in effective value function of circumferential vibration acceleration in wheels during simulation, assuming that there is the increase of wear in working surfaces of teeth.

The carried out simulation examinations proved that there is a linear dependence between dynamic surplus and effective value of circumferential vibration acceleration of wheels. Linear dependence changes with tooth wear.

In the case of helical gears an exemplary diagram of dynamic surplus changes in effectiwe value function of circumferential vibration acceleration of wheels has been presented in Fig. 3.b. The diagram is based on simulation calculations. Linear dependence between dynamic surplus and acceleration effective value of vibrations has high value of R^2 correlation. Simulation examination carried out on a model where shearing stress between simple replacing spur wheels (dependent base) have been taken into consideration. However different geometric features of meshing and transmission errors have been taken into account as well. This acceleration effective value of circumferential vibrations of wheels proves to be a good symptom of dynamic load changes in teeth.

2. LABORATORY EXAMINATIONS

In gear drives it is difficult or even almost impossible to measure circumferential vibrations of wheels. It is also quite difficult to mount vibration transducer on the toothed wheel gear and the reception of the signal coming from rotating (together with the wheel) transducer is not easy either. Therefore wireless method of measuring vibrations by radiotelemetric system can be applied as well.

In case if there is no possibility of mounting the transducer on the toothed wheel, evaluation of dynamic loads can only be based on vibration measurements of selected points of housing.

2.1. Evaluation of dynamic loads of teeth in wheels based on vibration measurements of gear transmission housing

There is hardly any correlation between transmission housing vibrations and wheel circumferential vibrations. Therefore they cannot be used directly for evaluating changes of dynamic loads of teeth. Possibility of making use of vibration signal in transmission housing as a symptom of toothed wheel dynamic changes has been analysed in this paper as well as the way of selecting this signal for diagnosing. Measurements together with vibration analysis of transmission housing and toothed wheels were carried out during the so called active experiment at FZG work-stand.

2.1.1. Active experiment

The experiment was carried out at specially prepared work-stand presented in Fig. 4. The (1,2) tested pair of toothed wheels can work at different speed and load which can be controlled within certain ranges by a system of torsional shafts, tightening clutch and levers with weights. Piezoelectric transducers have been placed on no.2 wheel and in selected points of transmission housing which is marked in Fig. 4.

Diagram of experimental measuring system is presented in Fig. 5.

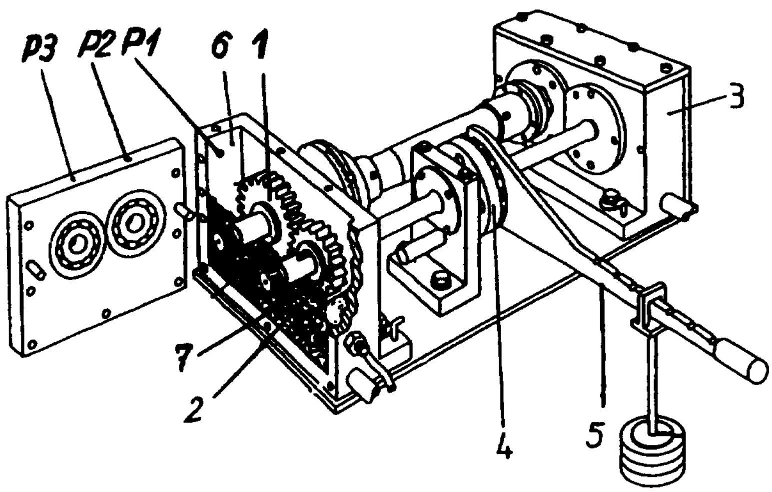


Fig. 4. FZG testing stand
1 – pinion,
2 – wheel,
3 – closing transmission with high strength wheels,
4 – tightening clutch,
5 – loading lever,
6 – tested transmission,
7 – piezoelectric transducer mounted on the wheel,
P1, P2, P3 – points of vibration measurements on transmission housing

