THE MODELLING OF NOISE EMISSION BY THE GEARBOX OF VEHICLES GAZ 3110, 31105

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Abstract

In the presented work the sound generation design procedure is presented. On its basis the mathematical model of calculation of the noise emission by a gearbox of cars of family GAZ-3110, 31105 is created. Diagrams of emission of noise by a gearbox depending on frequency of rotation of its entrance shaft and a backlash in gearing of teeths are received. They show, that in case an angular backlash in gearing radiated noise increases by 2-3 dB for each 0.5 degrees. As a result the methodology of the analysis of degree of wear of gearbox's elements on the basis of noise spectrum should be created. We present the methodology of noise analysis of overmuch action of teeth based on the model of noise emission of elastic plates. According to the developed methodology we got the dependences of the acoustic emission of the gearbox for GAZ vehicle for the whole range of working speed of the engine from different level of wear of gears' closed pairs. The analysis of the received data showed the increase of the acoustic emission by 1.2-1.5 times at the frequencies corresponding the overmuch action of the gears. This data can be used for analysis of the noise of a car to separate the noise of the gearbox, and also for counting and estimating the possible noise of a car.

Keywords: *mathematical model, gearbox, emission of noise, spectrum*

The operation of the gearbox (like any gear mechanism) is not noiseless. Usually, the basic reasons of noise in gearbox are overmuch action and wear of teeth, the violation of coaxiality of shafts and bearings, too little oil in the casing. Too much noise is the first sign of malfunctions, which can lead to the breaking down of teeth or gearbox.

It would be quite useful to approximately determine the degree of wear of teeth with frequency analysis of noise in octave and three-octave frequency bands. But the correct procedure of frequency analysis is difficult as it needs to be conducted in the room with special acoustic conditions, where other sources of noise like engine and so on can be eliminated. So, the experimental noise analysis and the separation of gearbox's noise can sometimes be not satisfactory because of difficulties connected with analysis of the received data.

As a result the methodology of the analysis of degree of wear of gearbox's elements on the basis of noise spectrum should be created. We present the methodology of noise analysis of overmuch action of teeth based on the model of noise emission of elastic plates.

Effective sound pressure in the media is directly proportional to the particle velocity [1]. In the near sound field the wave has flat front and the magnitude of effective sound pressure P:

$$P_a = \rho \cdot c \cdot V_a, \tag{1}$$

where: V_a – vibration velocity of the surface of emission [m/s], c – speed of sound in the media [m/s], ρ- density of the material of the media [kg/m^3].

Acoustic power, emitted by the plate [W]:

$$W = \rho \cdot c \cdot S \cdot V_a^2 \cdot j, \tag{2}$$

where: S – the square of the plate [m^2], j – index of emission of the plate.

As all the parts of the gearbox take part in the translational motion, the stroke will appear only in case the parts with different relative speed collide. The tooth of the gear strikes with the tooth of the wheel, and the speed of the binding of the teeth can be counted theoretically. The number of the strokes per minute depends on the theoretical speed of rotation of gear and the number of teeth:

$$P = \frac{\omega_T}{2\pi} \cdot z \text{ [Hz]}. \tag{3}$$

When colliding the gear and wheel have the speeds of binding ω_1 and ω_2. During the first phase of collision the speeds of them equate [4]. During the second phase the motion of the bodies is determined by the coefficient of restitution k. When counting the collision of steel parts this coefficient k is considered to be 0.55.

The speed of collision U in the first phase is determined according the formula:

$$U = \frac{J_1 \cdot \omega_1 + J_2 \cdot \omega_2 \left(1 + \frac{2\pi R \cdot \arcsin(x / 2\pi R)}{x}\right)}{J_1 + J_2} \text{ [rad/s]}, \tag{4}$$

where: J_1, J_2 – momentums of inertia of the colliding bodies, kg/m^2; ω_1, ω_2 – speed of binding of colliding bodies, rad/sec; x – backlash in gearing, m; R – radius of wheel, m.

The speed of the colliding bodies in the second phase is determined with the formulas:

$$U_1 = U \cdot (1 + k) - k\omega_1, \tag{5}$$

$$U_2 = U \cdot (1 + k) - k\omega_2, \tag{6}$$

where:

U₁ , ***U₂*** – speeds of the colliding bodies in the second phase, rad/sec.

In the beginning of the stroke the bodies of gear wheels start emitting wide spectrum of vibrations. The environmental air start dithering. Effective meaning of the intensity of the sound emission are determined with the formulas:

$$I_1 = \frac{1}{2} U_{1Y}^2 \cdot \rho \cdot c \cdot P \cdot j \cdot R, \tag{7}$$

$$I_2 = \frac{1}{2} U_{2Y}^2 \cdot \rho \cdot c \cdot P \cdot j \cdot R. \tag{8}$$

The total acoustic power is determined with the sum of acoustic powers of different sources. The acoustic power of each source is the product of acoustic emission intensity of the source and the emitting square [1]. Then:

$$W_1 = I_1 \cdot S_1, \tag{9}$$

$$W_2 = I_2 \cdot S_2, \tag{10}$$

where: S_1, S_2 – square of the outward surfaces of the colliding bodies [m^2], W_1, W_2 – acoustic powers of the colliding bodies [W].

