# The Noise Structure of Gear Transmission Units and the Role of Gearbox Walls

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*The noise emission of gear units (gearboxes) depends both on the disturbances (gear meshing, bearing operation, etc) and on the insulating capabilities and modal behavior of the housing. Natural vibrations of the housing walls can be prevented or intensified depending on design parameters. The mechanism of exciting and emission of transmission noise is defined by carrying out the process of propagation of excitation energy through the structure of power transmitters and by modal testing of the housing. The results of vibration and noise testing in comparison to the results of modal testing give the possibility of identification of noise structure for the chosen gearbox. Comparison and analysis of the results obtained lead to precise determination of the causes of creation of the total spectrum of gear transmission units.*

***Keywords:*** *gearbox housing, gears, vibration, noise, modal testing*

## 1. INTRODUCTION

In gear transmission units, sound is generated by excitation in gear teeth meshes and in rolling bearings. Impacts, sliding, rolling, etc. absorb disturbance energy in the elastic structure of machine parts and transmit it through the whole structure. Interior surfaces emit a part of this energy into the surroundings in the form of noise. Another part is converted into heat by damping. Figure 1 shows the structure of disturbance, damping and noise emission in gear transmission units. Some of the main parts of this process are as follows:

- Primary sound waves are caused directly in gear meshes and emitted into the interior of the gear unit. These waves penetrate through the housing walls into the surroundings. A part of the wave energy is damped in the gear unit walls.
- The elastic structure of the transmission unit parts (gears, shafts, housing, etc) absorb the dominant part of disturbance energy. This energy moves in the form of waves within these parts and a lot of it is damped. Exterior surfaces of the parts emit secondary sound waves inside the housing.
- Disturbance energy in the volume of the machine part may excite natural vibrations which can create tertial sound waves.
- The role of the housing is dual. It can simultaneously be the insulator of primary and secondary sound waves and the amplifier of the tertiary ones.

The acoustic emission of gear transmission units has been treated in a number of papers but this one considers the mechanism of noise generation and the role of the gearbox housing. This is a continuation of paper [2], which contains a numerical analysis of modal structure and natural vibrations. The manner of modal excitation is established by investigating excitation of modal vibration. Other papers present the possibility of numerical prediction of noise (papers [1] and [3]), and the reduction of gearbox noise by optimization of geometry of gear pairs has been treated in [10] and [9]. The effects of service conditions on sound intensity of the gearbox have been shown by Oswald, James, Zakrajsek and others in paper [6]. Due to the increasing working speeds, which affects intensification of sound intensity, Oswald, Choy and others have developed a global dynamic model which they use for simulation of gearbox dynamics [7], then a programme for calculation of sound intensity [8] and they have shown, on a housing with one deformable side, that its vibrations participate in the spectrum of gearbox noise. In that example, however, the biggest influence on the general level of noise is made by the frequency of gear teeth meshing. Further, Sellgren and Akerblom, in testing for the needs of "Volvo" [9] and Harris and others in [11] observe the problem by starting from the fact that the housing which is not rigid enough can have significant influence on teeth meshing (increase of transmission error). This certainly leads to intensification of disturbances in meshes and intensification of noise separately. In [12], Inoue has, by using FEM and BEM, and for the purpose of reducing vibrations, worked on optimization of simple shapes of the housing by considering the influence of the radius of rounding of the upper part.

This paper aims at determining the structure of noise emitted by the housing walls and establishing the correlation between the housing vibrations and the noise emitted. The noise structure in comparison with disturbance conditions (gear meshing) defines the main phenomenon in the mechanism of noise generation.

