IDENTIFICATION AND VERIFICATION OF SIMULATION MODEL OF GEARS WORKING IN CIRCULATING POWER SYSTEM

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Summary

This article presents executed stages of preparing and identification a power circulating gear testing machine. The purposefulness of modelling this test-stand was justified by its popularity and universal character that enabled to apply it in many laboratory research works. It was assumed that property defined and next identified model would be used to analyze dynamic phenomena in meshing and bearings of examined gears and it enabled to optimize their construction, especially to minimize their vibroactivity.

The described model combines advantages of two main modelling directions: it is characterized by very accurate analyzes of a model of toothed wheels pair, that includes nonlinear description of properties of both gears meshing in a test-stand (examined and closing), as well as it includes the influence of remaining system's elements, which enables to examine the dynamics of a system as a whole.

Calculation algorithm additionally includes possibility to simulate faults of wheels in the examined gear, such as chipping of tooth tips, tooth root crack together with possible faults of rolling bearings elements.

The article includes description of the construction of a test-stand, dynamic system model with detailed parameters as well as short characteristics of the simulation program and purposes of its main modules. Next the research works, which aimed at identification of selected parameters of the model, were presented. Afterwards it was verified whether the model was correctly fine tuned by means of measuring the velocity of transversal vibrations at the test-stand using Ometron VH300+ laser vibrometer, and results of laboratory examinations were compared with the results achieved during computer simulation.

Keywords: gearbox, power circulating gear testing machine, dynamic model.

IDENTYFIKACJA I WERYFIKACJA MODELU SYMULACYJNEGO PRZEKŁADNI ZĘBATYCH PRACUJĄCYCH W UKŁADZIE MOCY KRĄŻĄCEJ

Streszczenie

W artykule przedstawione zostały zrealizowane etapy opracowywania i dostrajania modelu stanowiska z przekładniami zębatymi pracującymi w układzie mocy krążącej (FZG). O celowości modelowania tego stanowiska zadecydowała jego popularność i uniwersalność, pozwalająca wykorzystać je w wielu badaniach laboratoryjnych. Założono, że poprawnie zdefiniowany a następnie zidentyfikowany model, zostanie wykorzystany do analizy zjawisk dynamicznych w zazębieniach i łożyskach badanych przekładni oraz pozwoli na optymalizację ich konstrukcji, w szczególności w kierunku minimalizacji ich wibroaktywności oraz rozbudowy bazy wiedzy diagnostycznej.

W pracy opisano budowę stanowiska, przedstawiono jego model dynamiczny z wyszczególnieniem uwzględnionych parametrów oraz krótką charakterystykę programu symulacyjnego i przeznaczenie jego głównych modułów. W dalszej części przedstawiono badania, mające na celu identyfikację wybranych parametrów modelu. Poprawność dostrojenia modelu została następnie zweryfikowana poprzez pomiar prędkości drgań poprzecznych wałów na stanowisku doświadczalnym z użyciem wibrometru laserowego Ometron VH300+, a wyniki badań laboratoryjnych zostały odniesione do wyników uzyskanych drogą symulacji komputerowej.

Słowa kluczowe: przekładnia zębata, stanowisko FZG, model dynamiczny.

1. INTRODUCTION

Many research centres deal with modelling of gears, as they are found to be useful tools for designing and for a dynamic analysis of power transmission systems.

The research works, described, inter alia, in [1, 2, 4, 8, 9], are facilitated by continuous development of analytical equipment and software which make it possible to enrich models with further, previously not taken into account, factors.

Creating models of devices already at their designing stage allows meeting the requirement of production cost minimization and, above all, optimizing the construction in terms of its durability, reliability and functionality, optimizing thereby the service costs.

Such direction in designing devices is facilitated by defining the functional changes which result from changes in the condition of the device during its service.

By using models of gears, it is possible to make complicated dynamic analyses to identify symptoms of faults in gears, which analyses could not be carried out in real conditions.

