gearing, dynamic model, vibroactivity

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INFLUENCE OF CONSTRUCTION FACTORS ON THE VIBRATIONAL ACTIVITY OF THE GEARING

Summary. The paper presents the results of numerical and experimental studies on the influence of selected factors on the dynamic effects and vibroactivity of the gearing. The studies were performed using a custom developed dynamic model of a test stand with the gears operating in the circulating power system.

WPŁYW CZYNNIKÓW KONSTRUKCYJNYCH NA WIBROAKTYWNOŚĆ PRZEKŁADNI ZĘBATEJ

Streszczenie. Artykuł przedstawia wyniki badań symulacyjnych oraz laboratoryjnych mających na celu określenie wpływu wybranych czynników konstrukcyjnych na zjawiska dynamiczne i wibroaktywność przekładni zębatej. Badania numeryczne przeprowadzono z użyciem opracowanego modelu dynamicznego stanowiska z przekładniami pracującymi w układzie mocy krążącej.

1. INTRODUCTION

Minimisation of gears' vibroacoustic activity requires an analysis of the mechanism of the dynamic forces occurrence in the meshing, transmission of these vibrations to the radiation point as well as various aspects related to sound radiation through the housing. It implies that there are several ways of decreasing the gear noise emission level by reduction of vibration in the meshing area, the effectiveness of vibration transmission and the effectiveness of the housing radiation.

This paper presents methodology of shaping the vibroactivity of the gearbox using a dynamic model of the gearbox and an FEM model of its housing. The methodology adopted for designing gearboxes is the same as used in paper [2]. It is presented in fig. 1.

2. DYNAMIC MODEL OF THE GEARBOX

For the study, a dynamic model was used of a stand with gearboxes operating in a circulating power system which enables examining the impact of internal and external induction on the dynamic forces in meshing and the forces in the system bearings. The model was developed using the Delphi software environment and it entailed mutual impacts of various external and internal factors occurring while a gear is operating in a power transmission system [5, 1]. The dynamic model's diagram has been provided in fig. 2.

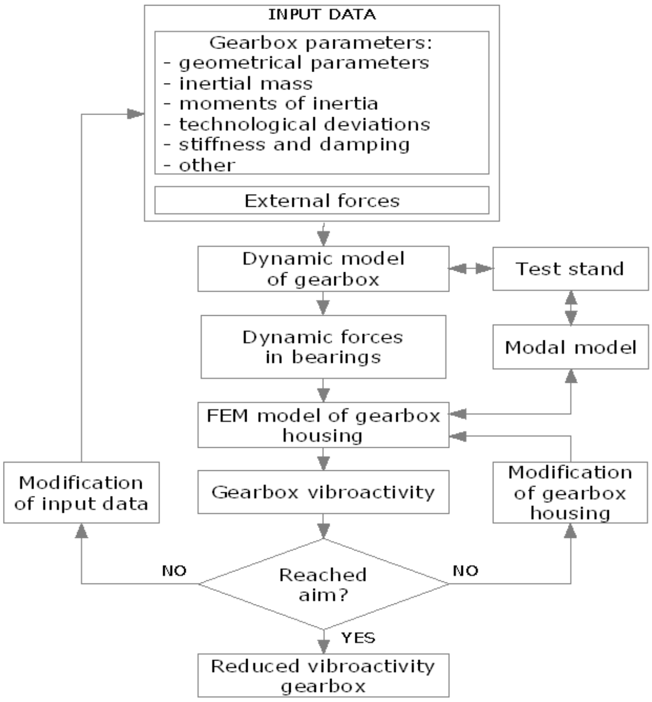


Fig. 1. Design methodology for gearboxes with reduced vibroactivity
Rys. 1. Metodyka projektowania przekładni o obniżonej aktywności wibroakustycznej

