A MULTI-BODY MODEL OF GEARS FOR SIMULATION OF VIBRATION SIGNALS FOR GEARS MISALIGNMENT

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Summary

In the paper a dynamic model of spur gears was developed and compared with an experimental data in case of simulation of vibration signals. Comparison was done on the basis of the signals generated by the model and the real object.

The experiment was conducted on an experimental gearbox named DMG-1, that allows to introduce other failures of gears. During the experiment correct work as well as incorrect work of the gears due to misalignment were investigated. In the study a multi-body dynamics model of the gears was developed. The MSC ADAMS software was used for model development and tests. Simulations were conducted for different operation conditions; two values of rotational speed and loading were studied. Behaviour of signals features was similar in both experiment and model for investigated technical states (correct work and misalignment), this was observable by increase of amplitudes for high order gear meshing frequency harmonics, in the model occurs modulations of meshing harmonics by rotational speed, either.

To sum up, in the study it was shown that it is possible to simulate vibration signals by the model of the gears created in multi-body dynamics software, what is evidenced by proper generation of gear meshing frequency harmonics, dependent on rotational speed. For simulation of misalignment the model exhibits greater sensitivity by bigger gain of examined parameters.

Key words: rotating diagnostics, modelling and signal processing for CM, diagnostics of mechanical systems/machines/components.

MODEL KÓŁ ZĘBATYCH DO SYMULACJI SYGNAŁÓW WIBRACYJNYCH DLA WSPÓPRACY KÓŁ Z PRZEKOSZONYMI OSIAMI

Streszczenie

W artykule został przedstawiony oraz porównany z danymi eksperymentalnymi model dynamiczny kół zębatych. Porównanie zostało przeprowadzone na podstawie sygnałów generowanych przez model i obiekt rzeczywisty dla różnych stanów technicznych.

Eksperyment został przeprowadzony na demonstracyjnym stanowisku do badań przekładni zębatych DMG-1, które pozwala na wprowadzenie różnego rodzaju uszkodzeń. Podczas eksperymentu badana była poprawna praca przekładni oraz stan niezdatności związany z przekoszeniem osi jednego z kół. W artykule został przedstawiony model dynamiczny kół zębatych zbudowany na podstawie metody układów wieloczłonowych. W celu budowy modelu i przeprowadzenia testów zostało wykorzystane specjalistyczne oprogramowanie komputerowe. Symulacje zostały przeprowadzone dla różnych warunków pracy: różne wartości prędkości obrotowej oraz obciążenia. Trend zmian cech sygnału wibracyjnego był porównywalny podczas eksperymentu i badań modelowych dla badanych stanów technicznych (poprawna praca, przekoszenie) - wzrost amplitudy drugiej harmonicznej częstotliwości zazębienia. Dla sygnałów otrzymanych z modelu również pojawiły się wstęgi boczne wywołane modulacjami przez prędkość obrotową wałów.

Podsumowując, w artykule została przedstawiona możliwość zastosowania dynamicznego modelu kół zębatych opartego na metodzie układów wieloczłonowych do symulacji sygnałów wibracyjnych. Częstotliwości zazębienia generowane przez model są zgodne z założonymi warunkami pracy (prędkość obrotowa, ilość zębów). Podczas symulacji stanów niezdatności model wykazuje większą czułość na wprowadzone uszkodzenie niż obiekt rzeczywisty.

Słowa kluczowe: Diagnostyka maszyn wirnikowych, Modele dynamiczne kół zębatych, Modelowanie uszkodzeń przekładni zębatych.

