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IMPROVEMENT OF VIBROACOUSTIC PROPERTIES OF TOOTHED GEARS THROUGH CONSTRUCTIONAL MODIFICATIONS OF GEAR TEETH

Summary: The paper draws attention to methods of minimizing values of vibroacoustic factors by reducing the sources. In the case of toothed gears such causes include, first of all, deviations in workmanship, excitations generated when teeth move in and out of mesh, as well as mesh stiffness variations. The paper presents results of experimental research on vibroacoustic properties of gear wheels with modified and non-modified high-profile teeth. Basing on these results, it was determined that the best results in counteracting vibrations and noise are obtained by using jointly an increased gear contact ratio and a tooth profile modification.

POPRAWA WŁAŚCIWOŚCI WIBROAKUSTYCZNYCH PRZEKŁADNI ZĘBATYCH NA DRODZE ZMIAN KONSTRUKCYJNYCH UZĘBIENIA

Streszczenie: W pracy zwrócono uwagę na metody minimalizacji wartości czynników wibroakustycznych oparte na zmniejszeniu przyczyn ich powstawania. W przypadku przekładni zębatych są nimi przede wszystkim odchyłki wykonawcze, wzbudzenia powstające przy wchodzeniu i wychodzeniu zębów w zazębienie oraz wahania sztywności zazębienia. W pracy przedstawiono wyniki badań doświadczalnych właściwości wibroakustycznych kół o uzębieniu wysokim modyfikowanym i niemodyfikowanym. Na podstawie tych wyników stwierdzono, że najlepsze wyniki w zwalczaniu powstawania drgań i hałasów przynosi łączne zastosowanie zwiększonego wskaźnika zazębienia i modyfikacji zarysu zębów.

1. INTRODUCTION

Elimination of exposure to mechanical vibrations by reducing them at the source [3] constitutes a technical solution that gives the best results, however, it cannot always be performed for technical or economic reasons. Limiting the generation of mechanical vibrations of toothed gears requires interference in their internal structure, which consists, among other things, in: employing adequate structural solutions that reduces the generation of mechanical vibrations, using materials with better vibration damping properties, or improving quality of the production and assembly of machine components. Such actions are usually possible only at the stage of designing and manufacturing a machine, so they are out of the scope of end user’s activities.

In the case of toothed gears, gear wheels and bearings are components that are particularly responsible for the vibroacoustic condition [4, 5]. These are also elements that should meet high durability requirements, and this fact has a substantial influence on the economic aspect of the use of toothed gears.

As it has been proved by numerous theoretical and experimental studies, the most important sources of vibrations are: errors (inaccuracies) in workmanship of gear wheels, excitations generated when teeth move in and out of mesh, as well as mesh stiffness variations [4, 9]. In order to reduce the influence of the aforementioned factors, the following solutions are used:

- improving the accuracy of workmanship of toothed gears
- making an adequate modification of the tooth shape, which decreases impulses produced by tooth meshing
- increasing the gear contact ratio.

Current trends towards constructional reduction of vibrations generated by gear wheel teeth are based on making an adequate longitudinal and transverse modification of a tooth and increasing the gear contact ratio by using high-profile teeth [10]. The gear contact ratio characterizes the smallest and the largest number of tooth pairs, which are in contact at the same time. Due to the vibroacoustic condition of a gear, there is striving for obtaining a value of the gear contact ratio equal to an integer greater than unity [11]. In the case of helical teeth it is relatively easy to obtain increased values of the gear contact ratio (by selecting an adequate value of the helix angle), however meeting this criterion with respect to wheels with straight teeth is more difficult. It is possible to increase the value of the gear contact ratio by selecting adequate values of geometric parameters of teeth. The parameters affecting the value of the gear contact ratio include: module, number of teeth, sum of the addendum modification coefficients, pressure angle, and tooth height. If the last two parameters are to be modified, it is necessary to use tools with non-standard tooth profile. Such teeth with an increased tooth height are called high-profile teeth.

When modifying the tooth profile, making such a modification is connected with selecting an adequate value of the modification, which should be equal to anticipated teeth deformations. Thus, this value must be adjusted to the loads transmitted by toothed gears. Apart from these modifications of the teeth profile, there appear trends towards modifying the tooth root and changing the transition curve in order to strengthen the tooth base. When designing a tooth profile, the influence of lapping is sometimes taken into account [8]. Such an example of comprehensive optimisation of teeth is presented in Fig. 1.