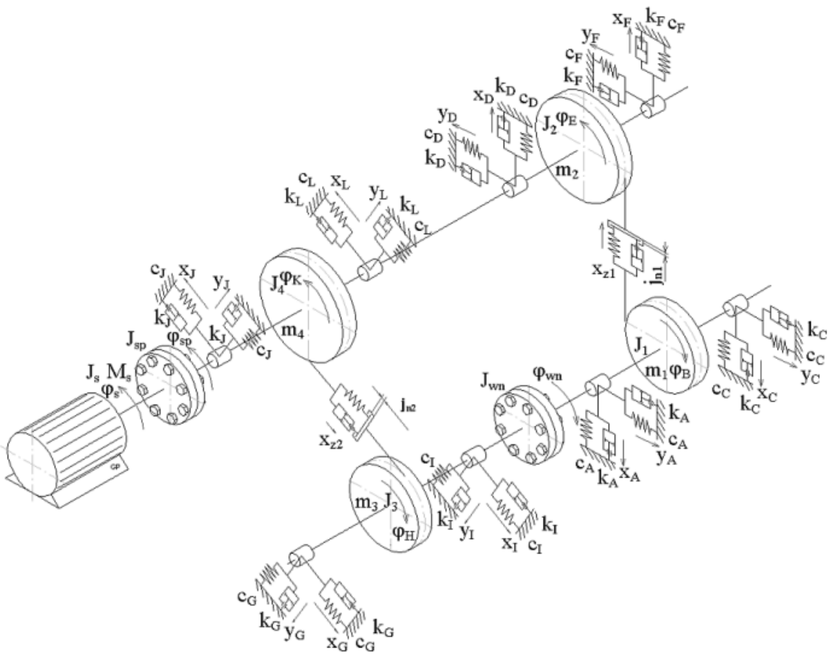


Fig. 2. Dynamic model of test stand with the gears operating in the circulating power system
Rys. 2. Model dynamiczny stanowiska z przekładniami pracującymi w układzie mocy krążącej

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 MSC Nastran software for the purposes of further calculations.

The dynamic model of the test stand, which is presented on the figure 3, takes into account: an extended description of the properties of meshing of the closing (2) and the tested gears (1), operation at a variable rotational speed, rotations of solids modelling the engine rotor, tension couplings (3), the pinion and the wheel of the closing gear and of the gear examined 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.

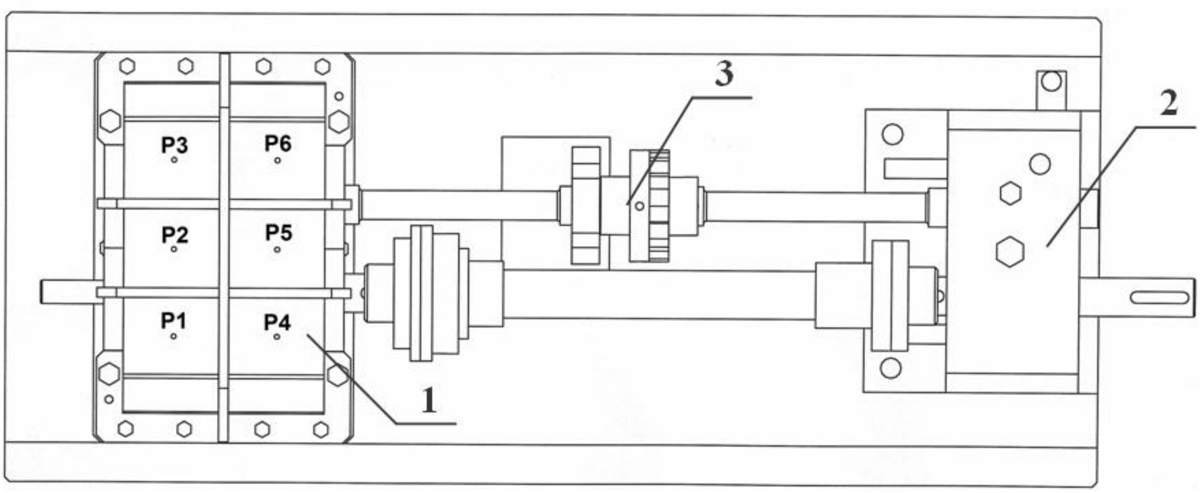


Fig. 3. A diagram of the test stand
Rys. 3. Schemat stanowiska badawczego

The model allows testing a wide range of design solutions. The variables used in the design process comprise [5]:

- the rotational velocity,
- the load,
- the geometry of toothed wheels determined by the number of teeth and the coefficients of addendum modification, module, helix angle, the width of meshing, and the backlash,
- tip and root relief, end relief, crowning,
- pitch, profile and helix deviations,
- deviations resulting from assembling the gear components,
- rigidity of bearing assemblies and shafts,
- damping in the meshing and in bearings,
- inertial mass and moments of inertia.