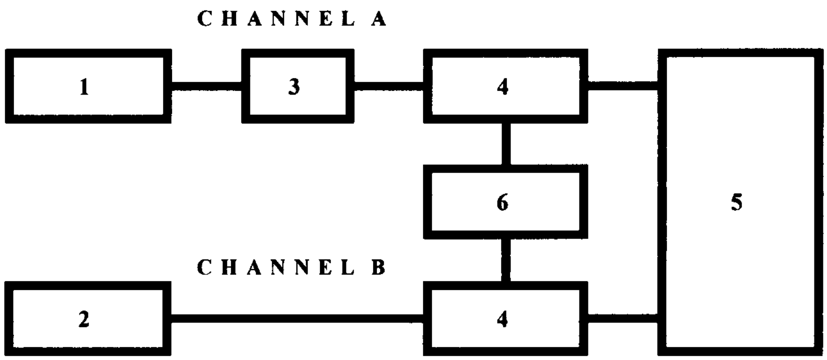


Fig. 5. Diagram of measuring system
1 – B&K 4335 piezoelectric transducer,
2 – VEB KD-35 71506 piezoelectric transducer,
3 – signal collector,
4 – PN-1 voltage preamplifier,
5 – GC-89 programmed signal analyser,
6 – (DC 15[V]) stabilized feeding

Measurement results for channel A and B were recorded simultaneously. Spur wheels have the following geometric parametres: 91.5 [mm] distance between wheel axles, 20 [mm] width of the wheels, 4.5 [mm] module, 16 teeth in a pinion, 24 teeth in a wheel. The wheels were made of 20H2N4A steel and then they were carbonized and hardened up to 60HRC hardness.

Three wheels of pre-programmed profile errors (f) were used during experiment interchangeably. The profile error was obtained during grinding process by setting different angles of tool profile. Measurements of errors of a tooth profile were carried out by PNC-40 measuring equipment (produced by Klingelnberg) coupled with microcomputer.

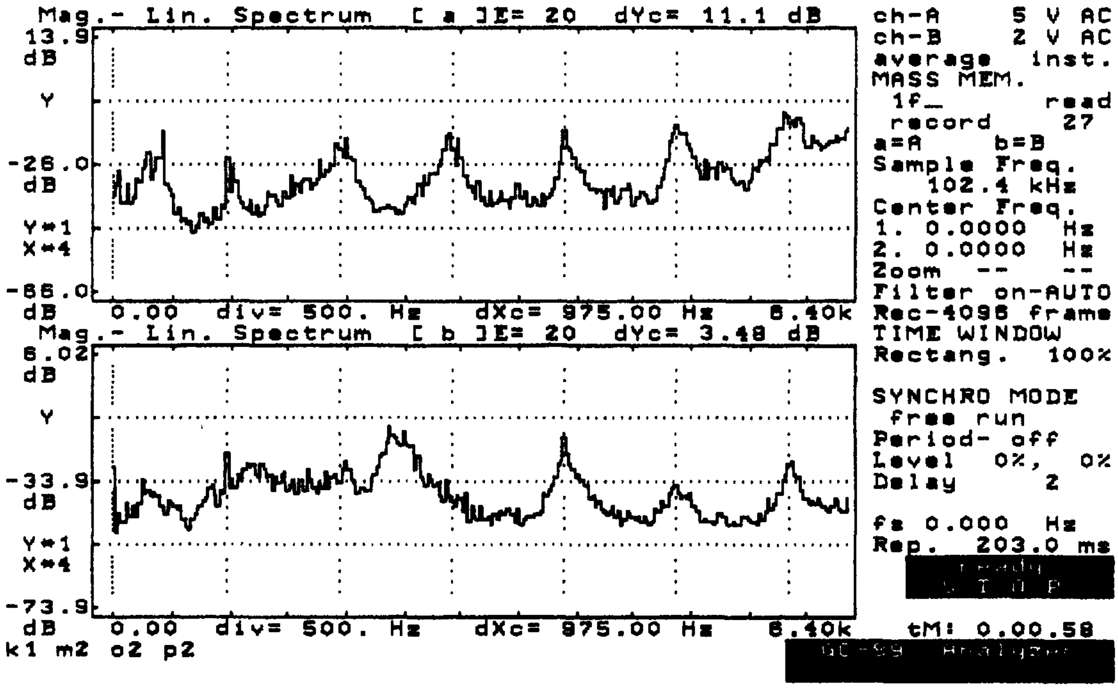


Fig. 6. Spectrum of circumferential vibration acceleration of a wheel (a channel) and vibration acceleration of point no.2 in transmission housing (b channel). Rotational speed of n_2 = 3750 rotation/min, load of Q = 3.85 MPa, f = -12μm

Total errors of pinion and wheels profile were -12μm, -37μm, -50 μm. Measurement conditions were as follows: for each pair of wheels there were three values of rotational speed of pinion - n_1 = 2871 [r.p.m.], n_2 = 3750 [r.p.m.], n_3 = 4410 [r.p.m.] and two values of [$Q = P/(b \cdot d_{w1})$] load intensity which correspond to the values of shaft turning moment of T pinion: Q_1 = 2.58 [MPa] (T_1 = 138 [Nm]), Q_2 = 3.85 [MPa] (T_2 = 206 [Nm]).