INTRODUCTION

Toothed gears are commonly used in various power transmission systems. It is important to collect information about their degradation processes to prevent significant damages during exploitation. Properly registered and processed vibration signals may serve as a source of diagnostic information [2, 3, 4]. In the mechanical diagnostics it is very important to find vibration signature for monitored failures. For classification of vibration signals there are used other techniques e.g. advanced signal processing algorithms or artificial intelligence methods [5, 1]. Some authors [12, 16] focused towards features that are used for the detection of gear faults. In this works authors propose categorization of these features into five different groups based on their pre-processing needs. Modelling the dynamics of gearboxes in useful in condition monitoring, it allows to simulate signals for other states of a gearbox that sometimes cannot be checked in experimental way. There are several modelling methods for simulating behaviour of the gearboxes in case of generated vibration signals: mathematic modelling, computer based finite element methods and multi-body dynamics approach. The multi-body analysis seems to be the most suitable for such analysis, because it allows the time domain integration of the solution, which captures the nonlinear effects of bearing stiffness and clearances, gear backlash, large rotations and other nonlinear phenomena [14]. Specialized multi-body dynamics software like ITI-SIM, SIMPACK, LMS Virtual.Lab Motion and MSC ADAMS allow to model three-dimensional gear bodies, tooth microgeometry, global and local tooth stiffness. One of the study describing multibody approach for simulating vibration behaviour of the gears can be found in [6]. The model permits the dynamical simulation of planetary gearboxes, considering the stiffness characteristics. New approach for modelling gear systems was presented by S. Ebrahimi et al. [7]. It assumes that teeth and the body of gear wheels are rigid but they are connected by elastic elements. In the [14] authors present a methodology for calculation of gear bearing forces, in focus on estimating gear noise. Other paper which treats about multibody dynamic model developed in multi-body dynamics software MSC ADAMS are presented in [8, 11, 15]. Kong et al. [11] presented dynamic simulations of the gears in mesh for other states (as broken tooth). In the [8] three models (an equivalent model, rigid-body model, and frequency-based model) were compared in case of simulation of the meshing forces. The model which assumes nonlinear contact algorithm between teeth of gears and geometric defects of gears like chipped tooth and eccentric tooth was presented in [15]. Authors present behaviour of the model in steady and transition state conditions. Referring to the study [6, 7, 8, 11, 14, 15] there is no presented comparison of the results from computer simulations with the experiment. In the real object there are interferences of vibrations from all power transmission system e.g. bearings, couples or motors. In the articles authors do not present quantitative comparison of the parameters of the signals for correct and incorrect work of the gearbox which is useful in condition monitoring.

In the presented study a dynamic model of spur gears was presented. The model was developed in multibody dynamics software MSC ADMAS. The purpose of the modelling was to simulate vibration signals for misalignment of gears. Misalignment is one of the most common gears manufacturing failure, it is associated with increased noise and vibration level, as well as wear of gear teeth and bearings failures. Gear misalignment excites second order or higher gear meshing frequency harmonics [13, 17]. The model was validated with experimentation on the basis of features extracted from vibration signals.

1. THE EXPERIMENT

The experiment was conducted in the Laboratory of Mechanical Diagnostics at the AGH University of Since and Technology in Cracow. The workbench consists of a motor, elastic clutch, two pairs of gears with gear ratio 1:1 and hydraulic pump 5 MPa. There is possible to manipulate rotational speed by steering system of the motor as well as changing value of the loading by the hydraulic pump. The experimental gearbox allows to introduce other failures of gears such as misalignment, increased centre distance of gears and pitting. In the experiment acceleration signals were registered on the bearing support in the vertical direction, the PCB 356A15 accelerometer has been used. A laser tachometer was used for measuring rotational speed. All the signals were measured by National Instruments C Series module 9223, with a sample rate up to 1 MS/s and simultaneous analog-to-digital converters. The measuring system is presented on the fig. 1.

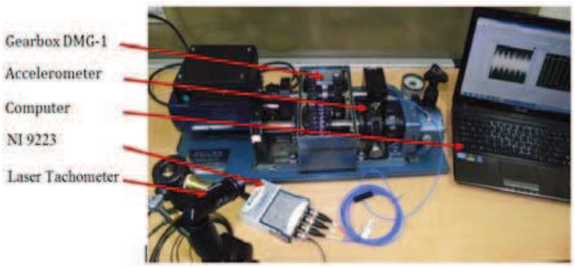


Fig. 1. Experimental gearbox and measuring system

During the experiment, correct as well as incorrect work of the gears due to misalignment were studied. All measurements were conducted in the steady-state conditions for other rotational speeds (400 and

800 RPM) and other values of the loading (1 and 3 MPa). The registered signals were filtered by cascade stop-band filter in such way to remove all frequencies, beside of first five meshing frequency harmonics. The width of the filter was set up to include second order sidebands.