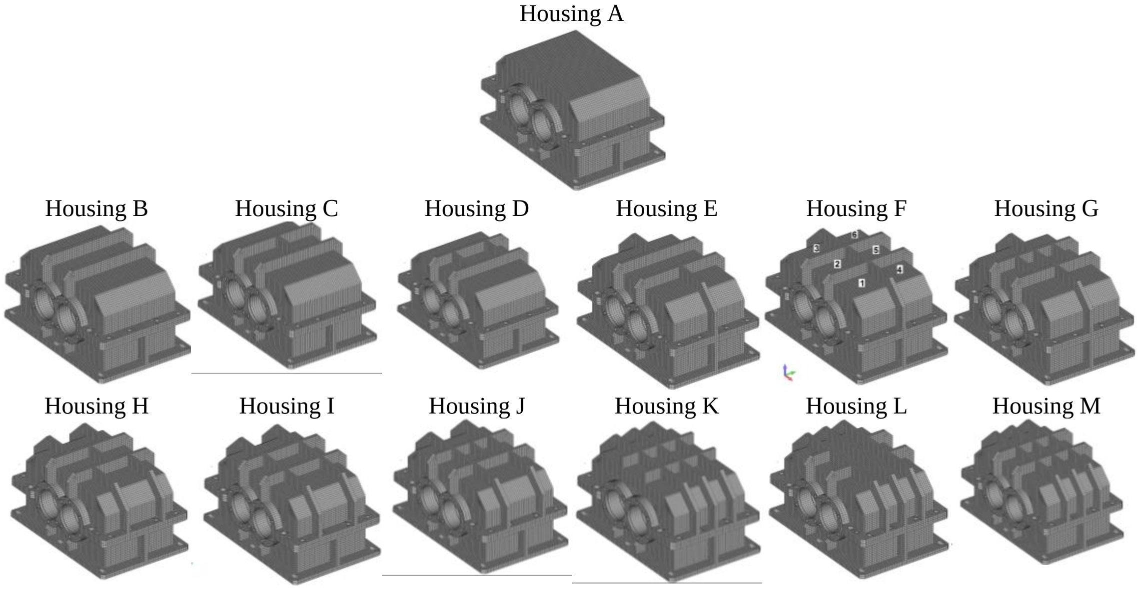
3. MODEL OF THE GEARBOX HOUSING

Different engineering solutions of the gearbox housing were modelled numerically using FEM. The housing solutions adopted, differing in the form of ribbing, are presented in Table 1. The housings were assumed to be divided in the wheel axis plane and made from welded steel sheets, 6 mm thick. Discretization of the models was performed using 8-node solid elements. Threaded connectors were applied in the models to connect the upper and lower parts of the housing, and the contact between them was modelled.

In studies [2, 6] experimental and theoretical modal analyses were made for the adopted engineering solutions for housings A, B and F. The experimental modal analysis performed for different variants of the housing enabled an assessment of the impact of the modifications introduced on the forms and frequencies of free vibration of the housing. The results of the studies enabled determining the degree of compatibility of FEM modal parameters with the parameters of real objects by determining the value of the MAC (Modal Assurance Criterion) factor. On the basis of the results of the experimental modal analysis, numerical models of housings were tuned accordingly. The compatibility of the results of the experimental and theoretical modal analyses of housing solutions allowed confirming the accuracy of the FEM models elaborated [2, 6].