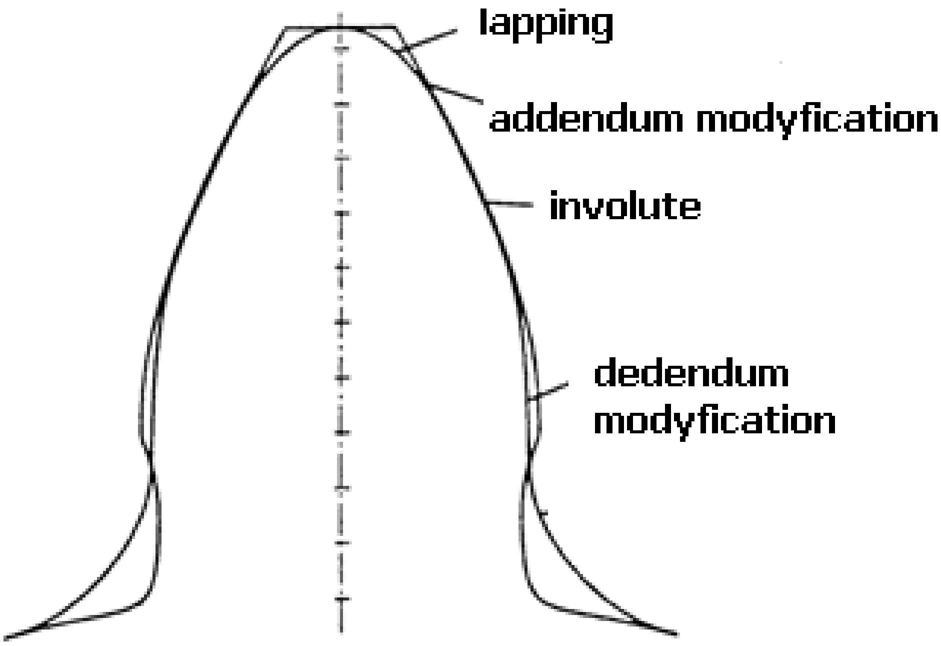

Fig. 1. Example of optimised tooth profile
Rys. 1. Przykład optymalizowanego zarysu zęba

When designing a modification of a tooth profile, the decrease in the gear contact ratio caused by technological treatment should be taken into account [8]. It is possible to reduce this value below unity, which makes correct mesh operation impossible. In such a case, the use of high-profile teeth is one of possible solutions.

This paper presents results of empirical research on the influence of geometrical features of a tooth on vibroacoustic properties of toothed gears with high-profile teeth. In particular, the study examined experimentally the change in the value of the sound level value and in the value of acceleration of gear housing vibrations caused by constructional modifications of teeth. The research concerned gear wheels with an increased teeth height and lowered pressure angle at the pitch diameter. For comparison purposes, gears with standard teeth were also studied.

2. CHARACTERISTICS OF THE RESEARCH METHOD

The experimental research, which constitute a significant part of this study and which aimed at determining the influence of gear wheel types on vibrations and noise of a gear, were performed at a test rig [6]. The measurements of acceleration of gear housing vibrations and acoustic pressure were performed with the use of sensors, piezoelectric microphones, and National Instruments' CompactDAQ system.

In the study, the parameters characterizing the vibroacoustic condition were considered in the function of mesh frequency, which is a parameter combining the rotational speed with the number of teeth, and which at the same time is the frequency that generates vibrations of the system. The mesh frequency [7] is calculated from the following formula:

$$f_z = \frac{n \cdot z_1}{60} = f_n \cdot z_1 \tag{1}$$

Acoustic properties of gear wheels were studied in the mesh frequency range fz = (160÷1100) Hz., which corresponded to the range of the rotational speed: n1 = 205÷3001 min^{-1}.

Dimensions and parameters characterizing the geometry of the studied teeth were put in a table. High-profile (WS-3.0) and standard (STS) teeth were selected for determining the changes in the influence of excitation sources (i.e. gear wheels) on vibrations. High-profile teeth were made in two versions: modified (modification value: 0.35 mm) and non-modified. The studied gear wheels were lubricated by spraying Liqui Moly's SAE 85W-140 mineral gear oil with the flow rate of 0.5 dm^3/min. The temperature of the oil was 25±1 ºC.

The obtained results of measurements were submitted to a statistical analysis, which aimed at determining the factors of equations describing the courses of acoustic power in the function of mesh frequency f_z. The calculations were performed with the use of the wspol_dyn1[1] program, developed especially for this purpose and working in the MATLAB 6.0 environment.

