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Effects of Construction Changes in the Teeth of a Gear Transmission on Acoustic Properties

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This paper presents results of experimental research on the acoustic properties of gear wheels with high-profile teeth with differentiated tooth height. Those results showed that gear transmissions with high-profile teeth have the best acoustic properties, with the value of the transverse contact ratio $\varepsilon_\alpha \approx 2.0$. They also showed that a reduction in tooth height, and thereby in contact ratio, increased the sound pressure level.

gear tooth height gear contact ratio sound pressure level

1. INTRODUCTION

The level of noise can be effectively reduced at the stage of designing structures, machinery and equipment [1]. Existing methods of reducing noise in the environment can be divided into (a) administrative and legal and (b) technical.

Within technical methods, it is possible to reduce noise annoyance at the workplace by

- reducing or minimizing noise emission at source;
- reducing vibroacoustic energy on the routes of its transmission;
- reducing noise immission in certain areas of the environment and in humans.

Eliminating exposure to mechanical vibrations by reducing them at source is the best technical solution [1]; however, it cannot always be used for technical or economic reasons. It may consist in a total elimination of, or reduction in, sources of vibrations. Vibrations are eliminated when production processes do not generate vibrations.

Gear transmissions, commonly used in power transmission systems of transport machines (e.g., automotive vehicles) are important in shaping the acoustic climate at the workplace. A reduction in the noise these devices generate improves the working conditions and thus workers' effectiveness.

A gear transmission acoustically and mechanically emits sound into the environment (Figure 1). Emission depends on

- the properties of sound transmission routes;
- the radiation of sound from transmission gears into the environment;
- the values of the forces inducing sound [2].

In the case of two first factors, reducing in noise emission is connected with redesigning the housing of the transmission, changing the way individual elements are connected and changing the location of the equipment. The greatest potential for reducing in noise is in decreasing the value of the forces inducing mechanical vibrations, i.e., the third factor, the primary source of sound.

In the case of gear transmissions, gear wheels and bearings are particularly responsible for vibroacoustic conditions [3, 4]. Industrial transmissions usually use rolling bearings, which do not generate noise above the value emitted by gear teeth, provided that they are in a good physical condition. The main causes of vibrations and noise generated by teeth have already been discussed.

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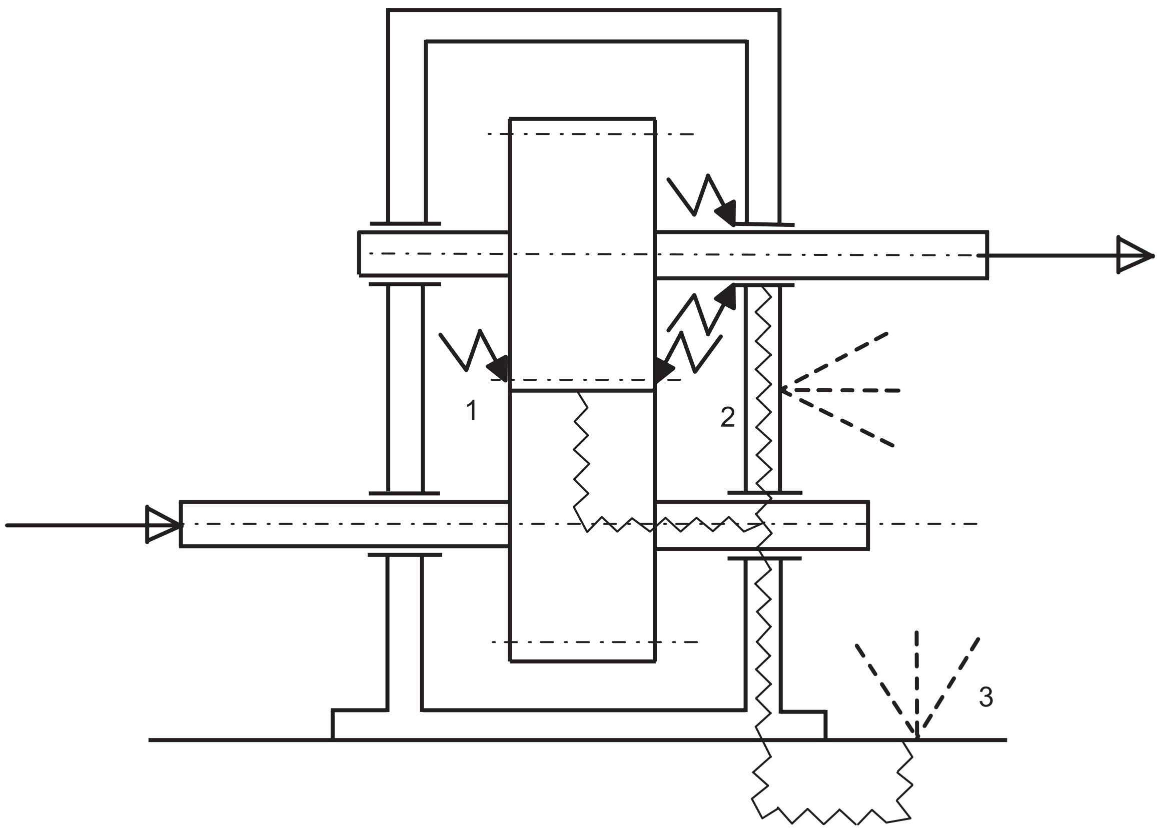


