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NUMERICAL RESEARCH OF TOOTHED GEARS GEOMETRY INFLUENCE ON POWER TRANSMISSION SYSTEM VIBROACTIVITY

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Abstract

The article presents results of numerical studies on the influence of selected factors on the dynamic effects and vibroactivity in power transmission system with toothed gear. Optimization of the constructions of the toothed gears is in many cases only possible by applying the numerical methods. Dynamic models allow determining the influence of a range of factors on the dynamic phenomena occurring during work of toothed gear. The studies were performed with use a custom developed dynamic model of a test stand with the gears operating in the circulating power system. It was assumed, that properly defined and identified model can be used to analyse dynamic phenomena occurring in meshing and bearings of toothed gears and allows optimizing their construction, especially allows minimization their vibroactivity. Numerical calculations were based on two sets of input data. Sets of input data consist information on all parameters included in the model, and which were determined during the identification process carried out in laboratory at two test stands. Selected results of simulations presented in this article prove, that identified dynamic model can be used to analysis of the impact of various constructional, technological and operational factors on the vibroactivity of the toothed gear.

Keywords: *power transmission system, toothed gear, dynamic model*

1. Introduction

Toothed gears are one of the main sources of vibration in power transmission systems. The reduction of vibrations and noise level generated by toothed gears can be attained among others by [6, 14, 15, 16]:
– reduction of vibration in the meshing zone,
– reduction of the effectiveness of the housing radiation,
– reduction of the effectiveness of vibration transmission.

Optimization of the constructions of toothed gears is in many cases only possible by applying the numerical methods. Extended dynamic models allow determining the influence of a range of construction and operation factors on the dynamic phenomena occurring during work of toothed gear [1, 2, 13]. Simulations and analysis of the results can lead to the minimization of dynamic effects in meshing zone and to meeting all assumed criteria by final product.

Descriptions of many models of gears and transmission systems can be found in literature. The studies were performed with use a custom developed dynamic model of a test stand with the gears operating in the circulating power system (FZG test stand) [8]. The model details have been presented in [9, 10]. The reason for using in the studies this dynamic model was high consistency of the computational results and the laboratory measurements (taken using nine pairs of gear wheels with different geometries). High complexity of the model allowed performing a wide range of analyses without disregarding any of the essential parameters describing the gear construction, which would not be possible if a simpler model was used.

In contrast to other models presented in the literature, it is characterized by a detailed description of the phenomena occurring in the meshing, with taking into account a non-linear description of meshing properties of both gears in a construction of FZG test stand. In addition, it takes into account the influence of elements of power transmission system – motor, driven machine, bearings, clutch and enables testing the dynamics of a power transmission system as a whole in variable operating conditions: rotational speed and load.

2. FZG test stand with gearboxes operating in a circulating power system

Test stand with gears working in a circulating power system (*Forschungsstelle für Zahnräder und Getriebebau-FZG*) is presented in Fig. 1. The test stand consists of an electric motor. It consists through a belt transmission drives the closing gear (1 on Fig. 1) and connected to it, by means of a torsional shaft (2), the tested gear (3). The experiments may be carried out at different loads controlled by means of torsional shafts, a tightening clutch and a lever with weights and at variable rotational speeds. Changes of rotational speed can be controlled by inverter, through which the motor is powered. The closing and tested gears have identical ratios and identical axle bases [3, 8, 10].