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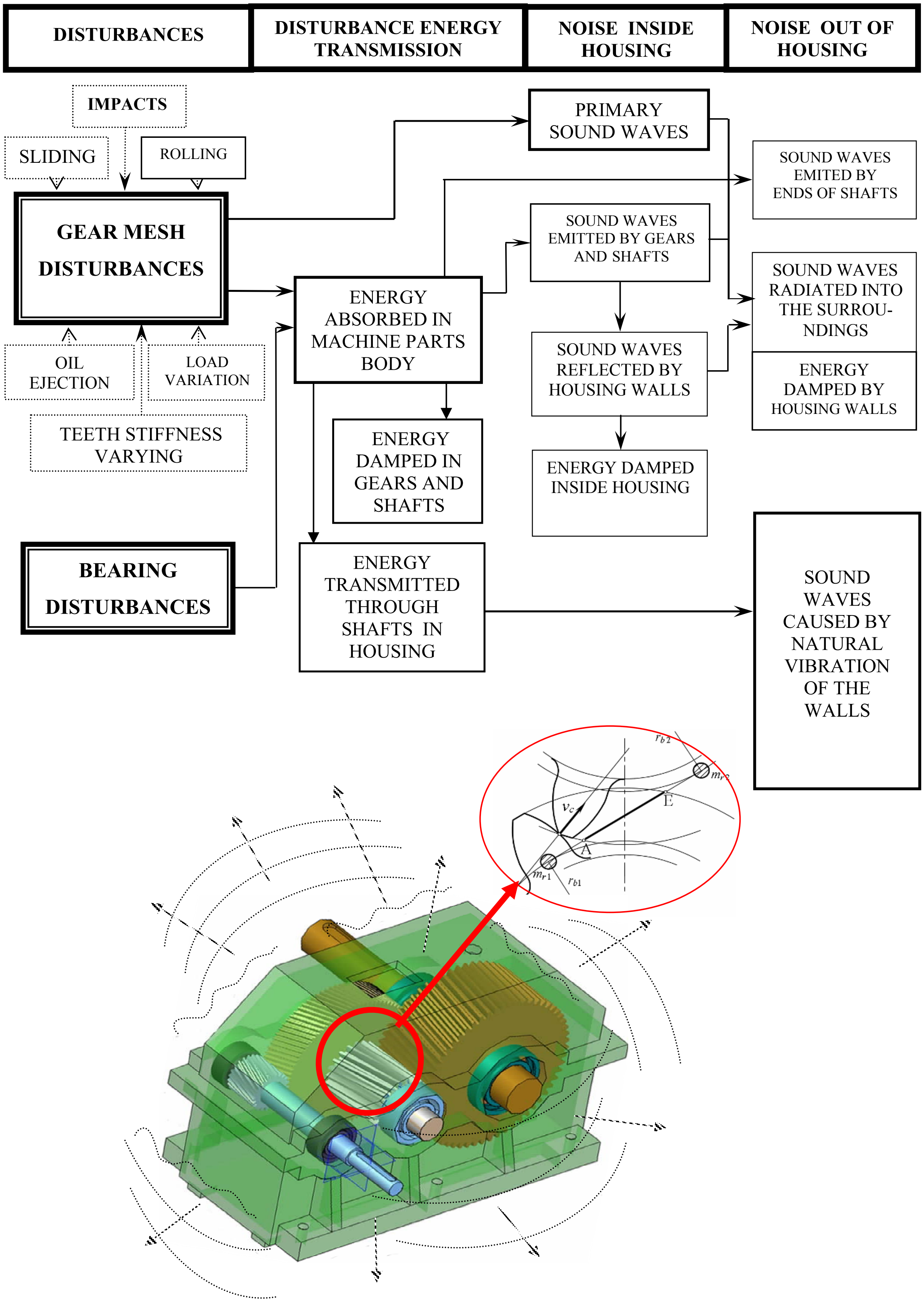



**Figure 1. General structure of the process of disturbance energy transformation**

## 2. TRANSMISSION OF ENERGY THROUGH THE SYSTEM STRUCTURE

Transmission of disturbance energy from the zone of teeth meshing to the housing walls is realized through the gear bodies, shafts and bearings (Fig.2). Energy is distributed through these parts by wave motion. A part of the energy is transmitted through the joints (contiguous surfaces) with considerable losses. A significant amount is lost at the passage of the energy from the gear to the shaft. The degree of reduction is similar at the passage from the shaft to the bearing and from the bearing to the housing. It is considerably increased at the passage of disturbance energy over the contiguous surfaces of the bearing balls. So, this refers to a relatively high number of contacts on the way of transmission from the teeth mesh to the housing surface. The ratio between the sound power $W_s$ and the disturbance power in the teeth meshes $W_d$ can be defined as a factor of transmission (transmissibility) of disturbance energy through the system structure.

