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Use of simulation and laboratory tests to shape the vibroactivity of toothed gears

Abstract

Vibroacoustic phenomena occurring in power transmission systems affect the durability and quality of the driven devices, and also allow for evaluation their technical condition. The article presents the results of laboratory tests aimed at comparing the vibroactivity of the toothed gear, in which the pairs of wheels differ in geometric parameters. The results of research carried on test stand are supplement with results of computer simulations. The proposed test method can be used to determine general vibration and noise reduction guidelines for power transmission systems with toothed gear. The results of the numerical and laboratory tests carried out at the transmission system with circulating power system show that the uncomplicated transformation of the gear geometry used in the toothed gear from the manufacturing point of view allows to reduce the level of vibroacoustic phenomena accompanying their work.

Keywords: vibroactivity, toothed gear, laboratory research, simulation research.

1. Introduction

The vibroacoustic phenomena accompanying the work of the gears and power transmission systems in which they work have many sources. Realization of the purpose function causes vibration and noise, inter alia, by the change of meshing stiffness during operation. Inaccuracies of production and assembly, wear processes and possible damage are some of the further factors. All of these affect both the quality of the system and its durability and the residual processes generated by them can be used to assess the technical condition of devices.

Ensuring correct operation of the gear unit and the longest possible life of the gear are sufficient reasons why shaping of vibroactivity of the gearing at the design stage becomes important. Numerous studies have shown that good results can be obtained in many cases through actions resulting in a slight increase in production costs. Optimization of the construction requires, however, research in the form of computer simulations or with use of the gear prototypes in laboratory conditions. In the first situation, it is necessary to have a gear model whose creation and identification can be extremely time consuming. The use of finished gear models may not produce the desired results if the selected model is too simplistic or does not take into account the well-chosen features of the actual toothed gear.

Laboratory tests characterize not full repeatability and high cost and time consumption of prototype construction.

The article presents the results of laboratory tests aimed at comparing the vibroactivity of the toothed gear, in which the pairs of wheels differ in geometric parameters. The results of laboratory tests are supplement by the results of computer simulations. The presented test method makes possible determination of general guidelines for the reduction of vibration and noise in power transmission systems with gear.

2. Object and aim of research

The object of the study was the toothed gears of the test stand with toothed gears working in the circulating power system. The research was conducted in the form of computer simulations and in laboratory conditions. The dynamic model of the power transmission system was used to numerical calculations. Simulation studies have taken into account conditions that can also be applied during laboratory experiments. The purpose of the calculations and measurements was to determine the potential for changes in the vibroactivity of the gears by modification of selected geometric parameters of toothed gears.

Due to the construction of the toothed gears working in the circulating power system, all pairs of gears have the same gear ratio and their geometric parameters provide the same wheel axis distance. Four sets of parameters describing the geometry of pairs of wheels were used. They had different values of the transverse contact ratio ε_α and face contact ratio ε_β. Identical parameters are used during simulation tests. Gear parameters are listed in Table 1.

Deviation values of the teeth of the wheels have been introduced into the simulation program after measuring the deviation of the wheels mounted in the gearboxes during laboratory researches. Measurements of deviations made it possible to determine the accuracy class obtained during production of gears.

Simulation studies were carried out using a developed dynamic model with toothed gears operating in a circulating power system. It includes all its main components, i.e. electric motor, two single-stage cylindrical gear units and shafts connecting the individual components of the analyzed test stand.