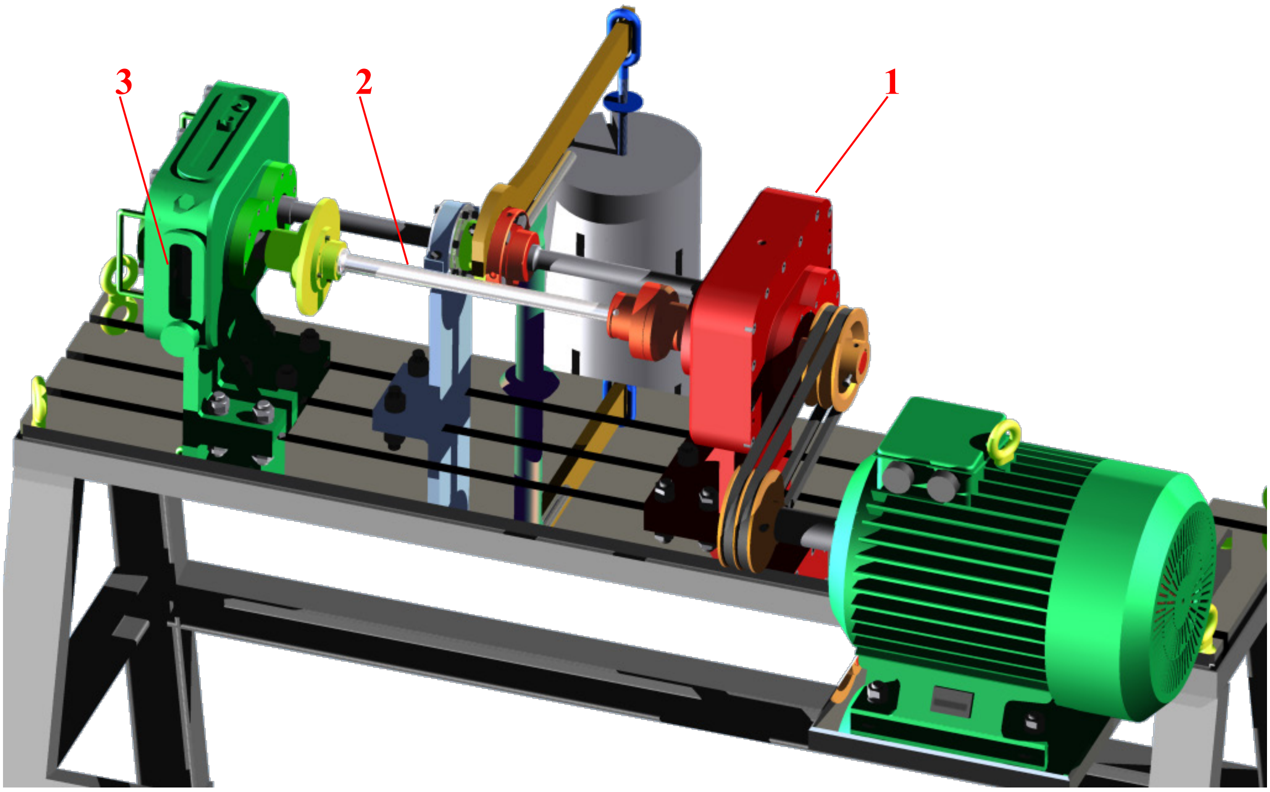

Fig. 1. Test stand with the gears operating in the circulating power system (FZG) [10]

The description of the dynamic model of FZG test stand has been presented in [10]. Model takes into account an extended description of the properties of meshing of both toothed gears, rotations of solids modelling the engine rotor, tension couplings, the pinion and the wheel of the closing gear and of the tested gear around the axis compatible with the direction of the axle of the gear shafts, the displacement in all bearings of the system in the direction of tangent force and normal force in the meshing, the torsional rigidity of the shafts, the rigidity of the supports, and damping in the bearings and shafts [4, 5, 12].

The simulation program consists of three main parts: a module of data input and preliminary calculations, a simulation module and a module of analysis of the results. The output files of the simulation are saved in a standard format of the MATLAB computational environment and they contain selected time courses of displacement, velocities, accelerations and forces. In addition, the time courses of the forces in bearings are saved in a standard version of the Nastran FEM software for the purposes of further research.

3. Research method

Numerical calculations were based on two sets of input data. They consist information on all parameters included in the model, and which were determined during the identification process carried out in laboratory at two test stands.