Computer simulations allow us to considerably reduce the time and cost of examination, ensuring, at the same time, stability of the conditions in which the examination is carried out.

The mentioned advantages of using models justify the usefulness of research on models. They also effectively facilitate reduction of the number of stand tests.

The correctness of tests performed with use of a model results from its proper identification. Since a model always constitutes certain simplification of the actual object, which results from the fact that it is impossible to take into account all parameters of the investigated system, those quantities whose influence on the investigated phenomenon is very small, should be left out. This will allow avoiding mistakes, shortening of calculation time and obtaining results which are qualitatively and quantitatively comparable to the results of measurements made on a real object.

The developed model combines the advantages of two main directions in modelling: it is characterized by a precise analysis of a model of a pair of toothed wheels, with taking into account a non-linear description of meshing properties of both gears in a stand (the tested and closing gears). In addition, it takes account of the influence of other elements of the system and enables testing the dynamics of a power transmission system as a whole.

2. DESCRIPTION OF THE TEST STAND

The test stand with gears working in a circulating power system (Fig. 1) consists of an electric motor, which through a belt transmission (1) drives the closing gear (2) and, connected to it by means of a torsional shaft (3) and coupling shafts (4), the tested gear. The tests may be carried out at different rotational speeds, the change of which is made smoothly by means of a frequency converter through which the motor is powered, and at a load controlled by means of torsional shafts, a tightening clutch (6) and a lever with weights. The closing and tested gears have identical ratios and identical axle bases [2, 4].

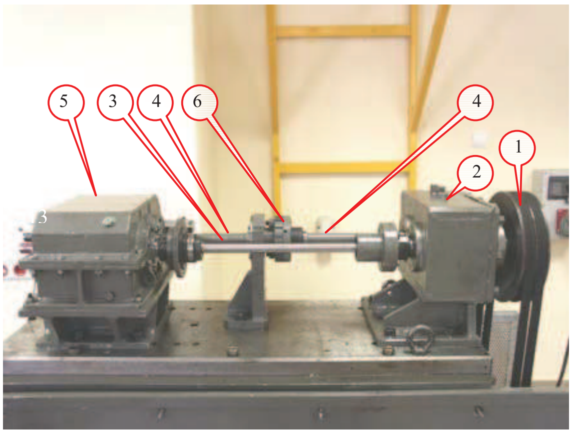

Fig. 1. Test stand

3. DYNAMIC MODEL

In a dynamic model of the test stand, the belt transmission connecting the motor with the closing gear was replaced with a tightening clutch. The model shown in Figure 2 allows taking into account, inter alia:

- the changeable rigidity of gear teeth meshing, friction and damping of vibration of meshing gears, deviations in gear pitches and position of the profile of each tooth, (according to PN-ISO 1328-1:2000), clearance between the teeth (j_{n1}, j_{n2}) – the developed description of meshing properties of the closing and testing gears is in line with the modelling direction proposed in [5, 6, 7],
- operation in conditions of a variable rotational speed and torque $M_s(n_s)$,
- angular displacements of the motor rotor (φ_s), tightening clutches (φ_{sp}, φ_{wn}), pinion and wheel of the closing gear (φ_H, φ_K) and tested gear (φ_B, φ_E) around the axis consistent with the direction of the gear shafts' axis,
- displacements in all system bearings in the direction of forces acting between the teeth ($x_A \div x_L$) and friction in meshing ($y_A \div y_L$),
- torsional rigidity of shafts,
- rigidity of supports $c_A \div c_L$,
- damping in bearings $k_A \div k_L$ and shafts,
- inertial mass and moments of inertia of the modelled elements of the stand ($m_1 \div m_4$, J_s, J_{sp}, J_{wn}, $J_1 \div J_4$).

The calculation algorithm additionally takes into account the possibility of simulation of faults in the tested gear, such as a crack at the tooth root or spalling of the tooth crest with possible concurrent occurrence of damage of rolling bearings components.

