IDENTIFICATION OF VIBROACOUSTIC FIELD OF COMPLEX MECHANICAL STRUCTURES ON THE EXAMPLE OF A TOOTHED GEAR

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Summary

In the paper global and partial measures of acoustic efficiency including the total and partial values of acoustic power of the sound emitted from definite parts of the considered structure were defined. Usefulness of the above measures of the acoustic efficiency for purposes of investigating the identification of sound sources of complex mechanical structures has been analyzed basing on a study of a single-stage gear transmission model.

Keywords: vibroacoustic diagnostics, noise sources, vibroacoustic efficiency.

IDENTYFIKACJA POLA WIBROAKUSTYCZNEGO ZŁOŻONYCH STRUKTUR MECHANICZNYCH NA PRZYKŁADZIE PRZEKŁADNI ZĘBATEJ

Streszczenie

W pracy zdefiniowano miary globalnej i cząstkowej sprawności wibroakustycznej, które zawierają całkowitą i cząstkowe moce akustyczne dźwięku emitowanego z określonych obszarów analizowanej struktury. Przydatność zdefiniowanych miar sprawności akustycznej do badań identyfikacji źródeł dźwięku w złożonych układach mechanicznych analizowano na podstawie badań pola akustycznego modelu jednostopniowej przekładni zębatej.

Słowa kluczowe: diagnostyka wibroakustyczna, źródła hałasu, sprawność wibroakustyczna.

1. INTRODUCTION

A few measures of acoustic efficiency were proposed for the analysis of vibroacoustic energy distribution in a near external acoustic field of complex mechanical structures. The defined measures include global and partial acoustic powers of the sound emitted from definite areas of the analyzed structure.

The power N of an acoustic wave emitted by a source equals to an integral over a surface S surrounding the source from a dot product of a vector of sound intensity $\vec{I}(r)$ and a related elementary surface vector $d\vec{S}(r)$ [1, 2, 3]:

$$N = \int_S \vec{I}(r)\, d\vec{S}(r) \qquad [\mathrm{W}], \tag{1}$$

where: $dN = \vec{I}(r) \cdot d\vec{S}$, $\vec{I} = E\big(p(r,t)\vec{v}(r,t)\big)$, $E(\ )$ - is a time-averaging operator, $p(r,t)$ – a value of acoustic pressure, $\vec{v}(r,t)$ – velocity of a particle at the point related to $p(r,t)$.

The acoustic power of the source may be determined basing on measurements of a normal component of sound intensity and acoustic pressure in the middle of every segment of the surface surrounding the source (point by point, one after another – in accordance with PN-EN ISO 9614-1 [1] or by scanning – in accordance with PN-EN ISO 9614-2 [4,5]). Hence, the partial acoustic power N_i is a time-averaged flux of acoustic energy I_{ni} flowing through a segment of the measurement surface S_i:

$$N_i = I_{ni} S_i \tag{2}$$

Thus, the measurement of the sound intensity on the surface surrounding the source of stationary noise enables determination of the acoustic power of the source or partial sources situated inside the space limited by the measurement surface.

The vibroacoustic efficiency η_{WA} is defined as a quotient of vibroacoustic power of a sound source N_{WA} (its mechanical structure) and power N_D delivered to the structure. In the model of power distribution in a mechanical structure it was assumed that the delivered power N_D used in a widely understood technological process consists of the effective power N_U, internally dissipated power N_S and externally dissipated power N_R according to equation (1):

$$N_D = N_U + N_S + N_R \quad , \tag{3}$$

where: $N_R = N_{WA} + N_I$, N_{WA} – the power of vibroacoustic processes, N_I – the power of other accompanying processes, e.g. thermal, diffusion and magnetic processes.

Taking the foregoing assumption into account, the vibroacoustic efficiency may be presented in the form (4):

$$\eta_{WA} = \frac{1}{\frac{N_U + N_S + N_I}{N_{WA}} + 1} \quad . \tag{4}$$

The relationship (4) enables assessment of contribution of the power of residual processes, i.e. the vibroacoustic ones, with respect to the delivered power N_D.

For identification of dominating sources of sound in a given structure and comparative analysis of partial vibroacoustic processes both with reference to characteristic frequencies of the analyzed object and for evaluation of acoustic energy distribution from a spatial perspective a global and partial acoustic efficiencies of mechanical structures were defined.