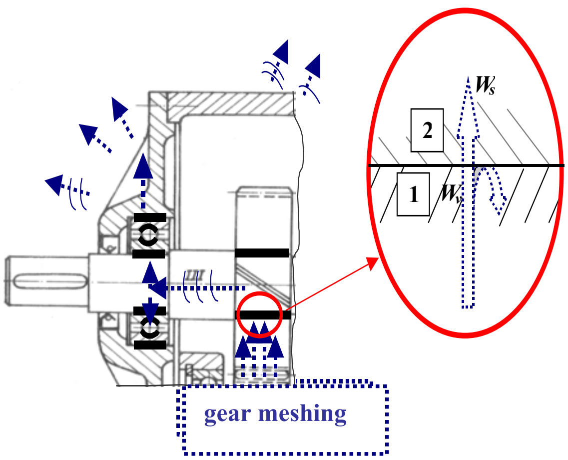



**Figure 2. Transmission of energy through the gear transmission structure**

One part of the sound power ($W_{s1}$) represents a part of the inside sound energy which comes through the walls. This energy transmission is realized by elastic waves through the wall thickness. The sound reduction is proportional to the sound frequency and the wall thickness.

Another part of sound radiation is transmitted from the gear contacts to the surfaces of the housing walls and emitted in the form of elastic waves into the surroundings ($W_{s2}$).

The first two parts of the sound power create forced waves in the elastic structure of the walls which radiate sound into the surroundings.

The third part of the sound power is a result of natural free vibrations, i.e. elastic waves of the housing walls ($W_{s3}$). By using the measured modal damping, the modal kinetic energy can be calculated by means of FEM method. In this case, the total kinetic energy is equal to the sum of the kinetic energy $E_{kj}$ of all modal shapes for a certain disturbance. If this disturbance is caused by gear meshing, the power of modal vibration is

$$W_n = \sum_{j=1}^{q} W_{nj} = \sum_{j=1}^{q} E_{kj} f_{nj} \tag{1}$$

where $q$ – the number of modal shapes and $f_{nj}$ - the natural frequency. It is possible to divide the total power transmission factor (transmissibility) into two parts:

$$\zeta_T = \frac{W_s}{W_d} = \frac{W_s}{W_d}\frac{W_v}{W_v} = \frac{W_v}{W_d}\frac{W_s}{W_v} = \zeta_{T1}\zeta_{T2} \tag{2}$$

where $W_v$ is the total wall vibration power.

The first part of the transmission factor is proportional to the vibration power ratio $\zeta_{T1} = W_v / W_d$, and the second one $\zeta_{T2}$ is proportional to the sound radiation in comparison with the wall vibration. The transmission factor $\zeta_{T2}$ is smaller if the material density $\rho_2$ of the surroundings is smaller in comparison with the wall density.

The housing walls are made of cast iron or steel with the high level of density $\rho_1$ and with the high elastic wave speed $c_{w1}$. Acoustic space is presented with the much smaller density $\rho_2$ and the wave speed $c_{w2}$ for the air.

The simplified formula [14] of disturbance power transmission from the walls into the surroundings (Fig. 2) is

$$\zeta_{T2} = \frac{W_s}{W_v} = \frac{1}{1 + \frac{1}{4}\left(\frac{\rho_2 c_{w2}}{\rho_1 c_{w1}} + \frac{\rho_1 c_{w1}}{\rho_2 c_{w2}}\right)^2} \tag{3}$$

By using this formula and the values of density and wave speed for steel and for the air, we obtain the ratio between the sound power and the wall vibration power $\zeta_{T2} = 4 \cdot 10^{-4}$. This means that an extremely small part of vibration energy is transmitted into sound energy, i.e. sound power.

## 3. MODAL VIBRATION OF THE HOUSING

The housing walls excited by disturbances which, through the bearings and shafts, come from the zone of teeth meshing, oscillate with natural frequencies [2]. Elastic deformations at wave motion and natural oscillation are complex. The process of excitation of main shapes of oscillation is also complex, as well as determination of levels of oscillation energy, that is the ratio between the excitation energy and the emitted energy. For the purpose of accurate definition of the causes leading to excitation of certain modes, possible shapes of oscillation and natural frequencies have been firstly defined by applying the FEM method, and then modal testing of the gearbox housing has been performed. The results show that only a small number of modes have been excited in the observed range of up to 3000 Hz. The analysis of the results obtained shows that modal oscillation will be excited if it is coincided by modal:

- directions of deformations,
- if the excitation acts at the point of the biggest deformations and
- if the excitation frequency is equal to the natural frequency of the corresponding modal shape.