On Fig. 2 are presented the comparisons of the RMS values of the transverse vibration velocity of the gear wheel shaft measured in laboratory on the FZG test stand and obtained from the dynamic model. The graph presents the transverse vibration velocity as a function of the rotational speed and proves the accordance the simulation and experimental data. The verification was carried out in a wide range of load and speed of the nine different pairs of gears. All studies are presented in the work [10].

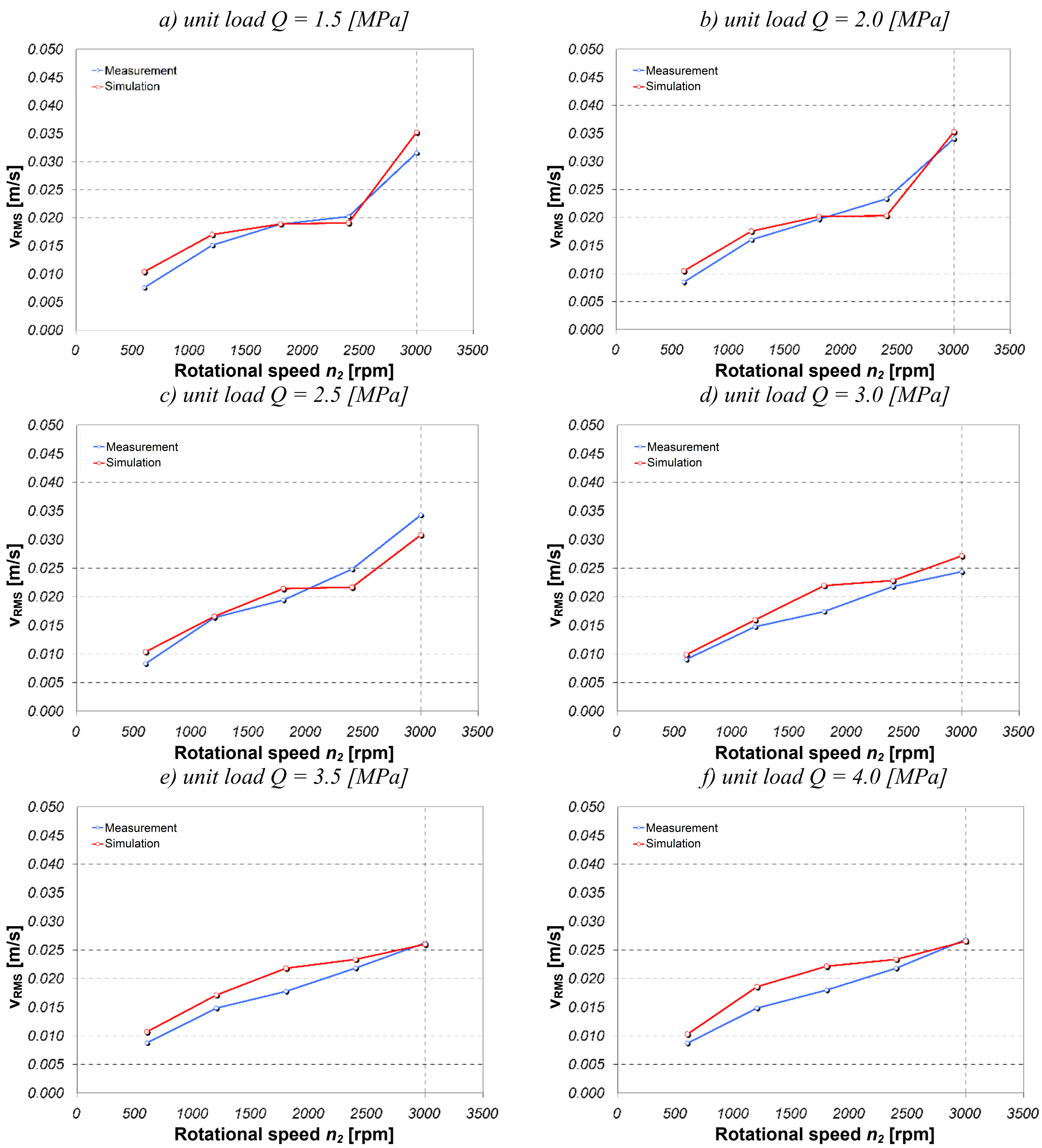

Fig. 2. Comparisons of the RMS values of the transverse vibration velocity of the gear wheel shaft, measured at test stand with the gears operating in the circulating power system and obtained from the model [10]