Tab. 1. Parameters of gears used in research

	Pair of wheels no. 1	Pair of wheels no. 2	Pair of wheels no. 3	Pair of wheels no. 4
Number of pinion teeth, z_1	19			38
Number of gear teeth, z_2	30			60
Module, m_n, mm	3.5			1.75
Transverse pressure angle, α_0, °	20			
Helix angle, β, °	$11^1/_3$	15	18	15
Axis distance, a_w, mm	91.5			
Pinion profile shift coefficient, x_1	0.630	0.500	0.70	0.794
Wheel profile shift coefficient, x_2	0.633	0.295	0.171	0.795
Gear mesh width, b, mm	56			
Transverse contact ratio, ε_α	1.239	1.332	1.426	1.400
Face contact ratio, ε_β	1.001	1.318	1.574	2.636
Total contact ratio, ε_γ	2.240	2.650	3.000	4.036

The simulation program includes in the input data among others: the masses and moments of inertia of the main components, stiffness of shafts, the geometry of the toothed wheels and the deviations of their pitches, the type and position of the bearings, and the damping coefficients of vibrations in bearings.

Construction of the model is widely undertaken in works [3, 8]. The work [7] also shows the stages of identification and verification of the model, which made it possible to determine its correctness and applicability in various studies [4, 11, 12].

Simulation and laboratory tests were performed for the same rotational speeds and loads, in both cases the type, temperature, and oil level in the gearboxes were also the same. A test stand with toothed gears working in a circulating power system is shown in Fig. 1.

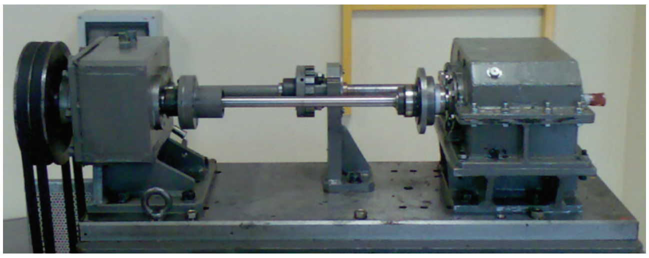

Fig. 1. Test stand with toothed gears working in a circulating power system

The transverse vibration speeds of shafts were achieved with use the Ometron VH300+ laser vibrometer. It allows non-contact recording of transverse vibration of elements, also rotating. Measurements were made at points A, B and C, while the housing vibration velocity at point 7 (as shown in Fig. 2). Shaft transverse vibration velocities were recorded in the respective directions of inter-tooth force.

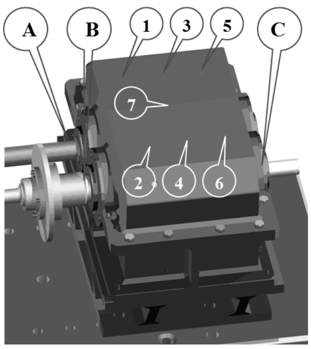

Fig. 2. Location of measurement points on the test stand with toothed gears working in a circulating power system

Measurements of sound pressure changes were carried out at a distance of about 0.5 m from point 7 using a capacitive directional microphone co-operating with the Norsonic signal analyzer.

Acceleration sensor measurements at points 1 ÷ 6 are made using the accelerometers of PCBs marked M353B15 and M353B16.

In addition, for the analysis of recorded signals, three reference signals were recorded in the form of pulses generated once per rotation of each shaft and once for a full cycle of teeth associations.

Values of all signals were recorded at a sampling rate of approximately 78 kHz on a measuring computer compatible with the eight-channel EC VibDAQ 8+ data acquisition card.

3. Research results

During simulation and laboratory tests, the rotational speed of the wheel was changed in range from 600 to 2400 rpm with step 600 rpm. Unit load values were 1.00 and 2.15 MPa. Each measurement lasted about 10 seconds. The test gear of the station has always worked as a reducer.

All recorded signals of accelerations and velocities of vibrations, signals of changes in sound pressure were time averaged. The transverse vibration velocity of the shafts in the bearing nodes, results of the simulation, was also averaged.

The influence of the change of geometric parameters on the vibration of the toothed gear was evaluated on the basis of the changes in the RMS values of the transverse vibration velocity of the shafts. Due to the fact that most of the vibrating energy of the housing is transmitted by the shafts and bearings, these values can be a good measure of the vibroactivity of the gears. In addition, the measured values of the vibration velocity of the housing and the changes in sound pressure were measured.