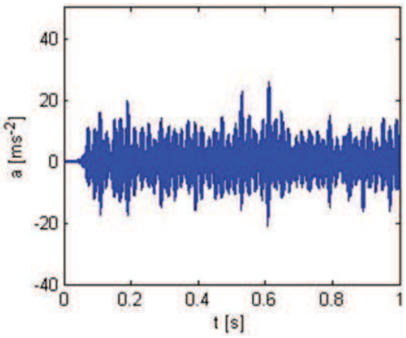


Fig. 2. Signal measured in the experiment for correct work of the gears, with high loading and rotational speed 800 RPM

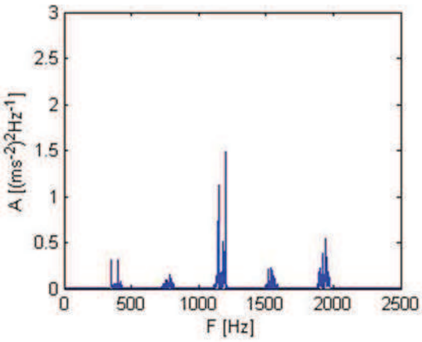


Fig. 3. PSD for signal measured in the experiment for correct work of the gears, with high loading and rotational speed 800 RPM

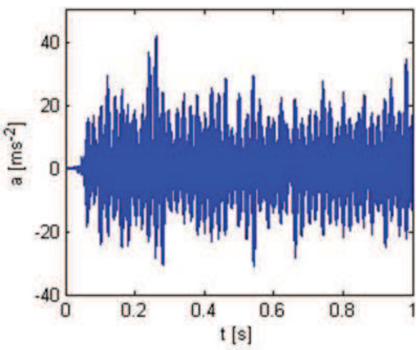


Fig. 4. Signal measured in the experiment for incorrect work due to misalignment of the gears, with high loading and rotational speed 800 RPM

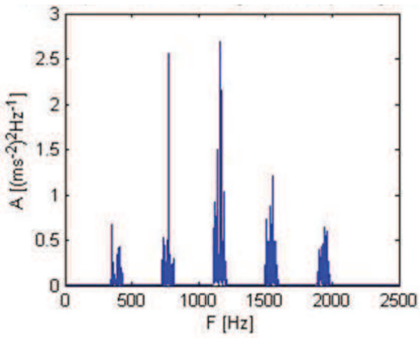


Fig. 5. PSD for signal measured in the experiment for incorrect work due to misalignment of the gears, with high loading and rotational speed 800 RPM

On the figures above example of the signals from the experiment were presented; the fig. 2 and fig. 4 present time signal for correct and incorrect state of the gears, the Power Spectral Density (PSD) of these signals was presented on the fig. 3 and fig. 5. On the presented figures it is observable that amplitude of the second meshing frequency harmonic significantly increase for misalignment.

2. THE MODEL OF THE GEARS

In the study a model of the gears was built in the multibody dynamics software MSC ADAMS. Firstly three dimensional CAD model of the gears was developed in the CATIA software. Next the CAD model was transferred into the ADAMS environment. In the multi-body dynamics software the model of the gears was properly constrained and the forces were applied. The parameters of the gears were presented in the table 1.

Table 1. Gear parameters.