Figure 1. A model of gear transmissions generating vibrations and noise.

2. PURPOSE AND SCOPE

This study should determine empirically the effects of construction changes in teeth on the acoustic properties of gear transmissions. The paper also discusses noise emitted by transmissions and presents current research in this field.

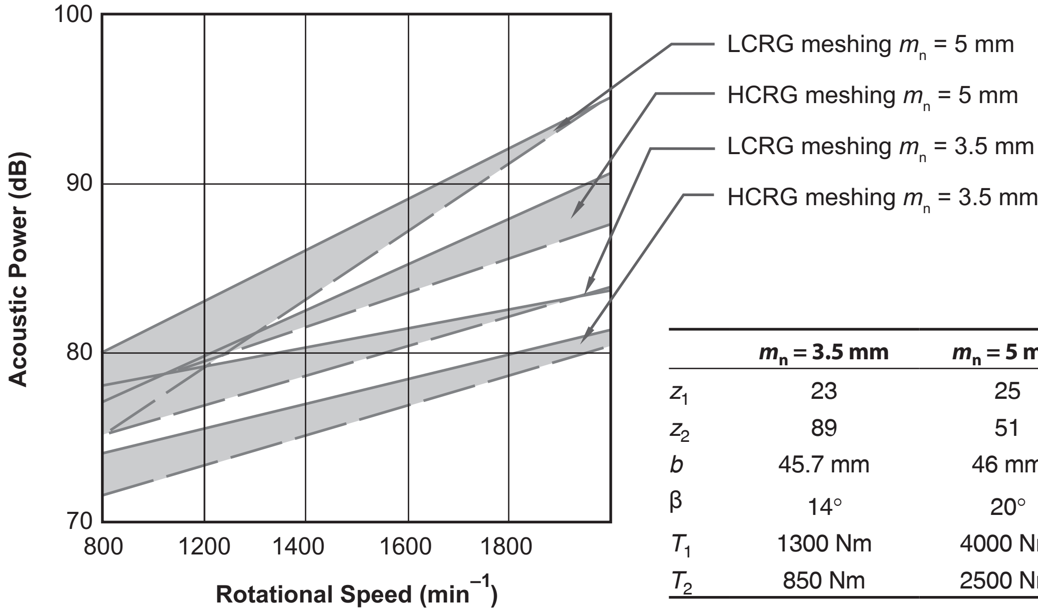
3. CURRENT RESEARCH

Research on reducing the emission of noise vibrations generated by the teeth of gear wheels has been conducted for many years. It demonstrated that high-profile teeth with the contact ratio close to the integer value of 2 had the best vibroacoustic properties in the case of straight teeth (which are usually used in planetary transmissions).

Weck measured the level of acoustic power determined for gear transmissions with different height and width of teeth [5]. Figure 2 shows that using high-profile teeth results in reduced noise in the entire speed range.