In the simulation computations, it has been assumed that the pitches of mating wheels have deviations. The maximum values of the deviations as well as the tooth form deviations were assumed basing on PN-ISO1328-1:2000 standard as allowable values in a specific class.

As a result of simulations, among all the calculated values, the following series were registered for further analysis purposes [12]:

- the engine rotational speed and torque,
- total intra tooth force in the tested gear,
- intra tooth forces between individual tooth pairs of the tested gear together with the numbers of the teeth in contact,
- the moment at the pinion shaft and at the shaft of the tested gear,
- the forces and velocities of transverse vibrations in the bearings of the tested gear,
- the velocities of the circumferential vibrations of the pinion and wheel,
- the efficiency of the tested gear.

As a result of each simulation, the time series of selected values were obtained corresponding to the gear operation during five wheel turns following the completed start up. To ensure full comparability of the results the recording was started each time at the moment the first tooth of the pinion met the first tooth of the wheel. The assumptions for the calculations were: minimum number of samples taken during the contact of the teeth at the base pitch is 1000; the calculations are performed applying the division of each of the wheels into 25 elementary wheels with straight mesh (regardless of the actual mesh type).

The analysis of the dynamic effects in the gear was performed primarily based on the values of dynamic forces in the meshing at the elementary cross sections of the wheels. These values together with the numbers of the teeth allowed calculating the total intra teeth forces carried by subsequent mating teeth pairs. The intra teeth forces defined for all the teeth pairs in contact allowed calculating the total intra teeth force. In the evaluation of the vibroactivity dynamic load coefficient K_d was used. Dynamic load coefficient is defined by formula [7, 8]:

$$K_d = (F_{stat} + F_{dyn}) / F_{stat}, \tag{1}$$

where:
F_{stat} – static intermesh force,
F_{dyn} – dynamic surplus of intermesh force.

4. Selected simulation results

In this section selected results of research for two from many analysed in [10] factors was presented. In each case was tested influence of chosen parameter in function of rotational speed of wheel mounted in tested toothed gear of simulated test stand.

The basic parameters of wheels mounted in tested gear of simulated FZG test stand are presented in Tab. 1.

Tab. 1. The parameters of the wheels mounted in the tested gears of the FZG test stand

Number of pinion teeth, z_1	16	[-]
Number of gear teeth, z_2	24	[-]
Helix angle, β	0	[º]
Module, m	4.5	[mm]
Transverse pressure angle, α_0	20	[º]
Axis distance, a_w	91.5	[mm]
Gear mesh width, b	20	[mm]

Calculations were realized in wide range of rotational velocities and for two values of unit load. Rotational speed of wheel in tested gear was changed in range n_2 = 0÷3600 rpm with step 300 rpm. In each simulation was taken into account this same distribution of teeth pitch deviations.

4.1. Periodic deviation of tooth profile

Due to different values of the periodic deviation of tooth profile form of the mating teeth, the value of total deviation may be positive or negative. The sign of the deviation, as shown in Fig. 3 and 4, has a significant impact on the dynamic loads in the spur gear.