Fig. 6 presents examples of averaged spectrum of circumferential vibration accelerations of a wheel as well as spectrum of vibration acceleration of point no.2 of housing. Vertical dotted lines mark harmonic frequencies of wheel meshing. It can be noticed that in each selected point on transmission housing, free vibration frequencies of the housing are dominating in the spectrum. Free vibration frequency in point no.2 is 2.4 [kHz].

Examinations of vibration transmittance function from toothed wheel to housing have also confirmed domination of free vibrations. Vibration transmittance function was determined by impact forcing. Furthermore, amplitude of transmittance function reaches maximum values for free vibration frequencies of a selected point of housing during the carried out examinations.

The growing total profile error of pinion tooth and wheel causes increase of the effective value of wheel vibration signal. This dependence has been shown in Fig. 7.

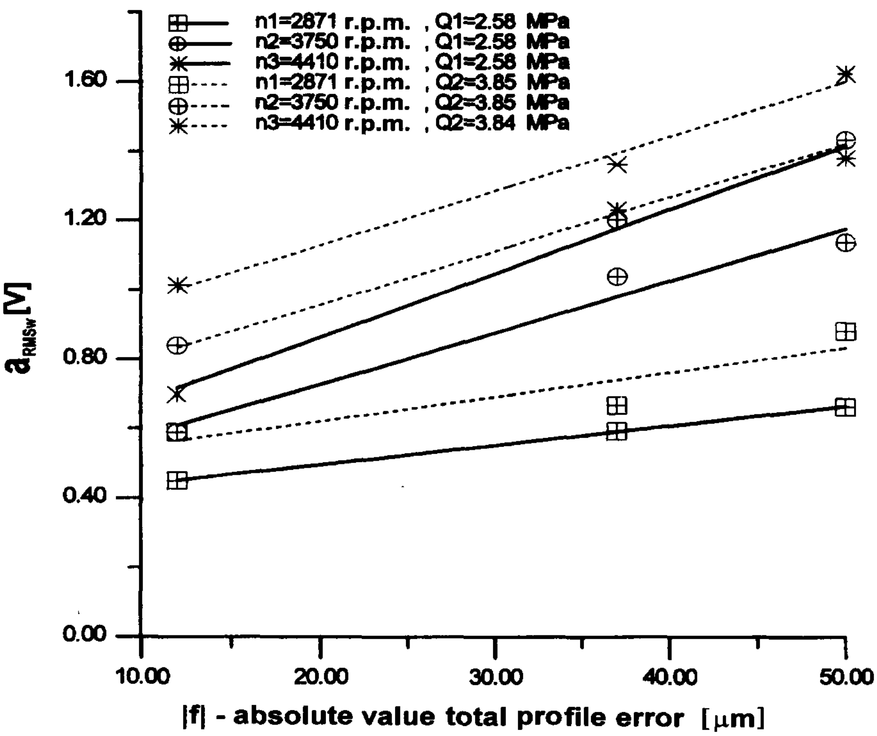


Fig. 7. Diagram of effective value of wheel vibration signal in (a_{RMSw}) wheel in deependence of absolute value of total profile error for different loads and rotational speed.