Winter did experimental research on noise generated by gear transmissions with wheels with standard, low- and high-profile teeth [6]. High-profile teeth produced two different results. For the lowest load, wheels with high-profile teeth were among the most noisy, whereas for the highest load, they were the most quiet. According to Winter, at low loads on wheels, manufacturing deviations had a decisive effect on noise, while at higher loads, the effect of the increased value of the transverse contact ratio was favourable.

Knabel also studied noise generated by a transmission with wheels with different tooth height [7]. He found that noise decreased in spur high-profile teeth with the contact ratio $\varepsilon_\alpha = 2.15$ as compared with standard teeth. Joachim and Lauster found that using high-profile teeth in spur gears resulted in noise reduced by ~5 dB as compared with standard teeth [8]. Weck and Lachenmaier [9, 10] and Weck, Lachenmaier and Goebbelet [11] measured acoustic power as a function of rotational speed, determined for gear transmissions with different height and width of teeth. They also showed that high-profile teeth were acoustically better than standard ones [9, 10, 11].



	m_n = 3.5 mm	m_n = 5 mm
z_1	23	25
z_2	89	51
b	45.7 mm	46 mm
β	14°	20°
T_1	1300 Nm	4000 Nm
T_2	850 Nm	2500 Nm

Figure 2. Acoustic power for gear wheels with different tooth height [5]. *Notes*. LCRG = low contact ratio gear; HCRG = high contact ratio gear; m_n = normal module; z_1 = number of teeth of the pinion; z_2 = number of teeth of the gear wheel; b = facewidth of gear; β = helix angle of tooth; T_1, T_2 = nominal load torque.

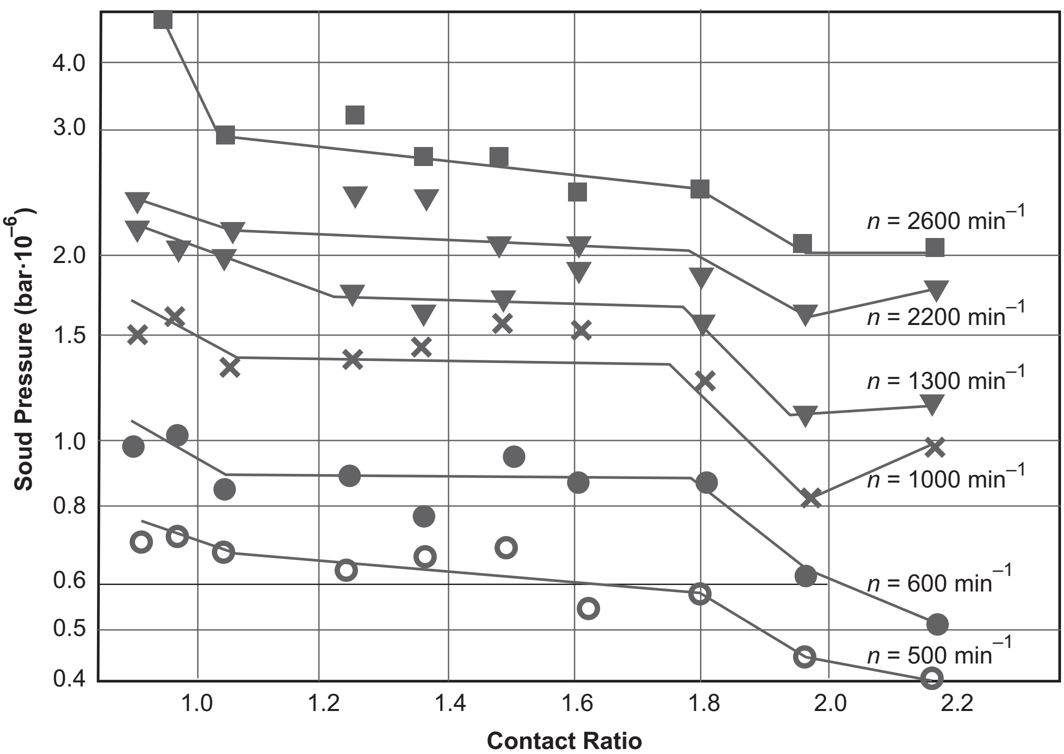


Figure 3. The effect of contact ratio on sound pressure [12]. *Notes*. Peripheral force P = 3.34 kN, n = rotational speed.