Total acoustic power W_C is the following:

$$W_C = W_1 \cdot W_2. \tag{11}$$

The level of the acoustic power is determined with the formula 12:

$$L_W = 10\lg\frac{W}{W_0}\ [\text{dB}], \qquad (12)$$

where: W – acoustic power of the source [W], W_O – threshold of the acoustic power, usually it is considered as $\boldsymbol{W_O} = 10^{-12}$ [W].

The mathematical and dynamic models of the gearbox for GAZ-3110, 31105 vehicles are created to determine the vibration velocity of the colliding elements.

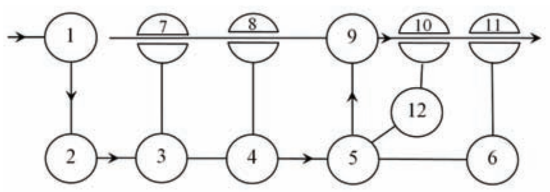

Fig. 1. The structural scheme of the 1st gear action for GAZ-3110, 31105 vehicles

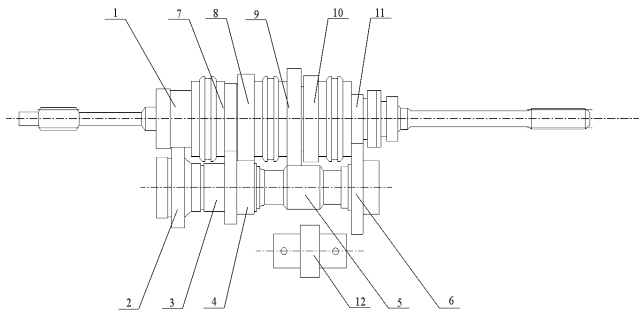

Fig. 2. The whole view of the shafting of the gearbox for GAZ-3110, 31105 (1 - main drive gear, 2 – countershaft driven gear of the 3rd gear action, 4 – drive gear of the 2nd gear action, 5 – drive gear of the 1st gear action, 6 – drive gear of the 5th gear action, 7- follower gear of the 3rd gear action, 8 – follower gear of the 2nd gear action, 9 – follower gear of the 1st gear action, 10 – follower gear of backward motion, 11 – follower gear of the 5th gear action, 12 – idle gear of backward motion's shaft)

Structural scheme of the gearbox's action is shown of the Fig. 1. Whole view of the poer shaft of the gearbox is shown on the Fig. 2. Dynamic parameters of gearbox's elements are shown in the Table 1.

Tab. 1. Momentum of inertia and inflexibility of bonds between gearbox's elements

Number of mass	Momentum of inertia **kgm²**	Number of mass	Momentum of inertia **kgm²**	Number of section	Torsional stiffness **Nm/rad**	Number of section	Torsional stiffness **Nm/rad**
1	$1.26\cdot10^{-3}$	7	$1.47\cdot10^{-3}$	1-2	3962482	8-9	318848
2	$1.21\cdot10^{-3}$	8	$1.63\cdot10^{-3}$	2-3	570587.6	9-10	928818
3	$6.12\cdot10^{-4}$	9	$2.08\cdot10^{-3}$	3-4	1678199	10-11	356046.9
4	$2.86\cdot10^{-4}$	10	$0.95\cdot10^{-3}$	4-5	327923.9	4-8	1469547
5	$2.06\cdot10^{-4}$	11	$0.79\cdot10^{-3}$	5-6	356617.2	5-9	2008219
6	$1.23\cdot10^{-3}$	12	$2.53\cdot10^{-4}$	3-7	649182.2	6-11	1679604
7				7-8	928818	1-7	3051831
8				10-12	482330	12-5	482330

System of difference equation, describing the torsional modes of gearbox's elements:

According to the developed methodology we got the dependences of the acoustic emission of the gearbox for GAZ vehicle for the whole range of working speed of the engine (800 – 5000 rpm) from different level of wear of gears' closed pairs (Fig. 3). The analysis of the received data showed the increase of the acoustic emission by 1.2-1.5 times at the frequencies corresponding the overmuch action of the gears. This data can be used for analysis of the noise of a car to separate the noise of the gearbox, and also for counting and estimating the possible noise of a car [2, 3].