3. RESULTS OF THE STUDIES

Exemplary courses of acceleration of vibrations of the gear housing containing gear wheels with the high-profile modified and non-modified WS-3.0 teeth and the standard STS teeth are shown in Fig. 2; they were obtained for the gear loaded with the nominal torque: M_{st}=10 Nm. In this figure it can be observed that the lowest values of acceleration of gear housing vibrations were measured when the gear wheels with high-profile modified teeth were working in the gear, while the highest values were obtained for wheels with standard teeth. In the mesh frequency range of 1000÷1100 Hz, local extrema were observed for the standard teeth.

[1] The author of this program is Jarosław Joostberens, Ph.D. Eng. from the Department of the Electrical Engineering and Process Control in Mining at the Technical University of Silesia.

Tab. 1.

Characteristic parameters of the studied pairs of gear wheels; Designations: M – modified, NM – non-modified

		WS-3.0 NM	WS-3.0 NM	STS
Kind of meshing		No modificated hight tooth meshing	No modificated hight tooth meshing	No modificated normal tooth meshing
Pressure angle, °	α	20		20
Addendum ratio	h_a*	1,30·m		1,00·m
Dedendum ratio	h_f*	1,70·m		1,25·m
Fillet	ρ	0,212·m		0,36·m
Gear contact ratio	ε_α	2,03		1,625
Number of tooths (pinion/wheel)	z	26/27		
Module, mm	m	4		
Face width, mm	b	10		
Pitch diameter (pinion/wheel)	d	104/108		

Fig. 3 illustrates the courses of the acoustic power level determined for the high-profile modified and non-modified WS-3.0 teeth and for the standard STS teeth for the gear loaded with the nominal torque M_{st}=10 Nm. It can be easily noticed that the high-profile modified teeth show particularly advantageous acoustic properties. The total influence of the modification and increased value of the gear contact ratio results in a considerable reduction of the noise generated by toothed gears. The standard teeth showed the worst properties. Similar dependences were obtained with respect to other considered loads from the range of M_{st}=20÷50 Nm.

4. SUMMARY

This study discussed the problem of a constructional reduction of vibroacoustic effects accompanying the operation of toothed gears, which is not an easy task considering the complexity of the system. Making a gear featuring reduced generation of vibrations requires considerable constructional and technological efforts, and such a gear must be then properly operated. It is important to be aware that the present state of technological progress in production of drives is very advanced, and any improvement in the vibroacoustic condition requires methods that have not been commonly used so far. One of such method consists in using gear wheels with non-standard parameters of tooth geometry, which enable obtaining high values of the gear contact ratio and making adequate modification.

Basing on the results of the studies of toothed gears, presented in the paper, there were found advantageous vibration properties of the high-profile teeth characterized by an integer value of the gear contact ratio. This improvement results from a decrease in the value of mesh stiffness variations, which is one of main causes of vibrations and noise generated by a gear. Besides, it has been found that the best results in counteracting vibrations and noise are obtained by using jointly an increased gear contact ratio and a tooth profile modification.

When considering vibrations of toothed gears, operating conditions of a gear should also be taken into account – this applies particularly to resonance states. Situations, where a frequency that generates vibrations corresponds to the natural frequency of a wheel or its subharmonic frequencies, should be avoided.