Niemann and Unterberger measured the sound pressure level during operation of spur gears with different height of teeth [12] (Figure 3). For the speed and load ranges covered by the study, the sound pressure level decreased near the value of the transverse contact ratio $\varepsilon_\alpha = 2.0$, while for the value of the contact ratio in the range of ε_α = 1.8–1.1, there were no local extrema. An increase in noise did not occur until $\varepsilon_\alpha = 1.0$.

Hösel carried out acoustic research on gear transmissions with helical teeth (high-profile and standard) [13]. To determine the effect of the transverse contact ratio, the outer diameters of the wheels were reduced, like in Niemann and Unterberger [12]. The following ranges of this ratio were thus obtained: standard teeth ε_α = 1.6–1.1, high-profile teeth ε_α = 2.0–1.1. The measurements showed that for the range of tooth height corresponding to of the transverse contact ratio ε_α = 2.0–1.7, the impact of ε_α on acoustic effects was small or nonexistent. Whereas for $\varepsilon_\alpha < 1.7$ and a decrease in tooth height, the noise level increased significantly.

Döbereiner studied helical and spur gears ($\varepsilon_\alpha = 2.0$ and ε_β = 0.0–0.5) with a reduced pressure angle measured at the pitch diameter of $\alpha_t = 17.5°$ [14]. Those measurements showed noise reduced by ~3 dB for helical teeth as compared with spur teeth in the entire range of rotational speeds. In addition, the effect of load on the level of emitted noise was small.

This review shows there have been relatively many studies on noise and transmissions with wheels with high-profile teeth. However, the results have varied considerably.

4. RESEARCH CONDITIONS

A test rig was used in the experimental research, which constituted a significant part of this study and which aimed at determining the influence of the types of gear wheels on vibrations and noise. The level of noise was measured with a Brüel & Kjær 2236 sound level meter (Denmark). This meter measures noise with the accuracy of ±0.1 dB. Acoustic power was determined in compliance with Standard No. PN-ISO 8579-1:1996 [15]. Figure 4 shows the test rig with the measuring system.

In this study, the parameters characterizing vibroacoustic conditions were considered as a function of the mesh frequency. The mesh frequency combines rotational speed with the

Figure 4. Test rig with the measuring system.

number of teeth; at the same time it is the frequency that generates vibrations of the system. It is calculated from Equation 1 [4]:

$$f_z = \frac{n \cdot z_1}{60} = f_n \cdot z_1. \quad (1)$$

Vibroacoustic properties of gear wheels were studied for the mesh frequency f_z = 160–1100 Hz, which corresponded to rotational speed n_1 = 205–3001 min^{-1}.

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.

The teeth had the following geometric parameters in common:

- module m = 4 mm,
- facewidth of gear b = 10 mm,
- pressure angle at the pitch diameter $\alpha = 20^{\circ}$,
- number of teeth of the pinion z_1 = 26 and the gear wheel z_2 = 27.

To increase the number of the variants of teeth, the tip diameter of wheels with high-profile teeth (WS-3.0) was appropriately reduced, so there were another seven variants. Table 1 lists the dimensions and parameters characterizing the geometry of the teeth.

The teeth were made in the accuracy class 7 according to Standard No. DIN 3962-1:1978 [16]. The test wheels were made of 40H steel submitted to quenching and tempering. The gear wheels were lubricated with Transol VG 320 mineral gear oil with the flow rate of 0.5 dm^3/min. The temperature of the oil was 25 ± 1 °C.

5. RESULTS

In the experiment, the sound pressure level was measured for

- eight variants of teeth;
- 31 values of the mesh frequency;
- five values of the static load torque (10, 20, 30, 40 and 50 Nm).

The mesh frequency was applied discretely. To present the results in the form of function graphs and thus to facilitate interpretation, the least square method was used for the approximation [17]. A custom program, which runs in MATLAB 6.0 (wspol_dyn1), was used to calculate the coefficients of the equations describing the experimental data. That software determines coefficients of a polynomial equation that best meets the criterion. Minimizing the value of residual variance σ^2 at simultaneous maximizing the value of the coefficient of determination R^2 was that criterion. The program calculated coefficients for polynomial equations from 1 to 30, and then selected the equation that best fulfilled the criterion.