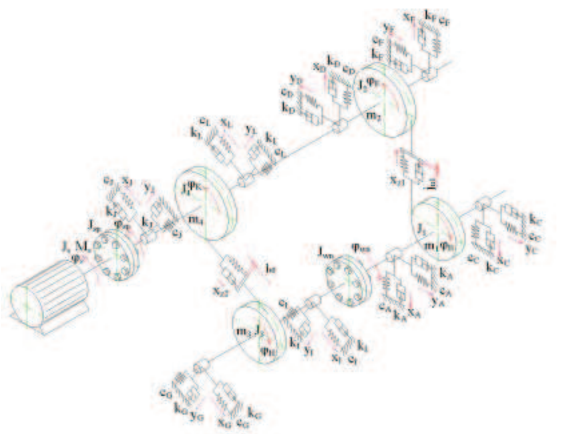

Fig. 2. Dynamic model of the stand

4. SIMULATION PROGRAM

The calculations are carried out using a program developed in the Delphi programming environment. For functional reasons, it has been divided into three parts: entering data and preliminary calculations (Fig. 3) simulation calculations and analysis of results. The stages of carrying out the calculations are schematically shown in Fig. 4.

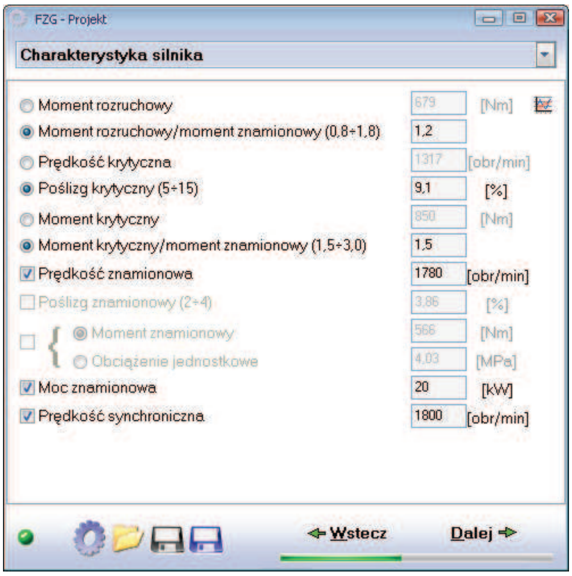

Fig. 3. Module window of data input and preliminary calculations with an active motor characteristics sheet

The Runge-Kutty 4 method was used to solve motion equations. The results of simulation calculations are recorded in a standard format of the Matlab program, what allows their processing using advanced methods offered by this calculation environment.

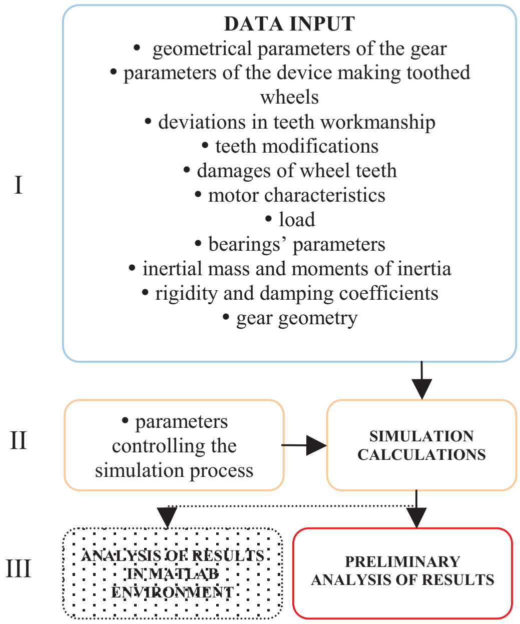

Fig. 4. Stages of simulation calculations

5. MODEL IDENTIFICATION

Application of the model for simulation tests requires previous tuning of its parameters. Only an identified model can be a tool enabling a reduction of the number of stand tests.