Parameter	Symbol	Value
Teeth	z1, z2	29
Modulus	m	3.065
Pressure angle	a	20 [deg]
Pitch circle radius	rp	44,435 [mm]
Outer circle radius	ra	47,5 [mm]
Base circle radius	rb	41,755 [mm]
Root circle radius	rf	40,605 [mm]

Two variants of gears dynamics modelling can be distinguished; models where all phenomena occurring in a power transmission system are included and models which take into account only phenomena inside of a gearbox. In the first kind of models dynamics of motor, couples, gears and working machine are included. Other approach considers physical phenomena occurring only inside of a gearbox, the time varying stiffness, technology of gears mostly affect on dynamics. This approach is used for calculating of dynamic forces in teeth or to identify indicators of teeth failures [18]. In the real object meshing stiffness vary in time, it depends from number of intermeshing teeth and deflection of a tooth by the action of normal force during meshing. Across of a path of action stiffness is changing as a parabola function [18, 19]. In the presented model meshing stiffness depends on number of teeth in contact and parameters of contact algorithm which assume elastic contact between teeth. Depending from backlash teeth may lose contact or work on opposite sides, this may induce large impact forces associated with consecutive single-sided or double-sided impacts, wrongly designed backlash may cause teeth interference and undercut [18]. On the fig.6 model view in ADAMS software was presented.

Fig.6. CAD model of gears

In the model gears and shafts are considered as rigid bodies, contact surfaces between teeth are flexible, the backlash between gears was assumed, either. The fixed joints between gears and shafts and revolute joint between shafts and ground were used. Constant rotational motion was applied on revolute joint on the shaft of the active gear and a resistive torque on the shaft of the passive one. The model assumes ideal involute teeth geometry. On the fig. 7 the model was presented.

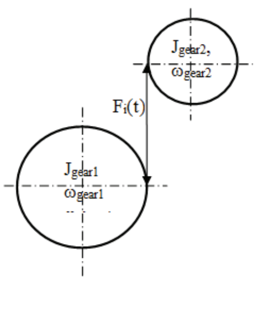


Fig.7. Model of gears

To simulate vibration signals during meshing, contacts and Coulomb friction between teeth of two gears were assumed. For contacts model the MSC ADAMS Impact algorithm was chosen, because of its robustness in numerical integration [15]. Equation (1) describes a contact force in the ADAMS Impact algorithm. The contact force is composed of two parts: the elastic component and the damping force, which is a function of the contact-collision velocity. By the definition of the step function (2), the damping force is defined as a cubic function of penetration depth [11].

$$F = \begin{cases} K(x_0 - x)^e + CS\dot{x} & x < x_0 \\ 0 & x \geq x_0 \end{cases} \tag{1}$$

$$S = \begin{cases} 0 & x > x_0 \\ (3 - 2\Delta d)\Delta d^2 & x_0 - d < x < x_0 \\ 1 & x \leq x_0 - d \end{cases} \tag{2}$$

In the equation (1-2) $\Delta d = x_0 - x_1$ describes deformation of a body, K contact stiffness, e force exponent, C damping parameter and penetration depth was assigned by d. Stiffness between teeth pair in contact can be described by the Hertz elastic contact theory (3). In this model stiffness was described by a pair of ideal cylinders in contact [9,11].

$$\begin{cases} K = \frac{4}{3} R^{\frac{1}{2}} E^* = \frac{4}{3}\left[\frac{i d_1 \cos(\alpha_t)\tan(\alpha'_t)}{2(1+i)\cos(\beta_b)}\right]^{\frac{1}{2}} E^* \\ \frac{1}{E^*} = \frac{1-v_1^2}{E_1} + \frac{1-v_2^2}{E_2} \\ \beta_b = \operatorname{atan}(\tan\beta \cos\alpha_t) \end{cases} \tag{3}$$

In the equation (3) the R describes equivalent radius of two contacting bodies, E^* equivalent Young's modulus of two contacting bodies, i gear ratio, d_1 diameter of standard pitch circle, α_t', α_t transverse pressure angle at engaged and standard pitch circle , β, β_b helical angle at the pitch and base circle, v_1, v_2, Poisson ratio of the pinion and gear. The Young's modulus of pinion and gear are described by E_1, E_2, respectively. In the model it was assumed that the gears are made from steel with Young modulus $2.1 \cdot 10^{11}$ Pa, and Poisson ratio 0.3. The equivalent Young modulus for two bodies in contact $1.153 \cdot 10^{11}$ Pa. In the contact algorithm the damping coefficient was assumed as 0,1% of contact stiffness. Determination of the force exponent was based on the experiment. The material of the gears is alloy steel the parameters for Coulomb friction were taken from the literature [11, 15]. Parameters used in ADAMS Impact algorithm are presented in table 2.