As approximation was done with a high-order polynomial, the courses were not linear, but they had an extremum, which was connected with the nonlinear effects in the transmission. Thus, it was sometimes difficult to explicitly determine which teeth have better properties. A range of mesh frequencies, in which some teeth are better than others, should be considered.

TABLE 1. Gear Wheels and Their Parameters

Gear Wheel	h (m)	ε_α	c_γ (N/(mm·µm))	n_E (min^{-1})
WS-3.0	3.0	2.030	16.94	7655
WS-2.9	2.9	1.898	15.99	7495
WS-2.8	2.8	1.763	15.03	7321
WS-2.7	2.7	1.625	14.04	7132
WS-2.6	2.6	1.484	13.03	6924
WS-2.5	2.5	1.340	11.99	6695
WS-2.4	2.4	1.192	10.93	6442
WS-2.3	2.3	1.039	9.84	6160

Notes. h = total tooth height, ε_α = transverse contact ratio, c_γ = average mesh stiffness, n_E = resonant rotational speed.

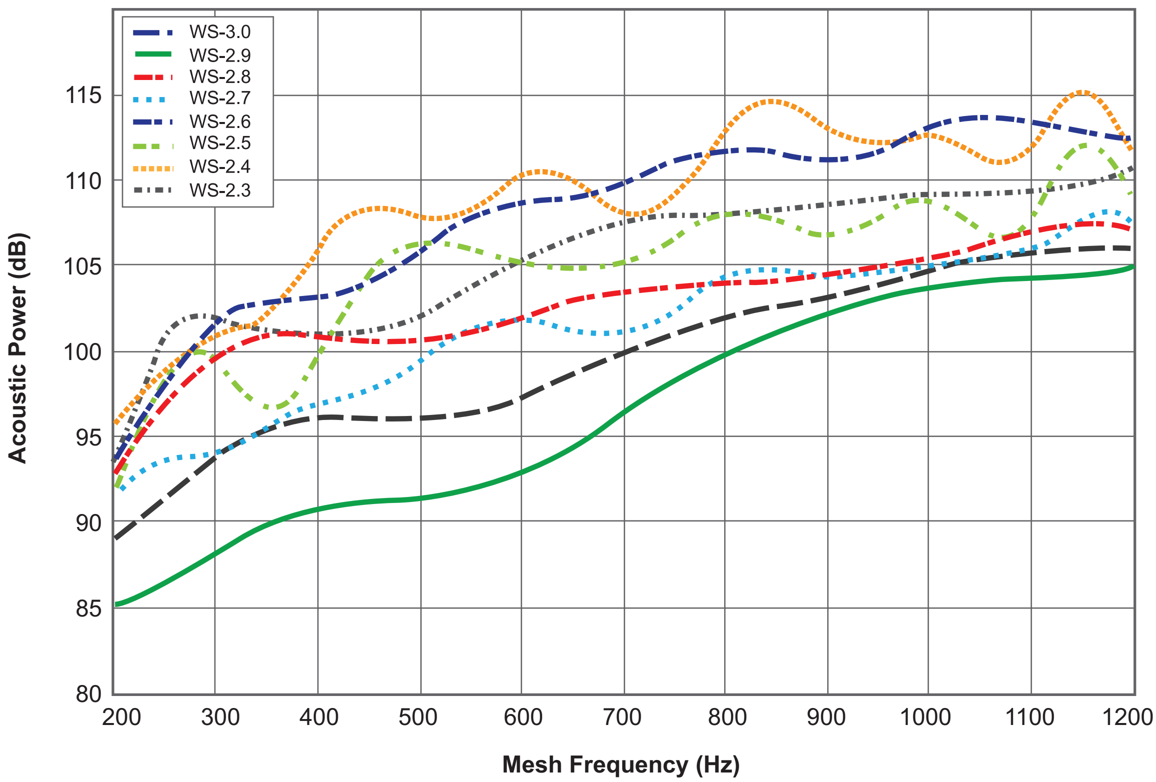


Figure 5. Acoustic power at the load M_s = 10 Nm for various teeth and frequencies.