The first stage of tuning the model parameters consisted in measuring the geometrical parameters and deviations in pitch of toothed wheels mounted at a laboratory stand and in determining the inertial mass and moments of interia of the modelled components, as well as torsional rigidity of the stand shafts. Basic geometrical parameters of toothed wheels of the stand-tested gear are compiled in Table 1.

Table 1. Parameters of toothed wheels of the tested gear

Number of pinion teeth, z_1	19
Number of wheel teeth, z_2	30
Angle of tooth line inclination, β	15 °
Addendum modification coefficient for the pinion, x_1	0,500
Addendum modification coefficient for the wheel, x_2	0,295
Nominal module, m_n	3,5 mm
Nominal pressure angle, α_0	20 °
Distance between axes of cooperating wheels, a_w	91,5 mm
Meshing width, b	56 mm
Meshing ratio, ε_α	1,32

Using the results of tests presented in [3], the model was tuned in the direction of determining efficiency values similar to those obtained during laboratory tests by means of the heat balance method. Concurrence of the efficiency values obtained from the simulation and from laboratory tests enabled determining the characteristics of friction coefficient in mesh (Fig. 5).

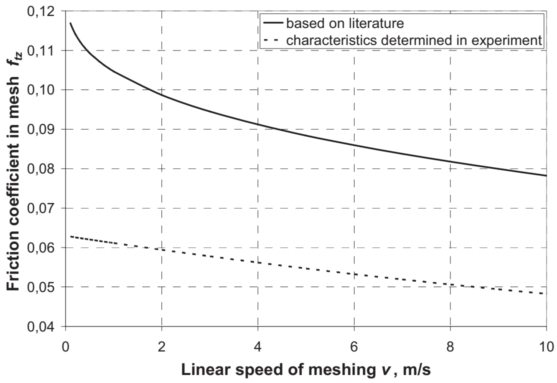

Fig. 5. Characteristics of friction coefficient in mesh

6. VERIFICATION OF THE SIMULATION TEST RESULTS

Verification of the model tuning correctness consisted in comparing the results of measurements of shafts' transverse vibration speed with the results of simulation calculations. The measurements were carried out on FZG stand (Fig. 1) using an Ometron VH300+ laser vibrometer in the direction of force acting between the teeth. Transverse vibration speeds of the pinion and wheel shafts were recorded for both, the tested and closing gear. Measuring points on the tested gear are presented in Fig. 6.