Table 2. Contact parameters for MSC ADAMS.

Parameter	Symbol	Value
Contact stiffness	K	$2.9852 \cdot 10^5$ [N/mm$^{3/2}$]
Damping coef.	C	3000 [Ns/mm]
Force exponent	e	1,3
Static friction coef.	υ_s	0,1
Static transition vel.	V_s	1 [mm/s]
Dynamic friction coef.	υ_d	0,08
Friction transition vel.	V_f	10 [mm/s]

For dynamic simulation of the model the integrator WSTIFF and the Stabilized Index-2 (SI2) formulation were chosen. The integrator WSTIFF allows control the numerical integration of the equations of motion for a dynamic analysis [10]. Parameters of simulations are like follows: number of steps 6000, simulation time 1s, rotational speeds 400RPM and 800RPM. Two values of torque were used during simulations 100 and 200 Nm. During the simulation force signal was measured in vertical direction between gear and shaft. On the fig. 8 and 10 signals obtained from the model tests were presented.

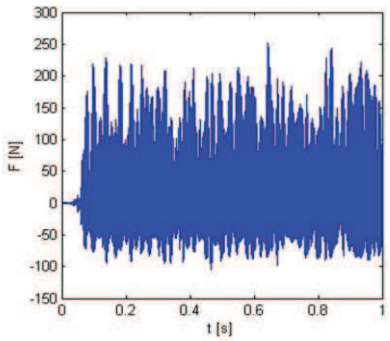


Fig. 8. Signal from model for correct work of the gears, with high loading and rotational speed 800 RPM

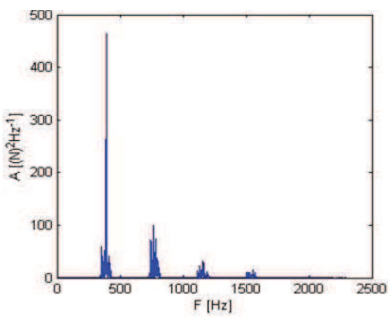


Fig. 9. PSD for signal from model for correct work of the gears, with high loading and rotational speed 800 RPM

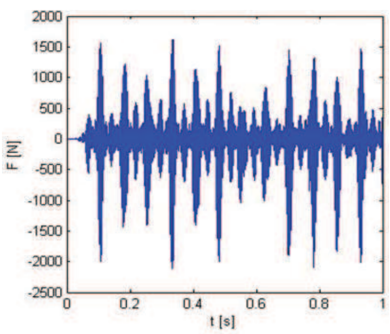


Fig. 10. Signal from model for incorrect work due to misalignment of the gears, with high loading and rotational speed 800 RPM

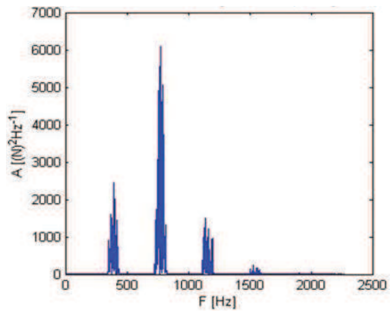


Fig. 11. PSD for signal from model for incorrect work due to misalignment of the gears, with high loading and rotational speed 800 RPM

On the plots above force signals between gear and shaft during the time of simulation are shown, as well as spectra of this signals. The signals were filtered by cascade pass-band filters in the same way as in the experiment; to receive only bands with harmonics of the meshing frequency. The location of the meshing frequency harmonics (fig. 9 and 11) is compliant with theory (4).