Table 1

FEM models of gearbox housing



In order to further verify the tuned FEM models of the gearbox housing, stand tests and numerical tests were performed which resulted in determining the values of normal vibration velocity in six selected measuring points on the upper cover of the gearbox housing. The compatibility of results of the active and numerical experiments made for three designs of gearbox housings enabled confirming the correctness of the FEM models elaborated.

4. RESEARCH RESULTS

Computations were based on sets of input data, consisting of the information on all the parameters included in the dynamic model (which were defined during the identification process carried out at two test stands).

- Tip and root relief

The influence of the tooth modification (e.g. according to Maag) may be evaluated by comparing the calculation results presented in figures 4 and 5. As seen from these figures, the tip and root relief reduces the dynamic load coefficient K_d [4] of the gears accuracy class 5 to 9.

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

The reduction is several dozen percent over the whole range of rotational speed investigated.

- Profile deviation

Due to different values of the periodic deviation of tooth profile form of the mating teeth, the total deviation may be positive or negative. The sign of the deviation has a significant impact on the dynamic loads in the spur gear, as shown in figures 6 and 7.

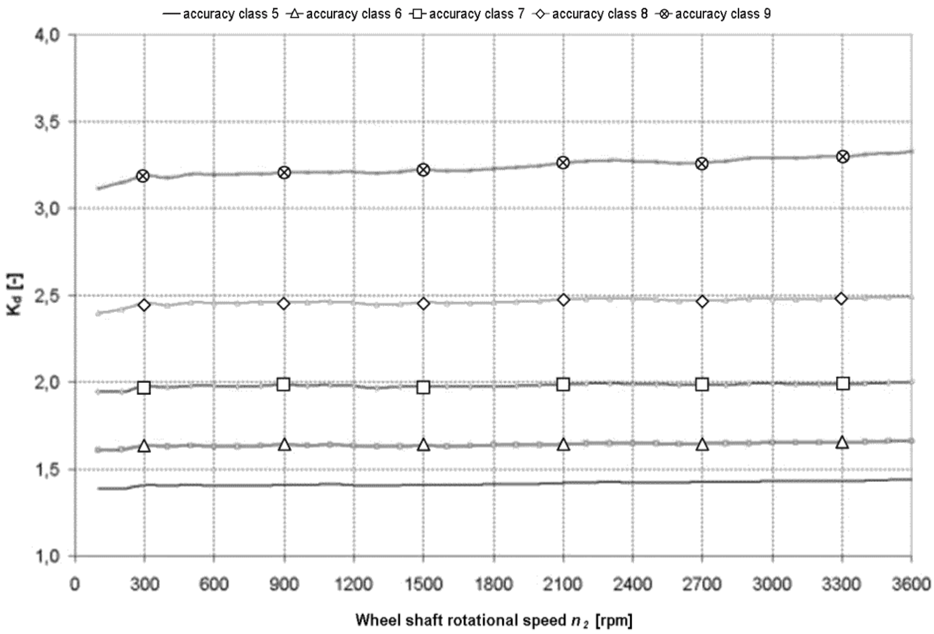


Fig. 4. The relationship of the dynamic load coefficient K_d on the rotational speed n_2 of the shaft of a wheel and on the accuracy class (non modified spur gears, unit load Q = 2 MPa)

Rys. 4. Zależność współczynnika obciążeń dynamicznych K_d od prędkości obrotowej wału koła n_2 oraz klasy dokładności (niemodyfikowana para kół zębatych, obciążenie jednostkowe Q = 2 MPa)