The RMS values of the transverse vibration speed of the shaft at the measuring point B for the unit load Q = 1.00 MPa, obtained from simulation calculations and laboratory measurements, are presented in Fig. 3.

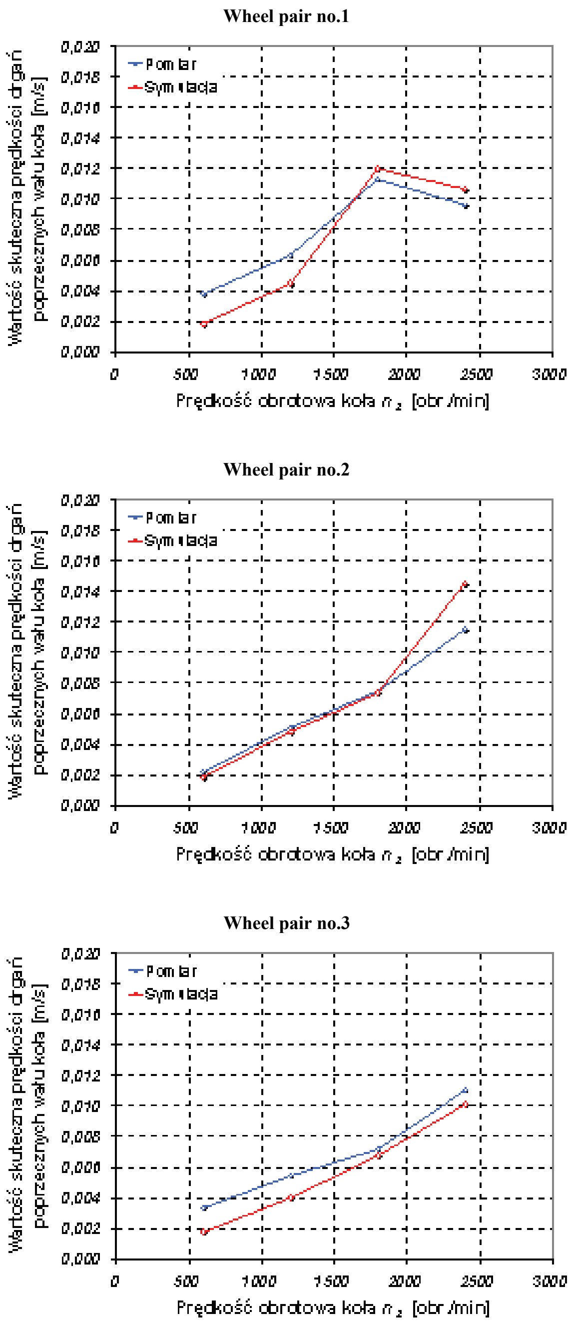

Fig. 3. Effective values of the transverse shaft vibration velocity (test point B) determined experimentally at the laboratory test stand and during simulations for pairs of wheels 1 ÷ 3; unit load value Q = 1.00 MPa

The obtained results confirmed that with the increase of the face contact ratio ε_β, the RMS values of the transverse vibration velocity of the shafts decrease. Significant reduction in gear vibroactivity due to the change in wheel geometry is evident in the comparison to the value of the pinion and wheel shaft oscillation values recorded during work of the gear test with the mounted pair of wheels 1 and 4 at the higher load values, i.e. when Q = 2.15 MPa.

Significant differences in transmission vibration levels are also confirmed by the housing vibration acceleration measurements, which is the main emitter of the gearbox noise [5, 9, 10]. In order to determine the vibration velocity, the measured and recorded acceleration values were subjected to an integration operation. The gear vibration on the housing was measured directly using accelerometers mounted in points 1 through 6 (as shown in Fig. 2). Figure 4 shows the average values of vibration velocity from all six measurement points.