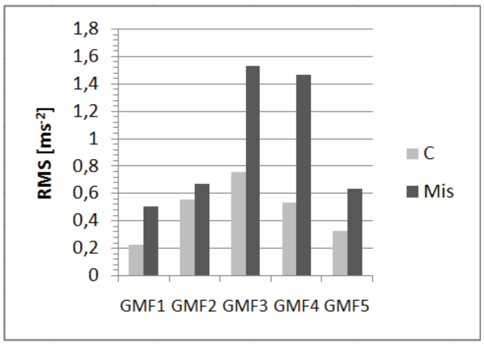
$$F_m = z * F_{rot} * n. \qquad (4)$$

In the equation (4) F_m meshing frequency harmonic, z number of teeth, F_{rot} rotational frequency and n harmonic number. For gears with misalignment of the axes on the spectra, modulations of meshing frequency harmonics by rotational speed were observable, also there was significant increase of the second meshing frequency harmonic.

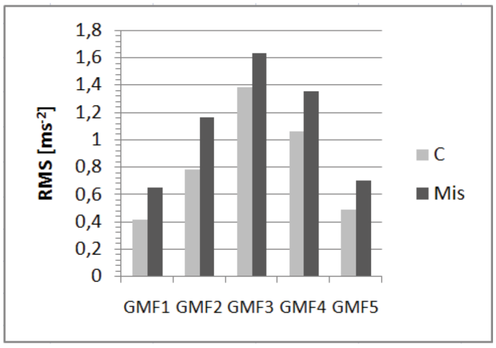
3. ANALYSIS OF THE RESULTS

In this chapter comparison of estimates of the signals from the experiment and the model was presented. Comparison has been done on the basis of energy value estimated in the bands for meshing frequency harmonics [12, 16]. Incorrect work of gears was due to misalignment of one gear axes about 1 degree. Measurements have been done for two rotational speeds: 400 RPM, 800 RPM and for case with low and high loading. In the study the signals were filtered by pass-band filter to receive signal containing particular meshing frequency harmonic and its second order modulation sidebands; in the study first five meshing frequency harmonics were considered. On the fig.12 the RMS values in the bands were presented, for signals measured during the experiment.

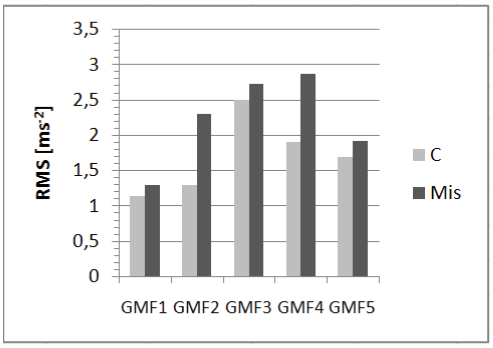
a)



b)



c)



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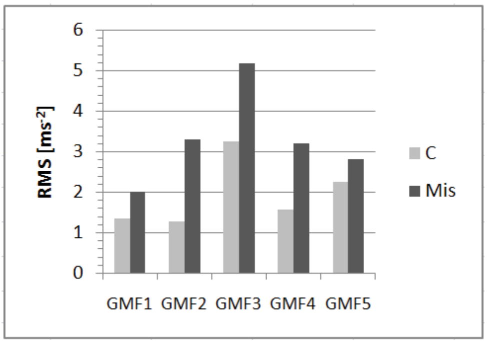
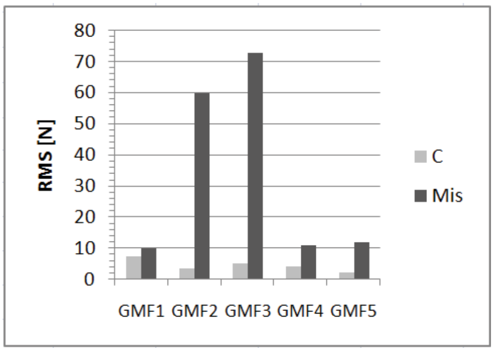


Fig. 12. RMS in bands for signals from experiment, a) 400RPM, low loading, b) 400 RPM, high loading, c) 800 RPM, low loading, d) 800 RPM, high loading. (C- correct state, Mis–misalignment)

a)