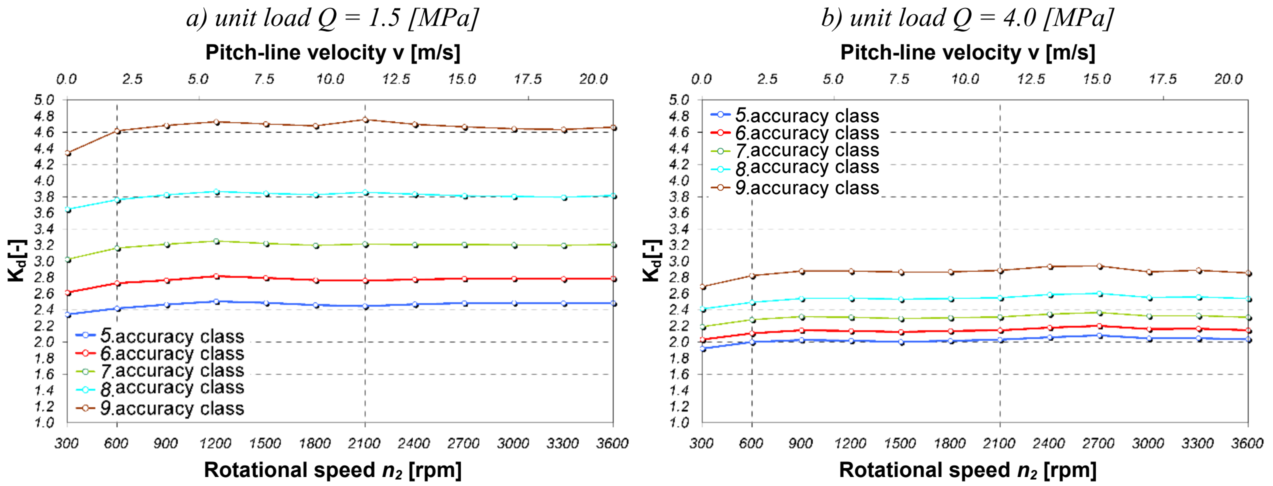

Fig. 3. Influence of quality class and rotational speed of wheel in toothed gear on values of dynamic coefficient K_d; negative value of total periodic deviation [10]

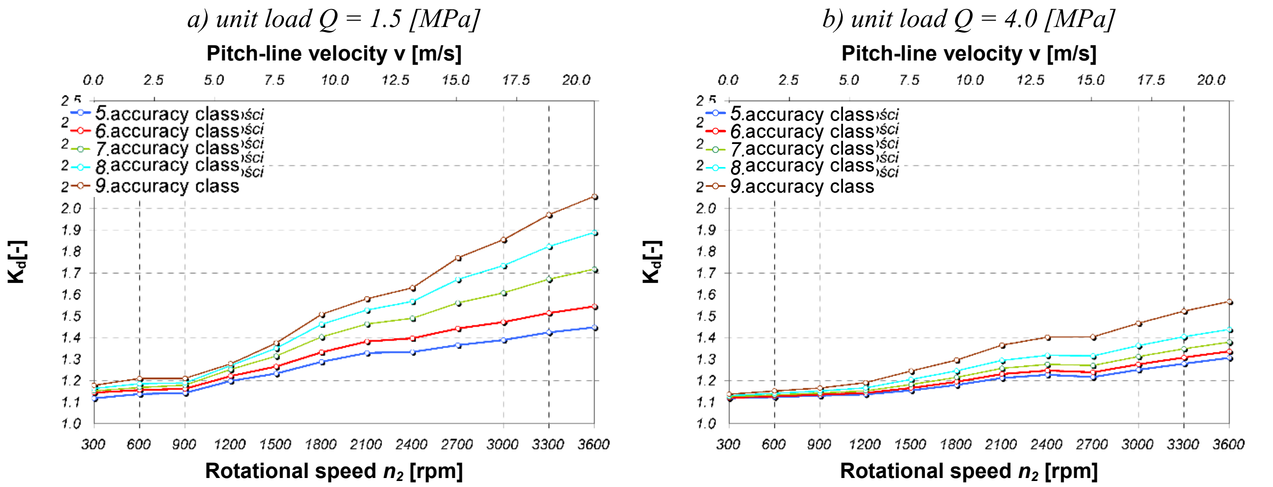

Fig. 4. Influence of quality class and rotational speed of wheel in toothed gear on values of dynamic coefficient K_d; positive value of total periodic deviation [10]

The gear wheels with positive values of total deviation have much smaller values of the dynamic load coefficient K_d as compared with the wheels with negative values of the deviations, regardless of the accuracy class of the mesh. This remains in agreement with results of other research.