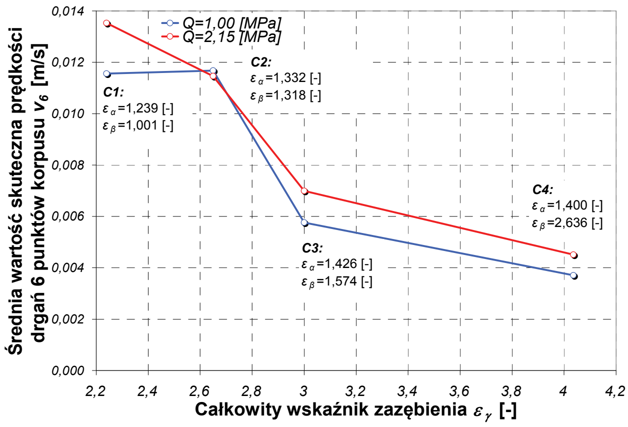

Fig. 4. The dependence of the mean RMS values, determined from the six measurement points located on the gear housing, as a function of the total contact ratio ε_γ. Results obtained at the test stand during research with pairs of wheels 1 ÷ 4 at 1800 rpm (shaft of gear)

The presented results show that the greatest vibration of the housing during all conducted tests was accompanied by the operation of the gear unit with the mounted pair of wheels 1 or 2. For the gear unit working with the mounted pair of wheels 3, especially 4 vibrations were characterized by a lower average value of body vibration velocity. The same dependence, as measured with the accelerator transducers, was obtained by analyzing the results of measurements made using a laser vibrometer.

Figure 5 shows the comparison of the effective values of the sound pressure of the gearbox operating at 1.00 and 2.15 MPa, determined experimentally at the test bench using pairs of wheels 1 ÷ 4 as a function of the total engagement index ε_γ. The measurement of sound pressure changes was carried out simultaneously with the measurement of the vibration speed of the housing of toothed gear. The pressure was measured at a distance of about 0.5 m above measurement point 7.

The presented results show that regardless of the load, the most advantageous results were obtained with a gear mounted with a pair of wheels no.4. For the other rotational speeds, the most advantageous properties, i.e. the smallest vibration intensity, were toothed gear with mounted pair of wheel no.3 or 4.

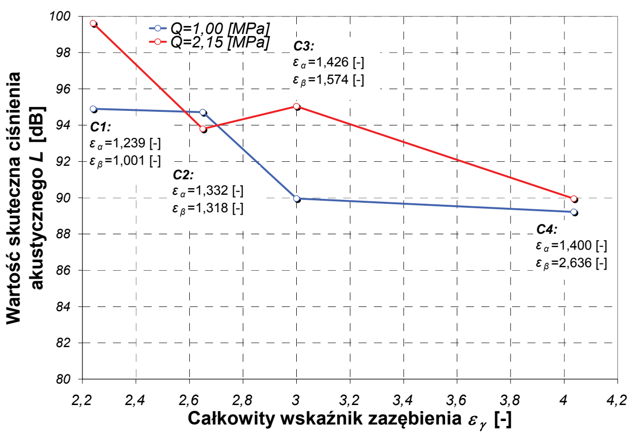

Fig. 5. The mean values of RMS sound pressure changes expressed in dB measured at 0.5 m above measurement point 7, as a function of the total meshing index ε_γ, results obtained with use pairs of wheels 1 ÷ 4 at a wheel speed of about 1800 rpm

4. Conclusions

As shown results of analysis of laboratory and simulation studies, some of which are presented above, it is possible to shape the vibration of the gear unit at the stage of its design. Simulations made using the most faithful model [1, 2, 6], i.e. model of appropriate complexity, allow to determine the direction of design changes, so that the generated vibration and noise generated by the toothed gear is reduced as much as possible.

The results of laboratory tests carried out on a real object - a test stand with gear working in circulating power system confirm the conclusions of the numerical tests. The simple change of geometry of the gear wheels used in the gearbox has in this case reduced the level of vibroacoustic phenomena associated with the transmission. Although the optimization of the construction brings undoubted benefits, it does not need to involve significant financial resources to achieve the desired results.

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