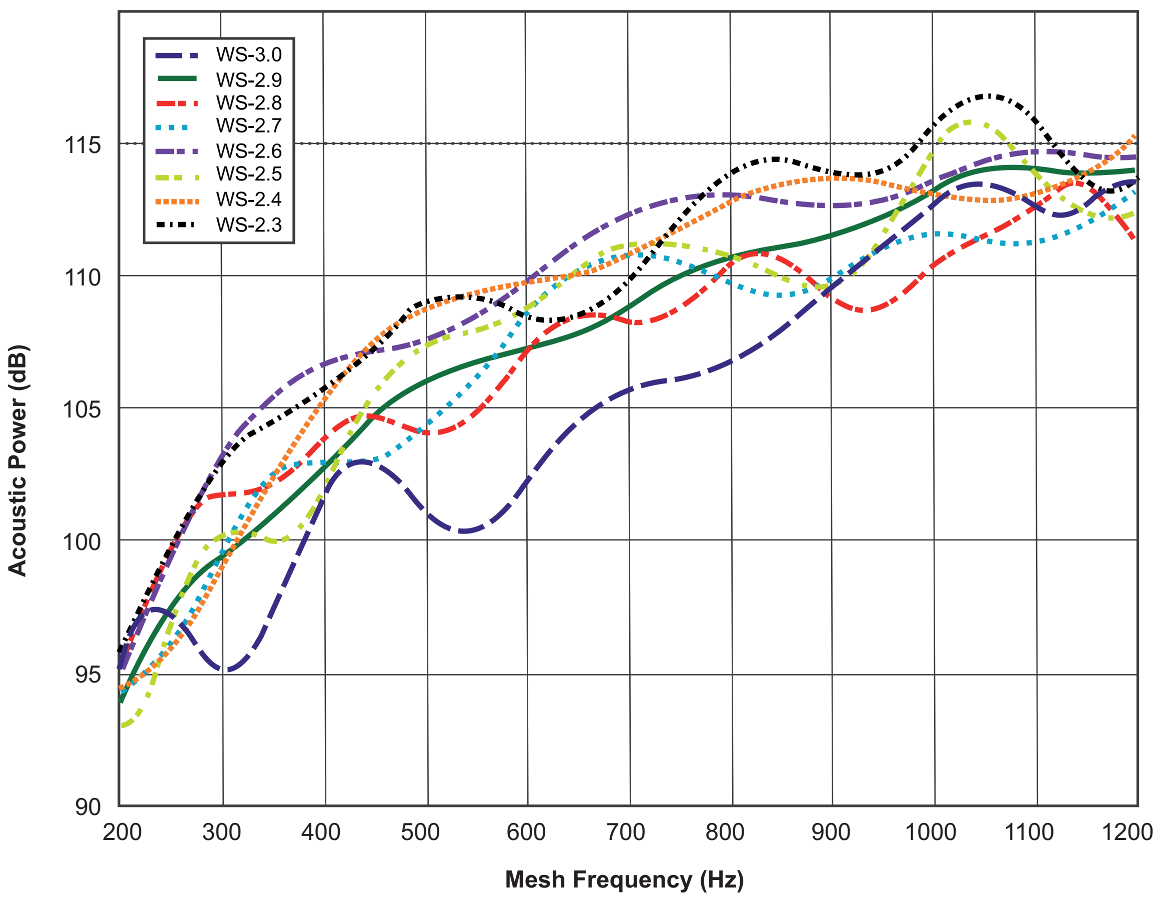


Figure 6. Acoustic power at the load M_s = 30 Nm for various teeth and frequencies.

Figure 5 presents curves of acoustic power obtained for all examined teeth as a function of the mesh frequency for the torque M_s = 10 Nm, which loaded the gear wheels. This figure does not show any relationship between tooth height (and thus the contact ratio) and acoustic power. The WS-2.9 teeth, which under the same conditions had the lowest vibrations [16], appeared to be the quietest. Next, in the order from the quietest to the loudest one, were the following teeth: WS-3.0, WS-2.7, WS-2.8, WS-2.5, WS-2.3, WS-2.6 and WS-2.4.