However, the modal shape of oscillation can also be excited when the excitation frequency is not identical with the natural frequency. The complex mechanism of excitation of certain modes as well as the results of numerical modal analysis and the results of modal testing of the gear transmission housing are treated in a detailed manner in [2, 13].

## 4. MEASURING AND ANALYSIS OF VIBRATION AND NOISE

The gearbox presented in Fig. 1 has been used as the subject of testing. Measuring and analysis of vibrations and noise have been performed by the application of the PULSE-system, B&K. Modal testing of the transmission unit housing has been carried out by means of impulse excitation – the modal hammer (Fig.3), and by measuring of vibrations which has been analyzed by using a FFT frequency analyzer. Some of the chosen results are presented in Figure 4. Vibration has been carried out in the area for bearings, by using a piezoelectric accelerometer (Fig. 3). Impact force (impact of the modal hammer) has been applied around the housing walls orthogonal on the wall.

Figure 3. Gearbox housing modal testing

In Figure 4, diagram (a) presents relative vibration response caused by impact in the area of the thin wall (Fig. 3). The response is very intensive for the high natural frequency of about 2.4 kHz. For impact in the area of bearings (area of thick wall), the response for that frequency is less (response diagram – 4b). The next response diagram (c-Fig. 4) is obtained by using impact in the gear tooth. Impact energy (disturbance energy) has to be transmitted through the gear body, through the shafts and across bearings, and then it excites natural vibrations of the housing walls. A very high level of disturbance energy dissipation causes very low level of responded natural vibrations, but it is obtained response of high number of natural frequencies.

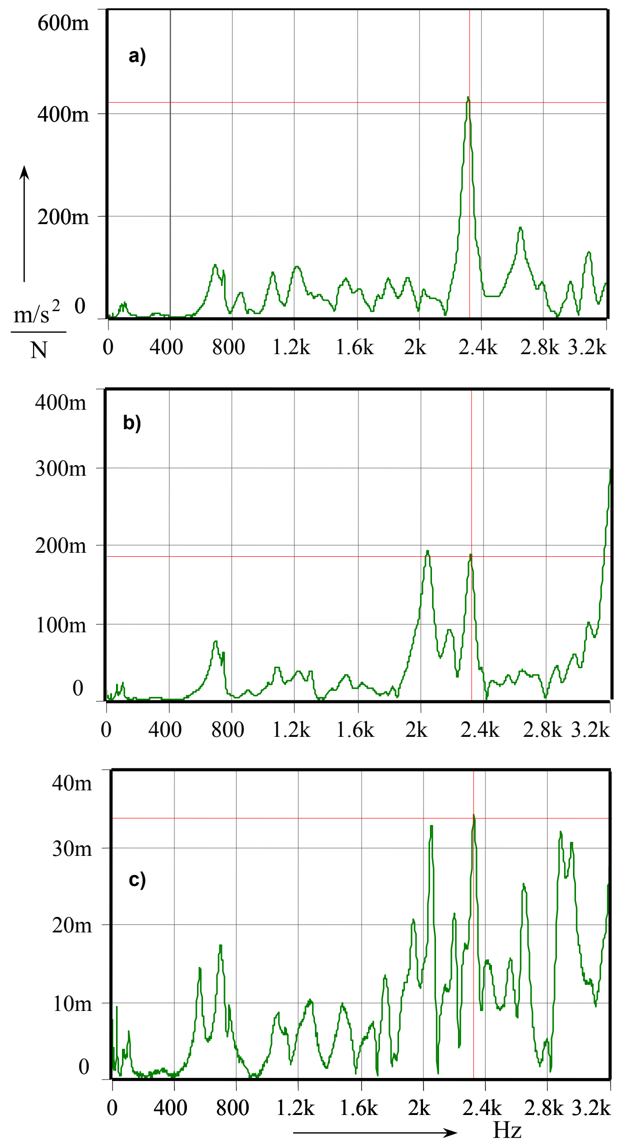



**Figure 4. Frequency response of housing modal testing: a) impact on the top wall of the housing, b) impact on the housing in the area of bearings, c) impact on the gears**

Table 1. shows all natural frequencies, which respond using the hammer modal test. The table

contains natural frequencies up to 1kHz with damping coefficients.