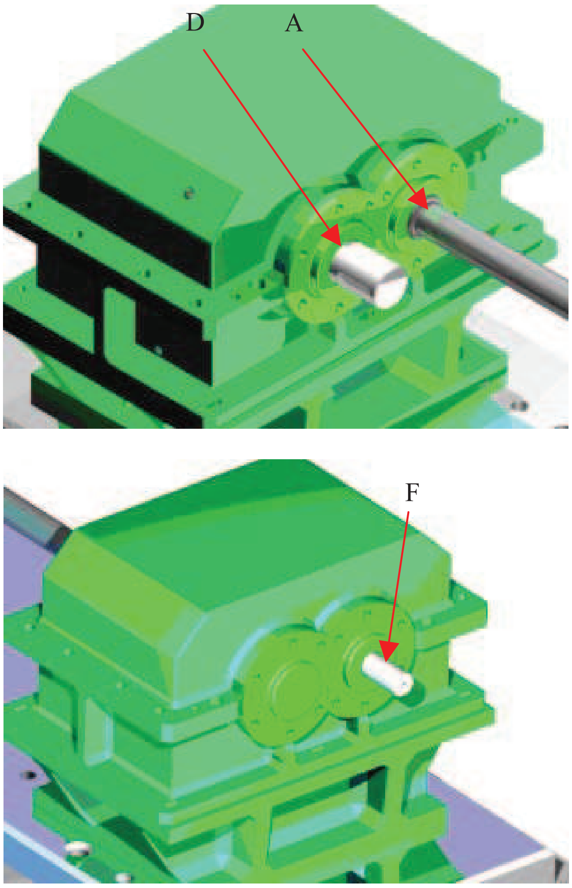

Fig. 6. Measuring points on the tested gear

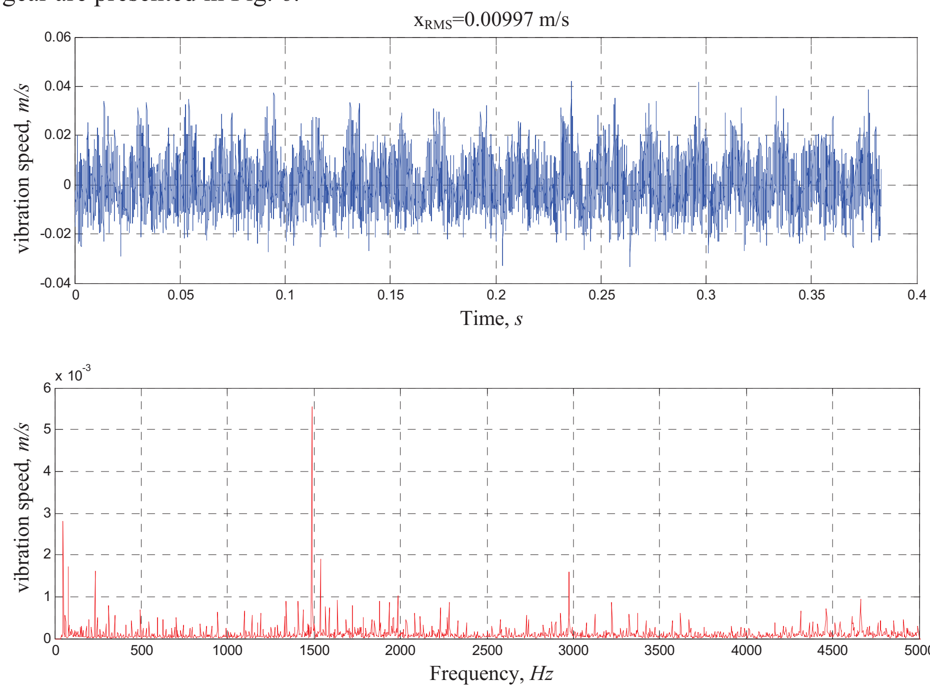

Fig. 7. Time and spectrum of transverse vibration speed of the wheel shaft in tested gear – measurement at a laboratory stand (measuring point F, wheel rotational speed $n_2 \approx 2975$ rpm, load intensity $Q \approx 1.5$ MPa)