Figures 6–7 show that increasing the load brought favourable acoustic features of WS-3.0. These teeth have the highest value of the contact ratio, which is an integer value at the same time. WS-2.9, which for the lowest loads were the quietest teeth, appeared to be much worse than WS-2.7 and WS-2.8 (lower value of the contact ratio).

The teeth with the lowest contact ratio, WS-2.3, did not have the worst acoustic properties. In addition to the curve at the load M_s = 40 Nm, in other cases these teeth had lower levels of the acoustic power than WS-2.4 and even WS-2.6, the teeth with a much higher contact ratio. However, even though WS-2.3 had the lowest contact ratio, they had an integer value of this ratio.

To determine the dependencies between noise and the construction of teeth (tooth height), the measured sound pressure level for the load M_s = 50 Nm and for three sample mesh frequencies were compared with trend lines. Figure 8 shows that a reduction in tooth height, and thereby in the gear contact ratio resulted in an increase in acoustic power. Niemann and Unterberger's results are very similar [12] (Figure 3).

6. CONCLUSIONS

Research on noise emission that accompanies operating gear transmissions showed that gear transmissions with high-profile teeth, with the transverse contact ratio $\varepsilon_\alpha \approx 2.0$, have the best acoustic properties. The favourable properties result from the equalized course of meshing stiffness. Reducing tooth height, and thereby the con-

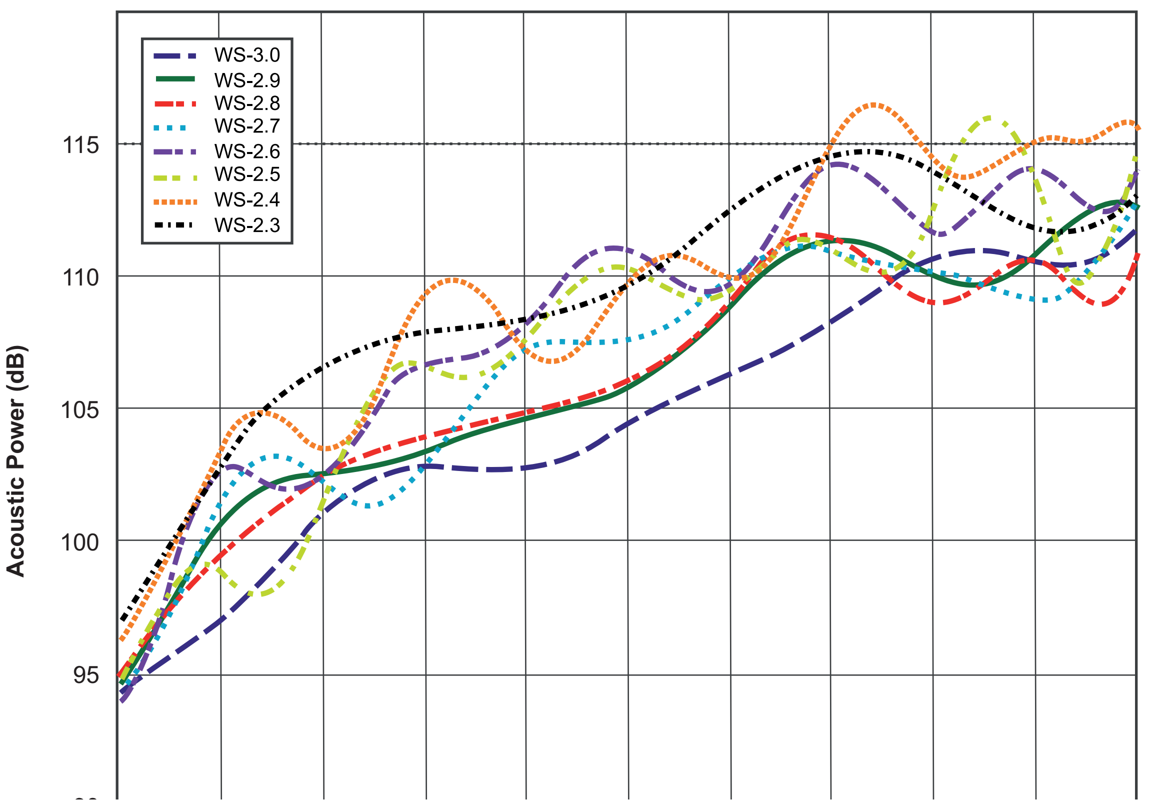


Figure 7. Acoustic power at the load M_s = 50 Nm for various teeth and frequencies.