**Table 1. Modal parameters**

| Trans. func. | $f_1$ | | | | $f_2$ | | | |
|---|---|---|---|---|---|---|---|---|
| | $f_r$ | $f_a$ | $f_b$ | $\eta$ | $f_r$ | $f_a$ | $f_b$ | $\eta$ |
| NZ00 | 330.0 | 326.2 | 331.2 | 0.0152 | 458.7 | 455.0 | 461.2 | 0.0135 |
| NY01 | 331.2 | 328.7 | 332.5 | 0.0115 | 460 | 457.5 | 462.5 | 0.0109 |
| NXY02 | 332.7 | 330.0 | 333.7 | 0.0111 | 462.5 | 455 | 463.7 | 0.0189 |
| NXY03 | 331.2 | 328.7 | 332.5 | 0.0113 | 462.5 | 460 | 463.7 | 0.0081 |
| NZ04 | 328.7 | 325 | 330 | 0.0152 | 456.2 | 452.5 | 458.7 | 0.0136 |
| NX05 | 330 | 327.5 | 331.2 | 0.0112 | 460 | 455 | 461.2 | 0.0135 |
| NX06 | 330 | 327.5 | 331.2 | 0.0112 | 460 | 458.7 | 462.5 | 0.0083 |
| NZ07 | 328.7 | 326.2 | 330 | 0.0116 | 457.2 | 453.7 | 458.7 | 0.0109 |
| NZ08 | 328.7 | 326.2 | 330 | 0.0116 | 456.2 | 452.5 | 458.7 | 0.0136 |
| NZ09 | 330 | 327.5 | 331.2 | 0.0112 | 457.5 | 453.7 | 460 | 0.0138 |
| NZ010 | 328.7 | 326.2 | 330 | 0.0116 | 456.2 | 452.5 | 458.7 | 0.0136 |

| Trans. funk. | $f_3$ | | | | $f_7$ | | | |
|---|---|---|---|---|---|---|---|---|
| | $f_r$ | $f_a$ | $f_b$ | $\eta$ | $f_r$ | $f_a$ | $f_b$ | $\eta$ |
| NZ00 | 616.2 | 606.2 | 622.5 | 0.0265 | 941.2 | 922.0 | 952 | 0.0319 |
| NY01 | 621.5 | 608.7 | 626.2 | 0.0282 | 943.7 | 923.7 | 947.5 | 0.0252 |
| NXY02 | | | | -- | 941.0 | 928.7 | 950 | 0.0226 |
| NXY03 | | | | -- | | | | -- |
| NZ04 | | | | -- | 936.2 | 918 | 957 | 0.041 |
| NX05 | 613.7 | 607.5 | 621.2 | 0.0223 | 955 | 942 | 960 | 0.0188 |
| NX06 | 616.2 | 600 | 618.7 | 0.0303 | | | | -- |
| NZ07 | 611.2 | 603.7 | 616.2 | 0.0205 | 941.2 | 933.7 | 956.2 | 0.0239 |
| NZ08 | | | | -- | 943.7 | 932.5 | 952.5 | 0.0212 |
| NZ09 | 610 | 595 | 616.2 | 0.0348 | 945 | 933.7 | 957.5 | 0.0252 |
| NZ010 | 602.5 | 585 | 608.7 | 0.0393 | 938.75 | 922.5 | 948.7 | 0.0279 |

The power transmission unit has been placed into an anehoic chamber (Fig.5) so that acoustic pressure could be used for the analysis. The microphone has been placed above the gearbox, at the distance of 0.5m.

**Figure 5. Anehoic chamber**

The gearbox drive has been realized by means of an electric variator with the rotation speed from the next door room in relation to the anehoic chamber. Measuring has been performed by using the excitation from the gears with defects (Fig. 6a) which have been caused on purpose. For the purpose of determining the effects of the housing walls, measuring has been performed when the upper part (cover) of the housing has been removed (Fig. 6b) and in the situation when the housing has been closed (Fig. 6c). The rotation speed of the input shaft has been varied within the range of 140-1100 $min^{-1}$. Two defects under the angle of $180^0$ have been made on the teeth of the driving gear, out of which one is smaller and the other one is bigger. An intense impact is realized at the entrance of these teeth, and it becomes intensified at coming of the tooth with a bigger defect. The photographs of the gearbox in the state of testing are presented in Figure 6.