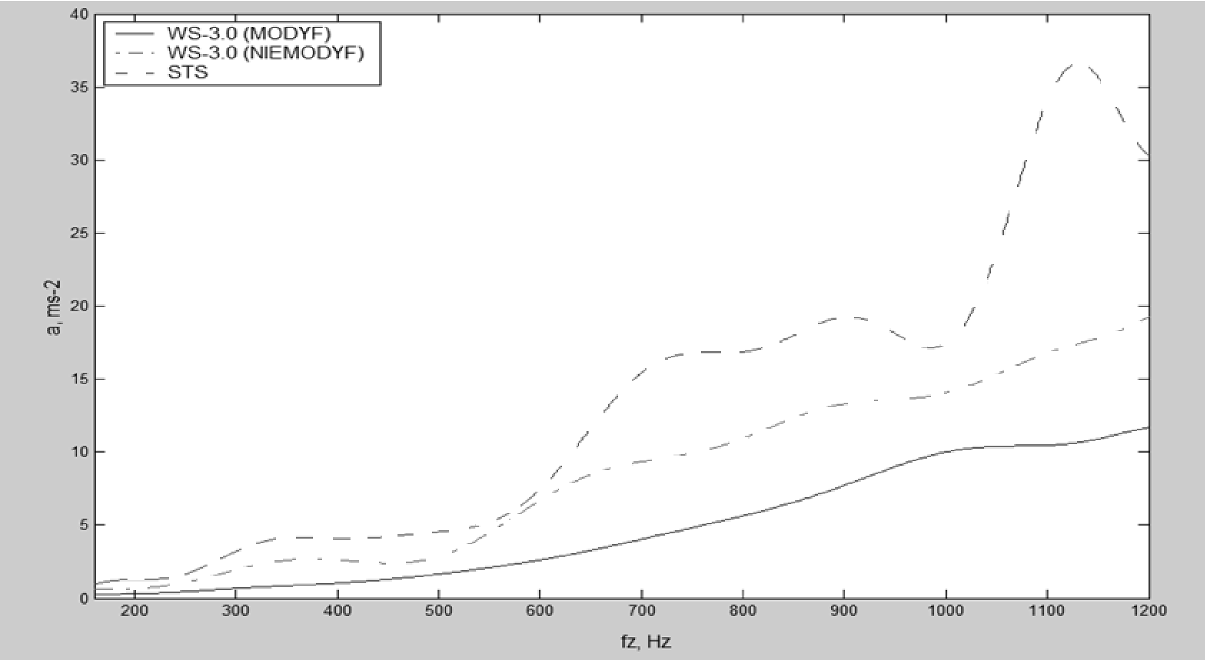

Fig. 2 The courses of acceleration of gear housing vibrations obtained for the studied teeth in the function of the mesh frequency for the gear loaded with the nominal torque Mst=10 Nm

Rys. 2 Przebiegi przyśpieszenia drgań korpusu przekładni uzyskane dla badanych uzębień w funkcji częstotliwości zazębienia przy obciążeniu przekładni momentem nominalnym Mst=10 Nm

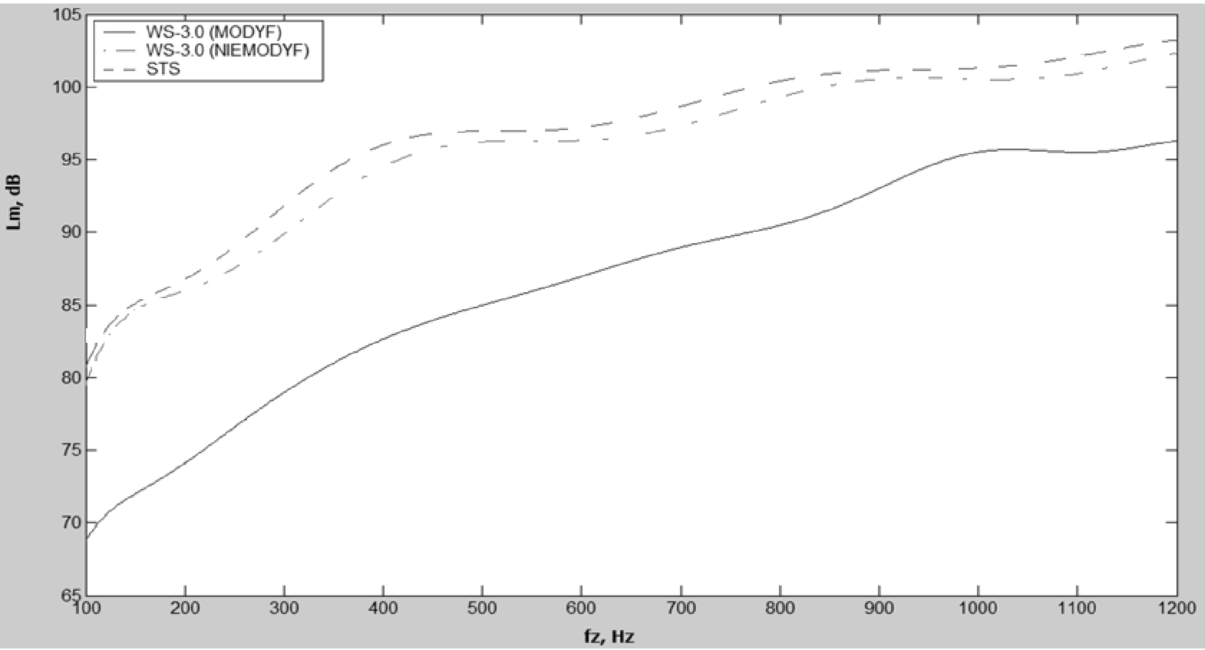

Fig. 3 The courses of acoustic power obtained for the studied teeth in the function of mesh frequency for the gear loaded with the nominal torque Mst=10 Nm

Rys. 3 Przebiegi mocy akustycznej uzyskane dla badanych uzębień w funkcji częstotliwości zazębienia przy obciążeniu przekładni momentem nominalnym Mst=10 Nm

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