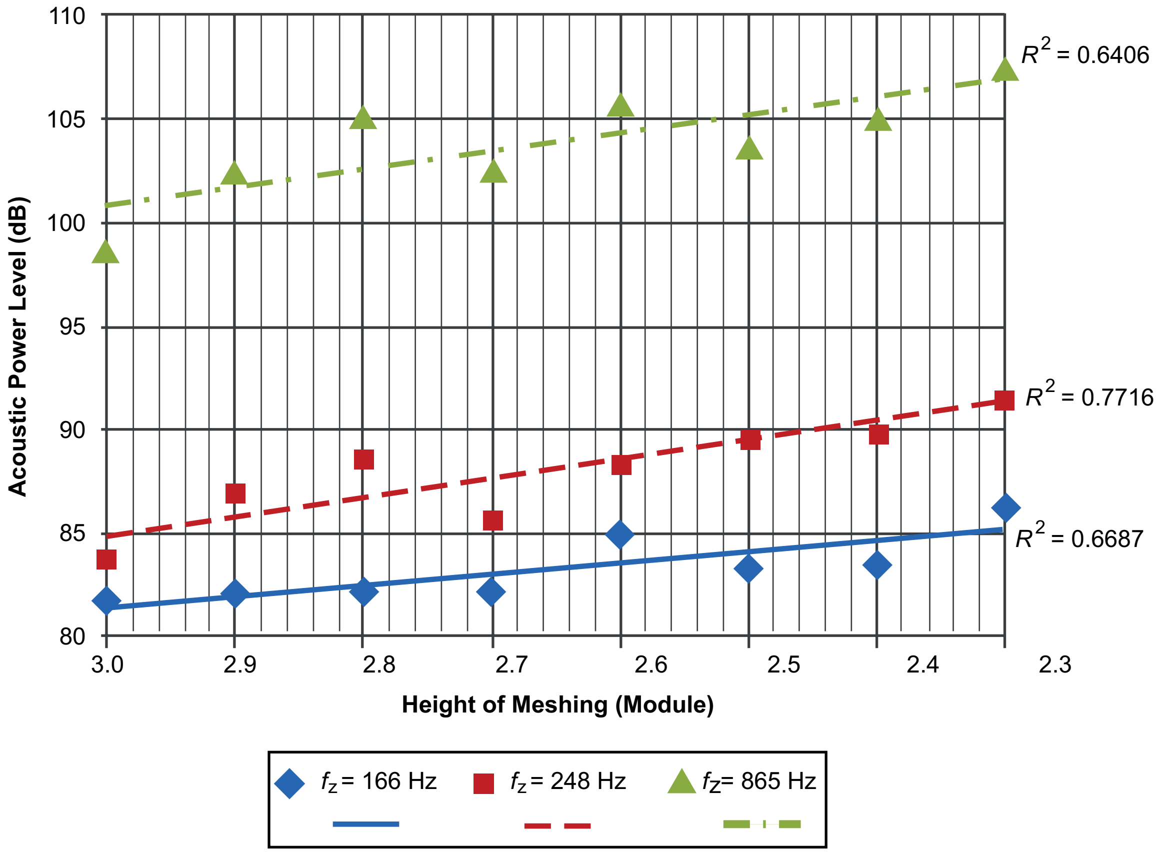


Figure 8. The effect of tooth height on acoustic power, and trend lines. *Notes.* f_z = mesh frequency.

tact ratio, causes an increase in the sound pressure level. The results of the study clearly prove that it is possible to reduce the intensity of harmful factors emitted by gear transmissions, i.e., mechanical vibrations and noise, by changing the construction of gear teeth.

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