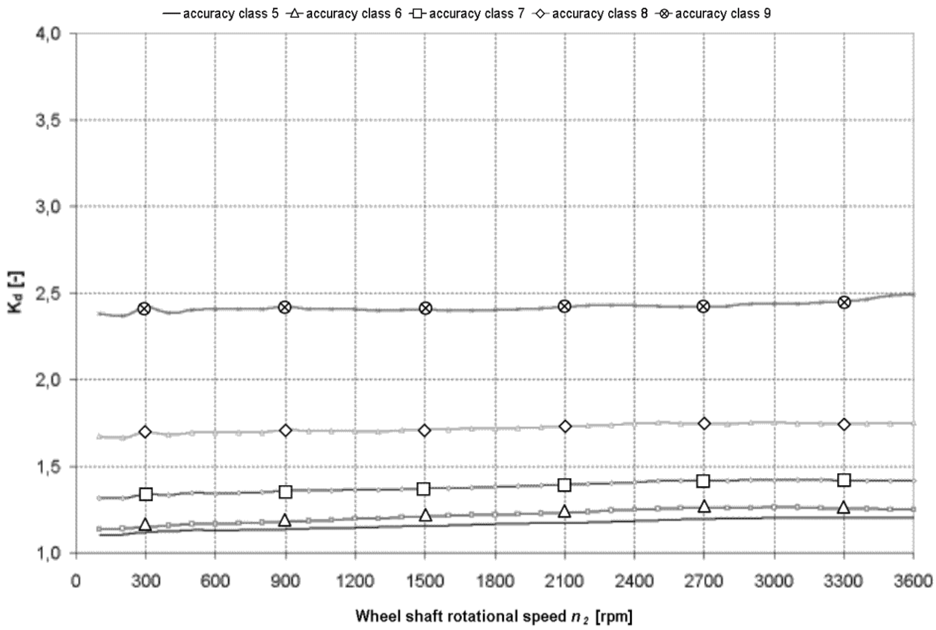


Fig. 5. The relationship of the dynamic load coefficient K_d on the rotational speed n_2 of the shaft of a wheel and on the accuracy class (according to Maag modified spur gears, unit load Q = 2 MPa)

Rys. 5. Zależność współczynnika obciążeń dynamicznych K_d od prędkości obrotowej wału koła n_2 oraz klasy dokładności (modyfikowana para kół zębatych-wg Maag, obciążenie jednostkowe Q = 2 MPa)

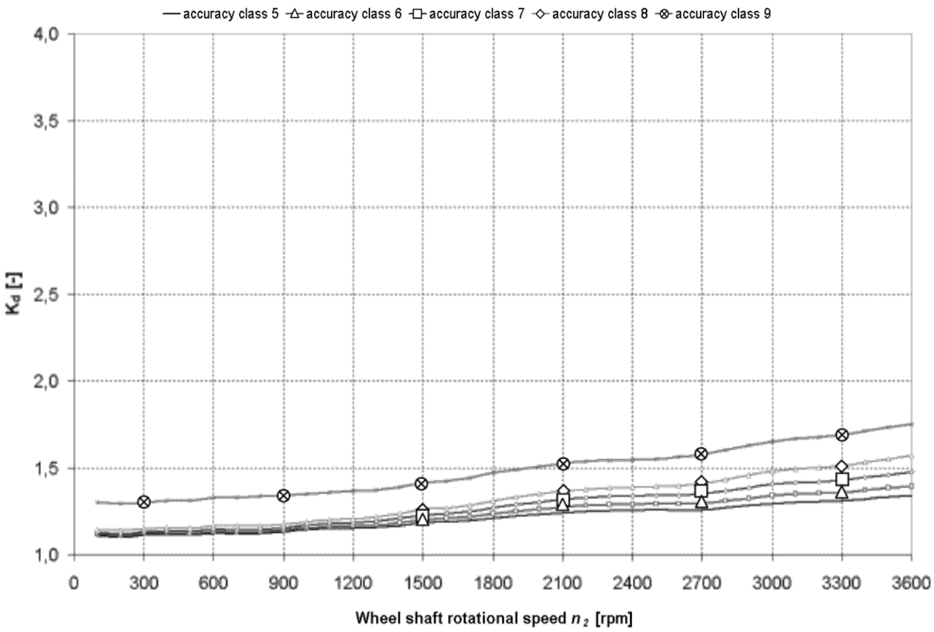


Fig. 6. The relationship of the dynamic load coefficient K_d on the rotational speed n_2 of the shaft of a wheel and on the accuracy class (spur gears with positive value of the profile deviation, Q = 3 MPa)