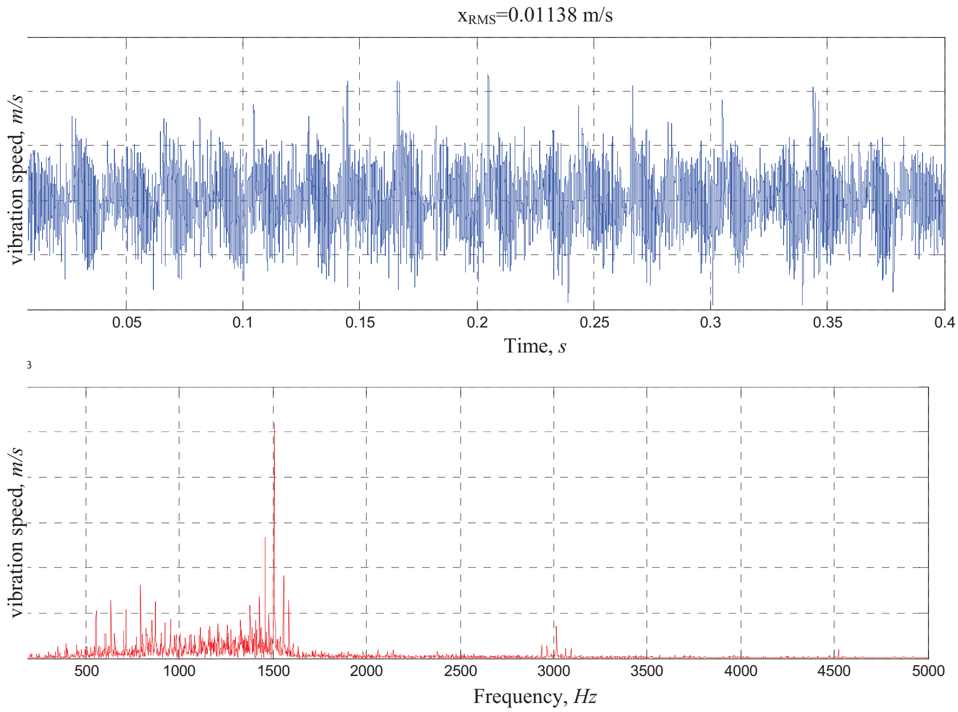

Fig. 8. Time and spectrum of transverse vibration speed of the wheel shaft in tested gear – simulation (measuring point F, wheel rotational speed $n_2 \approx 3013$ rpm, load intensity $Q \approx 1.5$ MPa)

Some examples of the speed and spectra of transverse vibration of the wheel shaft in the tested gear, obtained from laboratory measurements and from the simulation, are presented in Figs 7 and 8, respectively .

7. CONCLUSIONS

The stages of model identification presented using an example of the test stand's testing part more important from the point of view of further research, do not show all difficulties of the process. Obtaining of calculation results qualitatively and quantitatively concurrent with the results of laboratory measurements in a wide range of rotational speeds and loads requires carrying out a number of measurement series.

Since in the model tuning process there was no possibility of making measurement of values most useful for this purpose, e.g. forces acting between the teeth in both gears of forces in the stand bearings, it was necessary to make a large number of simulations, during which a change of selected parameters was applied, until obtaining satisfactory compliance of the results with the measurements made for the same conditions.

Obtaining compliance of the results in time and frequency domains enables taking advantage of the model in further tests and justifies this method of conducting research.

REFERENCES

[1] Dalpiaz G., Rivola A., Rubini R.: *Dynamic modelling of gear systems for condition monitoring and diagnostics*. Kongres Diagnostyki Technicznej, Gdańsk, 1996.

[2] Dąbrowski Z., Radkowski S., Wilk A.: *Dynamika przekładni zębatych. Badania i symulacja w projektowaniu eksploatacyjnie zorientowanym*. Warszawa-Katowice-Radom 2000.

[3] Łazarz B.: *Badania wpływu zużycia zębów kół na obciążenia dynamiczne w diagnostyce przekładni*. Rozprawa doktorska. Politechnika Śląska, Katowice 1996.

[4] Łazarz B.: *Zidentyfikowany model dynamiczny przekładni zębatej jako podstawa projektowania*. Instytut Technologii Eksploatacji, Katowice-Radom 2001

[5] Müller L.: *Nowy model dynamiczny przekładni walcowej o zębach prostych*. Przegląd Mechaniczny nr 21/1974.

[6] Müller L.: *Przekładnie zębate*. *Badania*. WNT Warszawa 1984

[7] Müller L.: *Przekładnie zębate. Dynamika*. WNT Warszawa 1986.

[8] Osiński J.: *Zjawiska dynamiczne w układach z przekładniami zębatymi*. II Ogólnopolski Sympozjum Naukowo – Techniczne „Przekładnie Mechaniczne Specjalne", Warszawa 1997.

[9] Wilk A.: *Wpływ parametrów technologicznych i konstrukcyjnych na dynamikę przekładni o zębach prostych*. Zeszyty Naukowe

Politechniki Śląskiej, Mechanika z. 72, Gliwice 1981.

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