$$\begin{cases}
J_1\ddot{\varphi}_1 + k_1\dot{\varphi}_1 + C_{1-2}\left(\varphi_2 - \varphi_1\right) = M_1 \sin \omega t, \\
J_2\ddot{\varphi}_2 + k_2\dot{\varphi}_2 - C_{1-2}\left(\varphi_2 - \varphi_1\right) + C_{2-3}\left(\varphi_3 - \varphi_2\right) = M_2 \sin \omega t, \\
J_3\ddot{\varphi}_3 + k_3\dot{\varphi}_3 - C_{2-3}\left(\varphi_3 - \varphi_2\right) + C_{3-4}\left(\varphi_4 - \varphi_3\right) + C_{3-7}\left(\varphi_7 - \varphi_3\right) = \\
= M_2 \sin \omega t, \\
J_4\ddot{\varphi}_4 + k_4\dot{\varphi}_4 - C_{3-4}\left(\varphi_4 - \varphi_3\right) + C_{4-8}\left(\varphi_8 - \varphi_4\right) + C_{4-5}\left(\varphi_5 - \varphi_4\right) = \\
= M_3 \sin \omega t, \\
J_5\ddot{\varphi}_5 + k_5\dot{\varphi}_5 - C_{4-5}\left(\varphi_5 - \varphi_4\right) + C_{5-9}\left(\varphi_9 - \varphi_5\right) + C_{5-12}\left(\varphi_{12} - \varphi_5\right) + \\
+ C_{5-6}\left(\varphi_6 - \varphi_5\right) = M_4 \sin \omega t, \\
J_6\ddot{\varphi}_6 + k_6\dot{\varphi}_6 - C_{5-6}\left(\varphi_6 - \varphi_5\right) + C_{6-11}\left(\varphi_{11} - \varphi_6\right) = M_6 \sin \omega t, \\
J_7\ddot{\varphi}_7 + k_7\dot{\varphi}_7 - C_{3-7}\left(\varphi_7 - \varphi_3\right) = -M_2 \sin \omega t, \\
J_8\ddot{\varphi}_8 + k_8\dot{\varphi}_8 - C_{4-8}\left(\varphi_8 - \varphi_4\right) = -M_3 \sin \omega t, \\
J_9\ddot{\varphi}_9 + k_9\dot{\varphi}_9 - C_{5-9}\left(\varphi_9 - \varphi_5\right) = -M_4 \sin \omega t, \\
J_{10}\ddot{\varphi}_{10} + k_{10}\dot{\varphi}_{10} - C_{12-10}\left(\varphi_{10} - \varphi_{12}\right) = -M_7 \sin \omega t, \\
J_{11}\ddot{\varphi}_{11} + k_{11}\dot{\varphi}_{11} - C_{6-11}\left(\varphi_{11} - \varphi_6\right) = -M_6 \sin \omega t, \\
J_{12}\ddot{\varphi}_{12} + k_{12}\dot{\varphi}_{12} - C_{5-12}\left(\varphi_{12} - \varphi_5\right) + C_{12-10}\left(\varphi_{10} - \varphi_{12}\right) = M_5 \sin \omega t.
\end{cases} \tag{13}$$

As we can see from the received results when in case an angular backlash in gearing radiated noise increases by 2-3 dB for each 0.5 degrees.

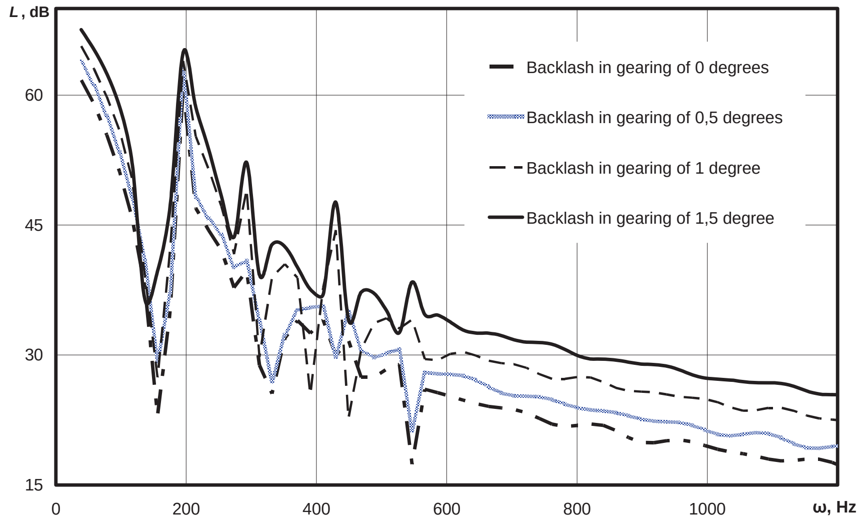

Fig. 3. The spectrum of vibroacoustic signal, at the rotating frequency 1000 rpm of intake shaft of the gearbox

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