Rys. 6. Zależność współczynnika obciążeń dynamicznych K_d od prędkości obrotowej wału koła n_2 oraz klasy dokładności (koła zębate o zębach prostych z dodatnią wartością odchyłki zarysu, Q = 3 MPa)

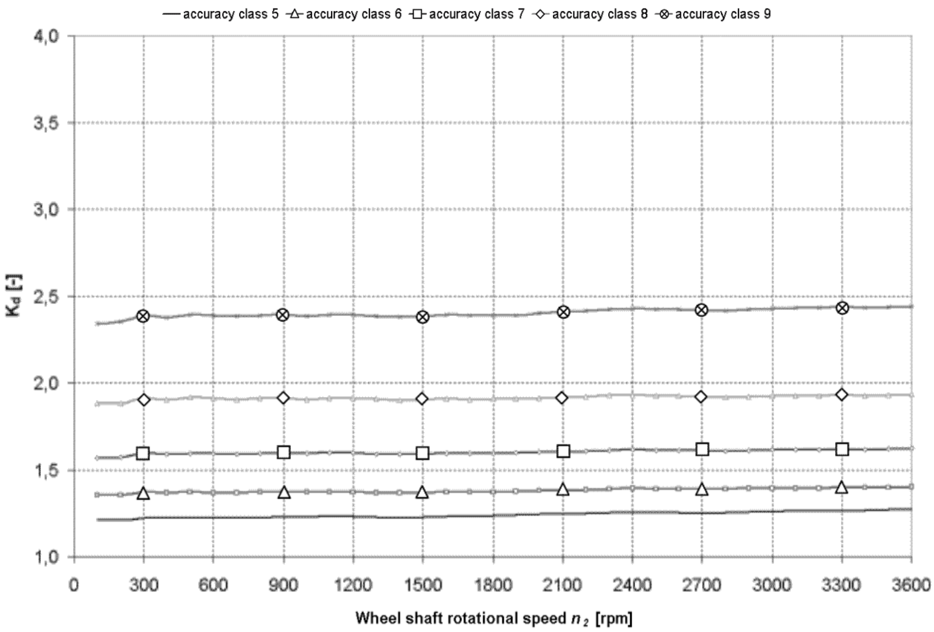


Fig. 7. The relationship of the dynamic load coefficient K_d on the rotational speed n_2 of the shaft of a wheel and on the accuracy class (spur gears with negative value of the profile deviation, Q = 3 MPa)

Rys. 7. Zależność współczynnika obciążeń dynamicznych K_d od prędkości obrotowej wału koła n_2 oraz klasy dokładności (koła zębate o zębach prostych z ujemną wartością odchyłki zarysu, Q = 3 MPa)

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 the previous researches of the authors.

- Ribbing shape

The adopted measure of vibroactivity of the housings [4] was the value resulting from the following dependence:

$$v_{me}^{2}=\frac{1}{n}\sum_{i=1}^{n}\sum_{j=k}^{l}(v_i(f_j))^2 \tag{2}$$

where:
n - the number of the measurement points adopted (n=6), k - lower range of the frequency analyzed, l - upper range of the frequency analyzed, v - vibration velocity, f -vibration frequency

In the calculations, a single impulse induction was applied in the bearing assemblies. Using the MSC Nastran software, the values of normal vibration velocities of some points on the upper cover of the housing were determined. The FEM models were developed for all the adopted ribbing designs (Table 1). Table 2 shows the percentage of changes in the weight of the analyzed housings compared to the housing without ribs (housing A).

Table 2

Change in % of the weight of housings in comparison with the housing without ribs

Housing solution	**B**	**C**	**D**	E	F	G	**H**	**I**	**J**	K	L	M
[%]	**4,62**	**5,22**	**5,81**	7,65	8,26	9,44	**11,29**	**11,89**	**12,48**	13,90	14,24	15,56

The changes in the value of the measure proposed, presented in dB, relating to the housing before modification (housing A) in the entire analyzed range of frequency (0 – 5500 Hz) for all of the analyzed engineering solutions of the upper cover of the gearbox housing are shown in fig. 8.