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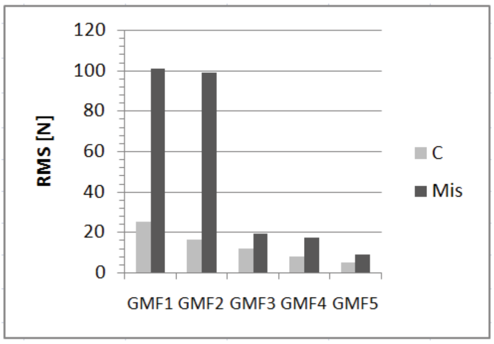

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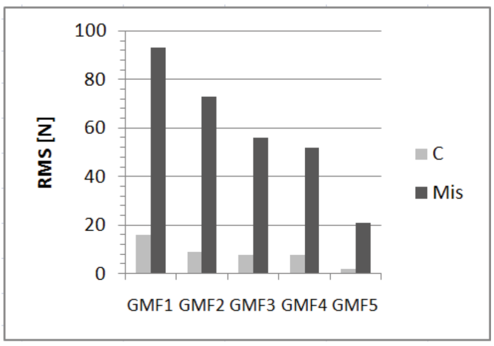

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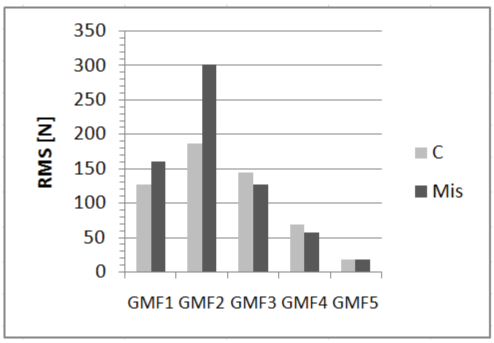

Fig. 13. RMS in bands for signals from model, a) 400RPM, low loading, b) 400 RPM, high loading, c) 800 RPM, low loading, d) 800 RPM, high loading. (C- correct state, Mis–misalignment)

It is observable that energy in bands increases for misalignment, particularly for higher harmonics. For experiment with low rotational speed increase of RMS about 20% was observable for misalignment of the gears comparing to correct state, fig. 12a) and fig. 12b). For second rotational speed, fig. 12c) and fig. 12d), the increase of energy in bands (about 30%) due to misalignment was observable, especially for GMF2, GMF3 and GMF4. For the model simulations behaviour of the examined feature was similar, fig 13. The significant gain of signal energy in case of gears misalignment can be observed. In every case there is significant increase of the second GMF.

To sum up, in the study it was shown that it is possible simulation of gears vibration signals by model of the gears created in multibody dynamics software, what was evidenced by proper generation of GMF harmonics. For simulation of gears manufacturing errors (misalignment) the model exhibits greater sensitivity than real object; bigger gain of RMS values in the bands due to misalignment comparing to the experiment.

3. SUMMARY

In the presented study the dynamic model of the spur gears was developed and compared with an experimental data in case of simulation of vibration signals for other technical states of the gears. In the paper manufacturing failure as misalignment of gears' axes were studied. To build and simulate the model of the gears MSC ADAMS software was used, simulations were conducted for different operation conditions of the gears. The model allows simulation of the force signals between shaft and gear during meshing, in the experiment the acceleration signals on the bearing support were measured, nevertheless indicators for misalignment were similar in both cases. It is observable by increase of amplitudes of higher order GMF both in the experiment and simulation, there also occurs modulations of GMF by rotational speed frequency in case of misalignment. The model exhibits greater sensitivity for misalignment what can be seen by greater increase of analyzed estimators comparing to correct work of the gears. This behaviour could be explained by simplifications assumed in the model like: lack of lubrication and application of rigid bodies for gears and shafts.

Presented three-dimensional model allows for simulation of vibration signals during meshing of the gears. In distinction from study conducted so far [6, 7, 8, 11, 14, 15] in the paper comparison of the model with experimental data was presented. The presented model should be improved. Instead of rigid-elastic model, fully elastic model could be build by application of flexible bodies, this allows to take into account deformation of a shaft and teeth in mesh. In the future model bearings should be included, either. Such improved model could be used for identification of vibration signature of gears failures in high-power machinery, what would be useful in condition monitoring.

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