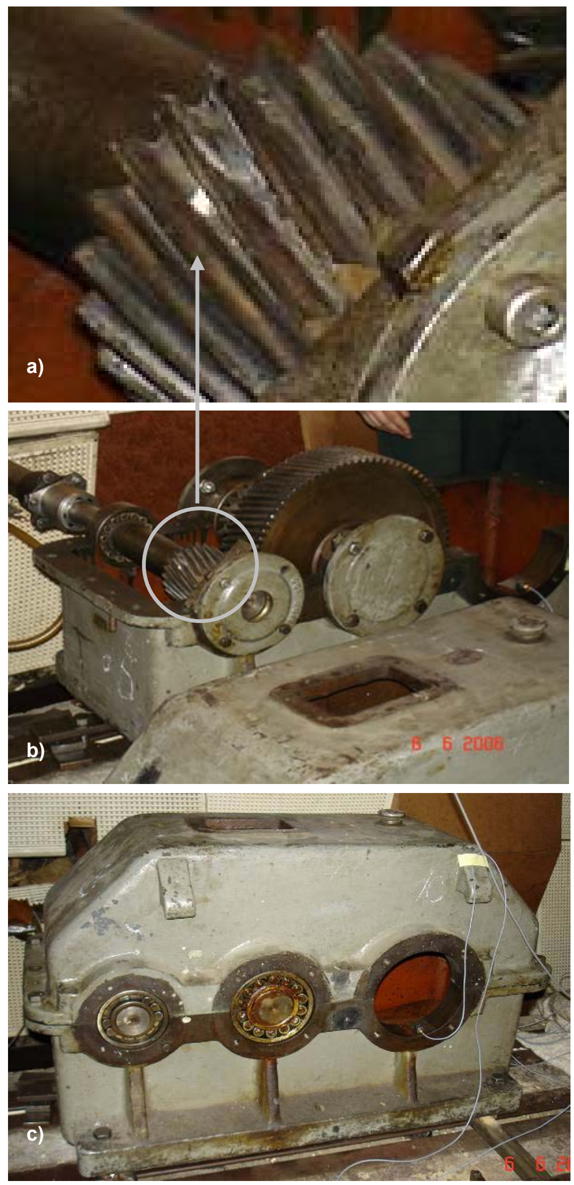
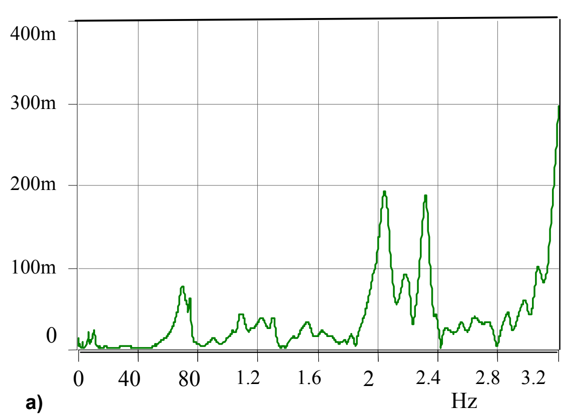