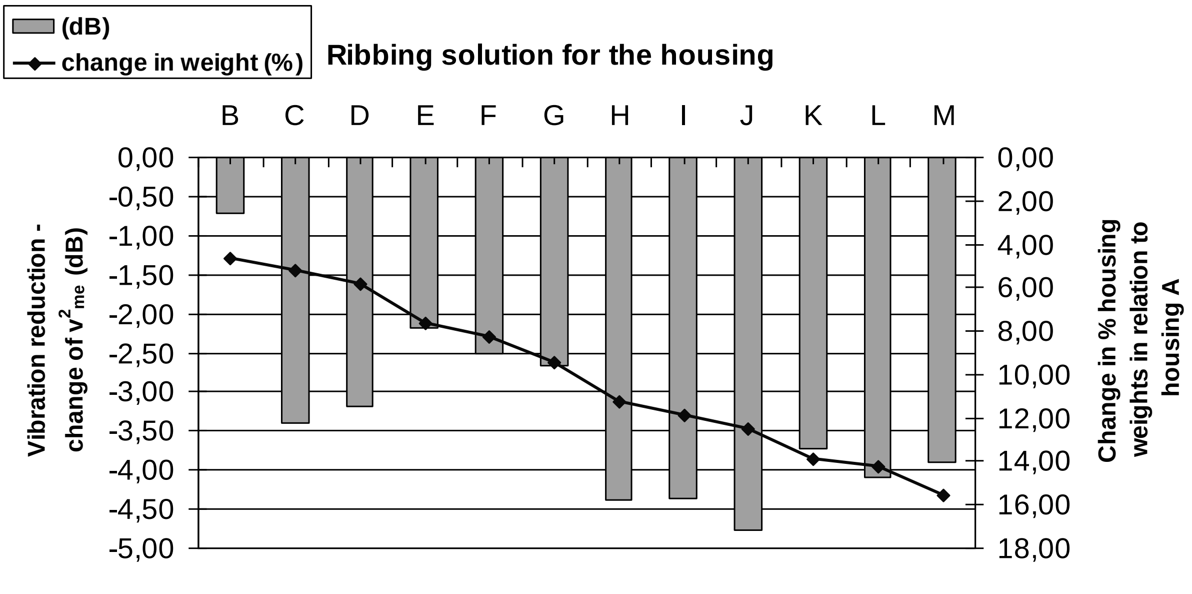


Fig. 8. The changes of the averaged value of vibration velocity v^2_{me} depending on the solution of ribbing of the housing in relation to housing A in the whole adopted range of frequencies

Rys. 8. Zmiany uśrednionej wartości prędkości drgań v^2_{me} w zależności od rozwiązania użebrowania korpusu w odniesieniu do korpusu A w całym przyjętym zakresie częstotliwości

5. CONCLUSIONS

Presented in this paper selected results of simulations are intended to provide basic computing capabilities of dynamic model, which was developed and described in [5]. The conducted comprehensive analysis of the impact of various constructional, technological and operational factors on the vibroactivity of the toothed gear was included in the work [5].

The results confirm a significant influence of additional ribbing of the gearbox housing on the vibration it generates. A significant decrease in the value of the measure proposed (ca. 4.5 dB) was obtained by a simultaneous application of two additional, parallel stiffening ribs and one rib perpendicular to the axis of the shafts (housings H, I and J). As can be seen in Table 2, the application of such solution of the housing ribbing is associated with an increase of its weight by 11.3-12.5%. At the same time, the application of simple ribbing in the form of two ribs parallel to the wheel axis (housings C and D) results in a reduction of the proposed measure of vibration by ca. 3.2-3.4 dB compared to the housing without ribbing, with a significantly lower increase of the weight by ca. 5.2-5.8%.

The results of the simulation studies described in the paper enable selecting in the designing process a solution of the housing ribbing with reduced vibroactivity, with simultaneously taking account of the weight criterion. Further studies will focus on analyzing the impact of engineering characteristics and wear of gearbox components on vibroactivity of drive systems, using the FEM numerical models of different engineering solutions of their housings.

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