4.2. Tip and root relief

The influence of the tooth modification [11] may be evaluated by comparing the calculation results presented on Fig. 5 (for toothed gear without modification of teeth) with results presented on Fig. 6 and 7 (adequately for modification type *A* and *B* – according to Maag [7, 8]).

As seen from these figures, the tip and root relief reduces the dynamic load coefficient K_d in the toothed gears for all studied accuracy class (5 to 9).

a) unit load *Q* = 1.5 [MPa] b) unit load *Q* = 4.0 [MPa]

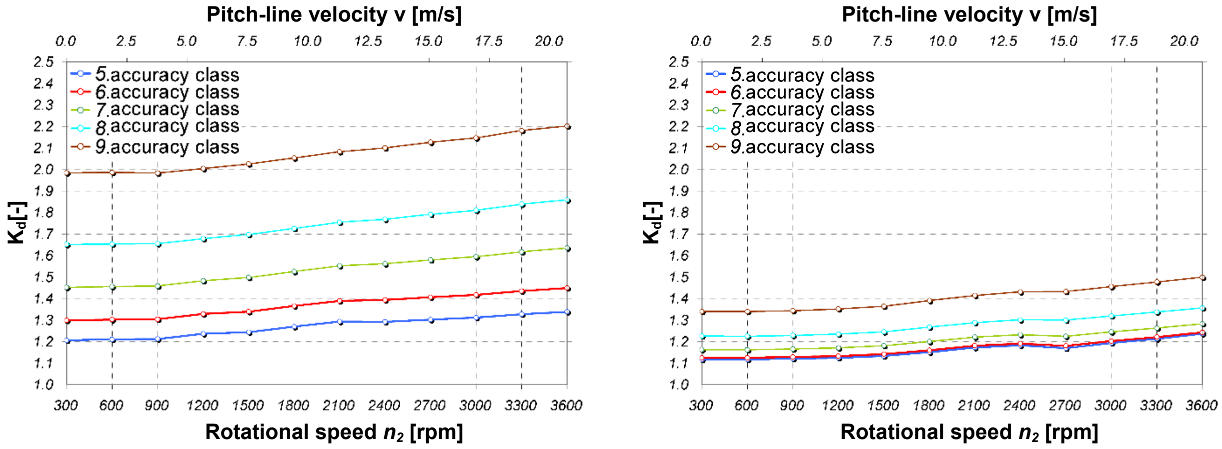

Fig. 5. Influence of quality class and rotational speed of wheel in toothed gear on values of dynamic coefficient K_d; toothed gear without tip and root relief [10]

a) unit load *Q* = 1.5 [MPa] b) unit load *Q* = 4.0 [MPa]