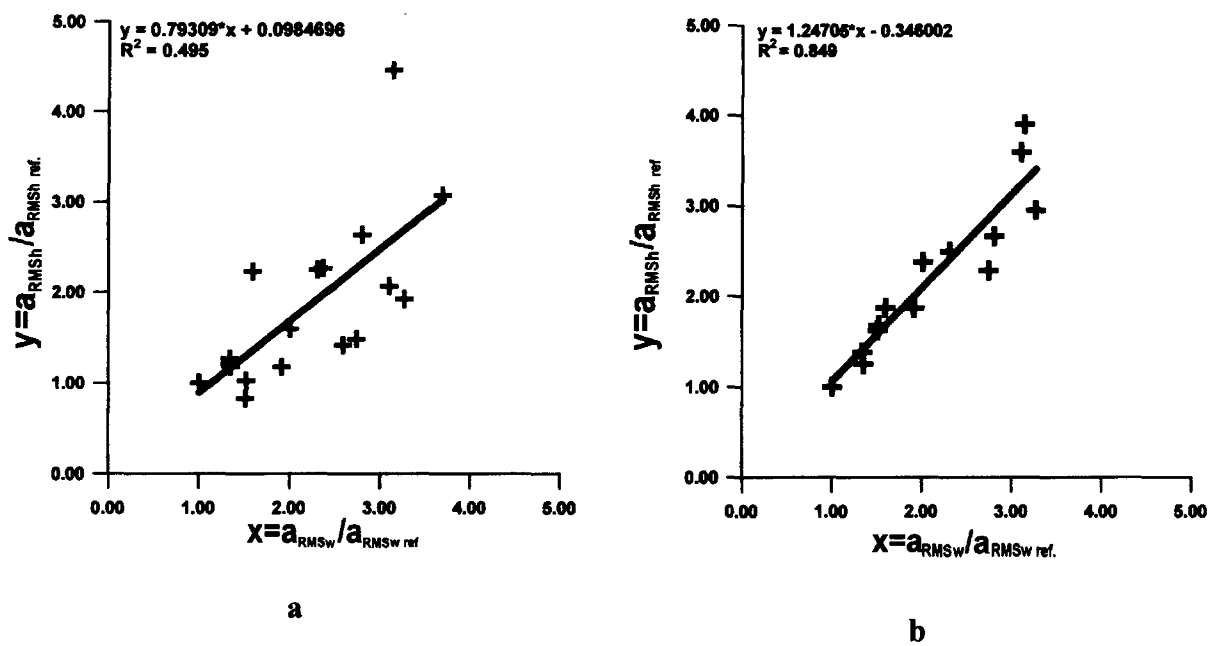


Fig. 8. **a**) relative effective value of vibration signal in point no.2 of transmission housing in dependence of relative effective value of wheel circumferential vibration signal, **b**) after application of 2.2–2.6 kHz blocking filter

Comparison of relative effective values of vibration signal of the wheel and point no.2 in transmission housing has been presented in Fig. 8a. These values were related to measured values at Q_l load, n_l rotational speed and f = 12 μm profile error value of toothed wheel pair. Calculated correlation coefficient is $R^2 = 0.495$ for linear approximation. Low value of correlation coefficient are due to dominating free vibration of frequency in vibration acceleration spectrum. This frequency is about 2.4 [kHz] and it certainly is an interference. Such interferring frequency was eliminated by (2.2–2.6 kHz) blocking digital filter in order to get a better correlation between vibrating signals of wheel and the housing.

Fig. 8b presents relative effective values of vibrating signal in point no.2 of the transmission housing. These values depend on averaged relative effective values of wheel vibration signal. Better correlation between examined vibration values of toothed wheel and housing was obtained in that respect. It is characterized by correlation coefficient: $R^2 = 0.849$ for approximation by straight line.

3. CONCLUSIONS

Carried out simulating examinations revealed that acceleration effective value of circumferential vibrations in toothed wheels seems to be a reliable symptom of dynamic load changes in teeth of transmissions at work. However, measurements of these vibrations in gear drives are made quite difficult because vibration signal is to be received from the rotating wheel. In some industrial drives it is possible to transfer the signal by radiotelemetric method applying specially designed and fed transmitters.

Most frequently, for practical reasons vibration measurements of selected points of drive's housing are carried out. These vibrations are usually not very well correlated with circumferential vibrations of toothed wheels. Therefore it is advisable to determine vibration transmittance function from toothed wheel to housing. As it was found during laboratory examinations, analysis of this function enables to determine free vibration frequency of the housing which interferes vibration signal. After appropriate selection of that signal by application of blocking filter it is possible to get better correlation of effective value changes of wheel and housing vibration signals. Effective value of housing vibration signal selected that way can thus be a symptom of dynamic load changes of wheels in gear drive at work.

Examination results of vibrations in gear drive housing in industrial work-stand at work proved suitability of the examined symptom for evaluating dynamic loads in toothed wheels as well as for determining the level of failure hazard due to exceeding the fracture fatigue strength of teeth.

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