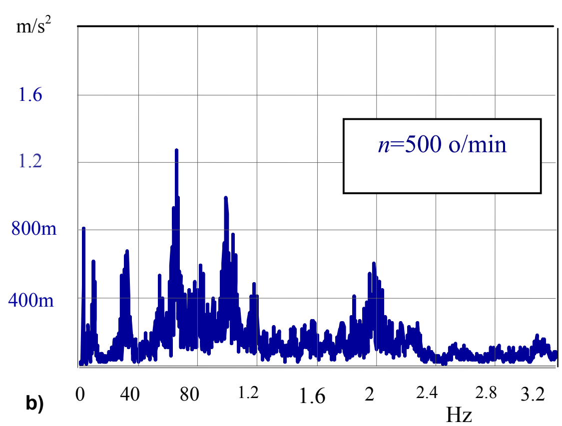

**Figure 6. Gearbox testing: a) gears with defects, b) housing without cover, c) closed housing**

### 4.1 Correlation between modal response and vibrations of the housing

The modal response of the housing (Fig. 7a) has been measured at the lateral side between two bigger holes for the bearings on the lower part of the gearbox housing, and it has been obtained by impulse excitation by means of a modal hammer, which means by the impact at the point where the bearing is supported, i.e. at the point where real disturbance at bearing operation is introduced. By comparing this response with the results of the modal analysis performed by means of FEM, it can be concluded that correspondence is satisfactory but that there are certain deviations. The comparison between numerical and experimental modal responses is not the subject of this paper – it compares the spectrum of housing responses at modal testing with the spectrum of vibrations measured at the same point (Fig. 7b) at meshing of gears with defects, at the number of revolutions n=500 o/min. Frequencies of vibrations are mostly the same in both spectrums. Therefore, it can be concluded that the gearbox housing vibrations are the consequence of natural oscillation and that the intensities of vibration responses (Fig. 7b) approximately correspond to the intensities of response at modal testing (Fig. 7a). The differences which are still present can be the subject of other, broader considerations.




a)




b)

**Figure 7. Spectrums of gearbox housing vibrations: a) at modal testing, b) at meshing of gears with defects**

The third conclusion which can be made on the basis of the spectrum of vibrations in Figure 7b refers to the frequency of excitation. The frequency of excitation is 16Hz for the rotation speed of 500 o/min and for two defects at the driving gear. The intensity of vibrations for this frequency is important but not the highest one. The intensity of disturbance energy which is entered into the elastic structure depends upon it.

### 4.2 Correlation between gearbox housing vibrations and noise

The next experiment should confirm the thesis that the noise emitted into the surroundings by the gearbox is the consequence of natural oscillation of the housing. For that purpose, Figure 8 presents two spectrums of noise, one referring to gears with defects/damages, at 500o/min (Fig. 8a) and the other one referring to the closed gearbox with the same gears and the same speed of rotation (Fig. 8b). By comparing these spectrums with each other and with the spectrums of vibrations in Figure 7, the following can be concluded:

- A very complex spectrum of noise of open gears has been obtained. They are mainly primary sound waves and their higher harmonics that have arisen by impact of teeth with defects
- The housing walls oscillate with natural frequencies and emit sound waves both into the surroundings and into the interior of the housing itself. The spectrum of acoustic pressure expressed in mPa (Fig. 8b) has been measured at the distance of 0.5 m above the closed gearbox. The levels for the frequencies equal to the natural frequencies of the housing are emphasized in this spectrum. This confirms the above mentioned thesis that natural frequencies of the housing are dominant in the sound spectrum.

Besides, the housing with its insulation should lead to reduction of the level of acoustic pressure in comparison with the noise of the gear without the housing (Fig. 8a). However, this has not happened. Significantly increased noise levels have been obtained. At the housing, there is a considerably big open hole. The housing has acted as a resonator
Box.

The angle speed of the driving shaft also significantly influences the level of the noise emitted. The change of speed results in the change of excitation frequency, the change of disturbance energy absorbed and the change of the level of noise for corresponding natural frequencies of the housing.

## 5. CONCLUSION

The methodology of identification of the power transmitter noise structure has been developed.

The initial hypothesis has been confirmed that the modal structure of natural oscillation of the power transmitter housing and the spectrum of the noise emitted are in full correlation.

Disturbance energy in teeth meshes excites vibrations of the transmission system (gears, shafts,