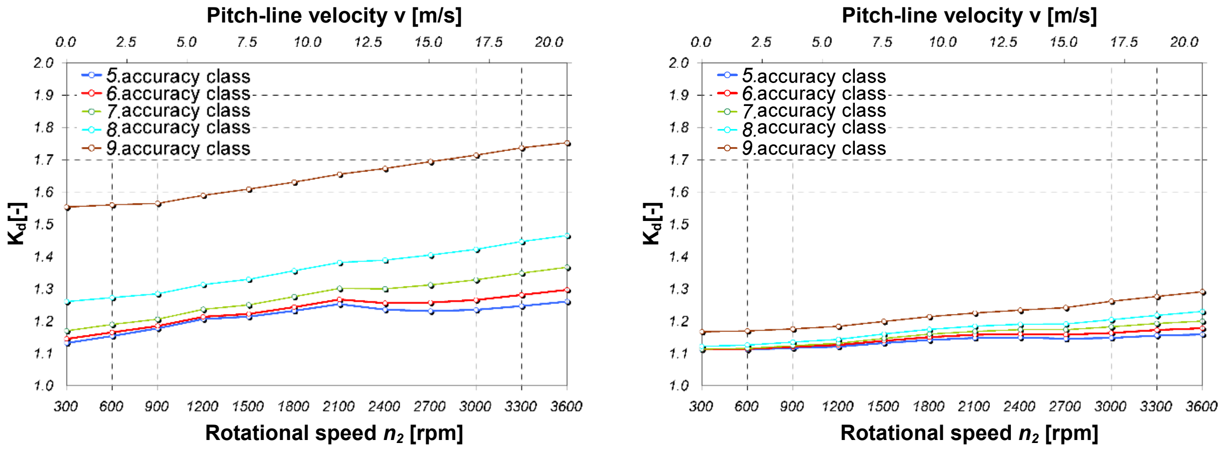

Fig. 6. Influence of quality class and rotational speed of wheel in toothed gear on values of dynamic coefficient K_d; toothed gear with tip and root relief – type A [10]

a) unit load *Q* = 1.5 [MPa] b) unit load *Q* = 4.0 [MPa]

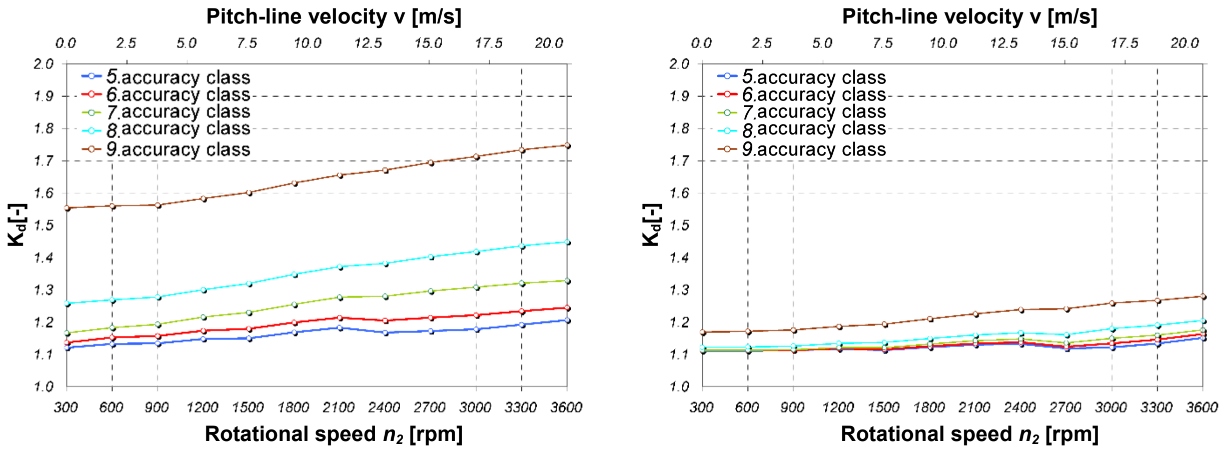

Fig. 7. Influence of quality class and rotational speed of wheel in toothed gear on values of dynamic coefficient K_d; toothed gear with tip and root relief – type B (according to Maag) [10]

5. Conclusions

Selected results of simulations presented in this article demonstrate, that identified dynamic model can be used to analysis of the impact of various constructional, technological and operational factors on the vibroactivity of the toothed gear. The described model was used in study to reduce the activity of gears.

The used model enables conducting analyses on the impact of all relevant factors on the vibroactivity of the toothed gear operating in the power transmission system, without disregarding the couplings, which occur between them.

The conducted comprehensive analysis of the impact of various factors on the vibroactivity of the toothed gear was included in the work [3, 14, 15, 16]. Obtained calculation results were then used as input data for FEM models during the research on the cumulative effect of the analysed factors and housing construction on the vibroactivity of the toothed gear. In this way, by combining the dynamic model with the FEM model of the gear housing, it is possible to carry out comprehensive numerical studies of toothed gears vibroactivity.

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