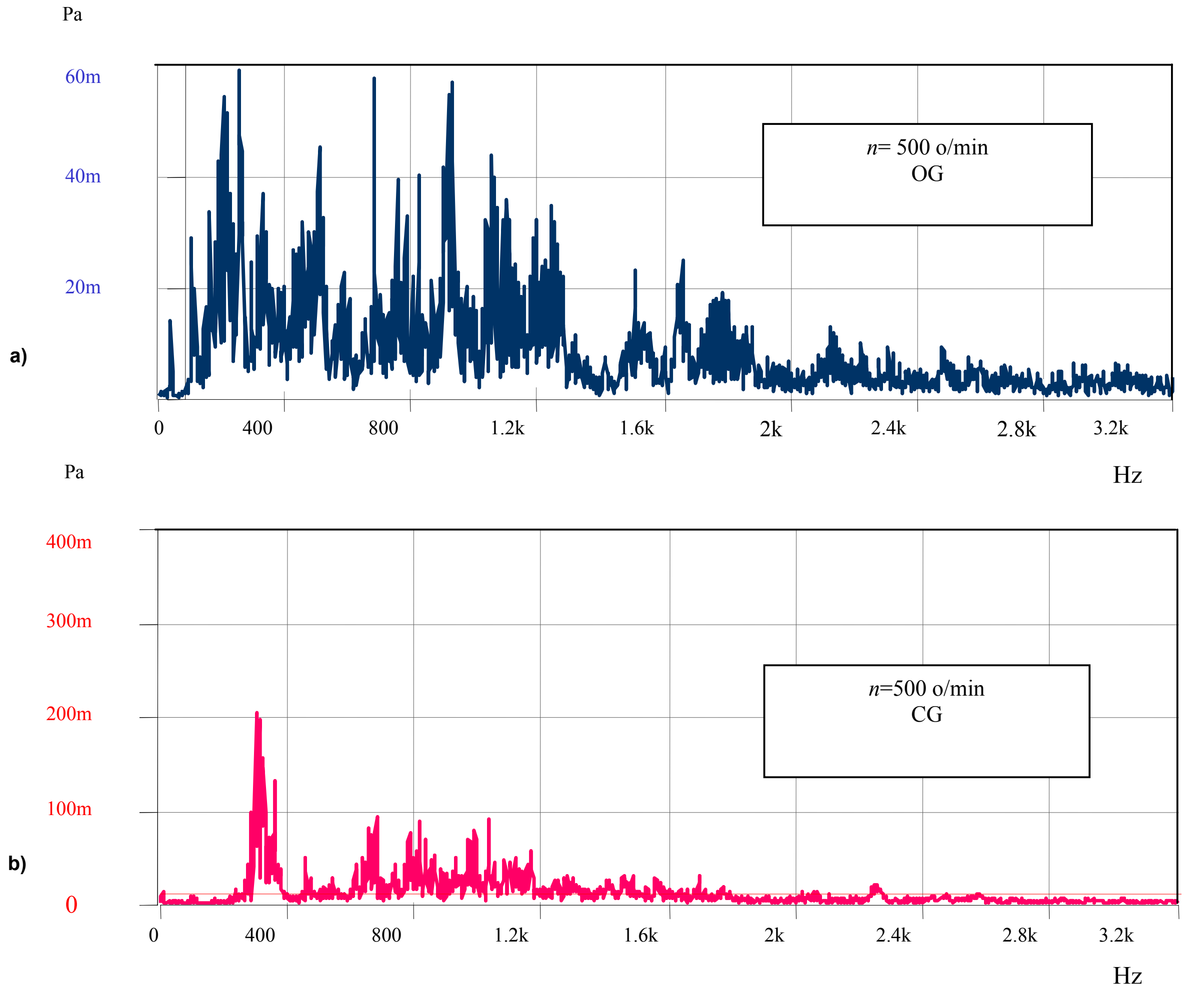



**Figure 8. Spectrums of transmission unit noise: a) open gearbox , b) closed gearbox**

bearings) as well as natural vibrations of the housing. The housing walls act as an insulator of primary sound waves, as transmission of secondary sound waves and as a generator of tertiary (structural) sound waves.

Detailed separation of these sound waves can be performed by a deeper analysis of the structure of the spectrums presented.

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## СТРУКТУРА БУКЕ ЗУПЧАНИХ ПРЕНОСНИКА СНАГЕ

**Снежана Ћирић Костић, Милосав Огњановић**

Емисија буке зупчаних преносника снаге (редуктора) зависи како од енергије поремећаја која се апсорбује при спрезању зупчаника, раду лежаја и др., тако и од изолационих могућности и модалног понашања кућишта. Терцијални звучни таласи настали услед сопственог (модалног) осциловања зидова кућишта могу по интензитету значајно да премаше примарне и секундарне звучне таласе услед спрезања зубаца. Простирање побудне енргије кроз структуру преносника и модално тестирање у циљу одређивања сопствених фреквенција и могућих облика осциловања кућишта су корак ка дефинисању утицаја кућишта на општи ниво буке редуктора. Мерење вибрација и буке са упоређивањем резултата добијених модалном анализом (МКЕ [2]) и модалним тестирањем, омогућавају идентификацију структуре буке изабраног редуктора. Упоређивањем са резултатаима мерења вибрација и буке дефинисан је механизам формирања структуре емитоване буке.