2. GLOBAL AND PARTIAL ACOUSTIC EFFICIENCIES OF MECHANICAL STRUCTURES

Partial acoustic efficiencies of a sound source – a mechanical structure are defined as follows:

1. The ratio of time-averaged acoustic energy flux in the i[th] third-octave band flowing through the N_{ijk} measuring surface segment to the averaged acoustic energy flux flowing through the same surface segment N_{jk}:

$$\eta_i = \frac{N_{ijk}}{N_{jk}} , \quad N_{jk} = \sum_i N_{ijk} , \tag{5}$$

   where i – is for the i[th] third-octave band, j,k – coordinates of the middle point of the measurement surface segment,

2. The ratio of time–averaged acoustic energy flux in the i[th] third-octave band flowing through the N_{ijk} measuring surface segment to the acoustic energy flux averaged in the same i[th] third-octave band flowing through a certain area or the whole measurement surface N_i (e.g. if the measurement is carried out at a cuboid surface, N_i is the acoustic energy flux averaged in the i[th] third-octave band flowing through this surface or through one of the walls of this cuboid):

$$\eta_{jk} = \frac{N_{ijk}}{N_i} , \quad N_i = \sum_j \sum_k N_{ijk} , \tag{6}$$

3. The ratio of time–averaged acoustic energy flux in the i[th] third-octave band flowing through the N_{ijk} measuring surface segment to the averaged acoustic energy flux flowing through a certain area or the whole measurement surface N (e.g. if the measurement is carried out at a cuboid surface, N is the time-averaged acoustic energy flux flowing through this surface or through one of the walls of this cuboid):

$$\eta_{ijk} = \frac{N_{ijk}}{N} , \quad N = \sum_i \sum_j \sum_k N_{ijk} \tag{7}$$

The global acoustic efficiency of a sound source – a mechanical structure is the ratio of time-averaged acoustic energy flux in the whole band of the measured frequencies flowing through the N_{jk} measuring surface segment to the averaged acoustic energy flux flowing through a certain area or the whole measurement surface:

$$\eta_G = \frac{N_{jk}}{N} \tag{8}$$

Thus, determination of the values of the individual measures of acoustic efficiency requires knowledge of acoustic power as a function of frequency of the wave flowing through measurement surfaces and their segments surrounding the investigated structure.

3. IDENTIFICATION OF A VIBRO-ACOUSTIC FIELD OF A TOOTHED GEAR MODEL

Usefulness of the foregoing measures of acoustic efficiency for identification of sound sources in complex mechanical structures was analyzed basing on the investigations of a single-stage toothed gear transmission model.

The research model of a single-stage toothed gear transmission (figure 1) consisted of two straight cylindrical gears, two equal shafts with spacing sleeves of outside diameter ϕ63 mm and pins ϕ40 mm. The shafts were fixed in ball bearings ϕ40/ϕ80/18 mm. The width of the toothed wheel rims amounted to 32 mm.
The welded casing of the gearbox was made of 5 mm thick sheet steel. The rings of the bearing seats were 9 mm thick. The casing lugs were made of 10 mm sheet steel.

During investigations the gearbox casing was not filled with oil and left open – without any cover. The assumption of such research conditions and the disproportion between the dimensions of the shafts and casing walls increased the predominant role of vibrations of the thin casing walls and generation of sound in the range of lower forcing frequencies.

For measurements of the acoustic power of the gearbox using intensity method were used a probe B&K 3584 with a 50 mm separator and an analyzer B&K 2145. The sound intensity level was measured in third octave bands with mid-band frequencies from 31.5 Hz to 1250 Hz. In this frequency range the error of sound intensity level approximation did not exceed 1 dB.

The acoustic power of the gear transmission was determined basing on the measurements of normal component of sound intensity I_{ni} and acoustic pressure p_i in the middle of each segment of the surface surrounding the source (point by point – according to PN-EN ISO 9614-1 [2]). Partial acoustic power N_i is defined as a time-averaged flux of acoustic energy I_{ni} flowing through a measurement surface segment S_i – relationship (2).

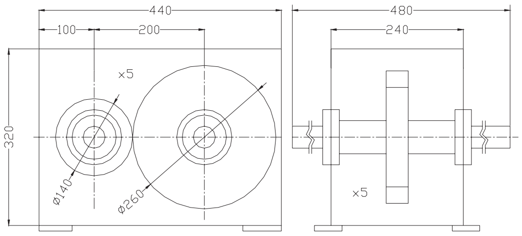

Fig. 1. Main dimensions of the gear transmission model

The measurement surfaces surrounding the gear transmission were parallel to the casing walls (at the distance of 250 mm measured from the middle of the intensity probe). It was defined by the casing walls (wall surfaces A, B, C) (figure 2).

Parameters of the individual measurement nets at the walls A, B i C (measurement surface segments) are listed in table 1.

The acoustic efficiency was investigated for two rotational speeds of the motor: 750 rpm and 1290 rpm (that corresponds to the frequencies f_1 = 12,5 Hz and f_2 = 21,5 Hz respectively).

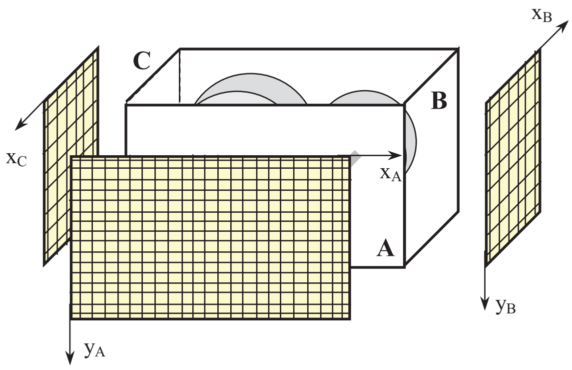

Fig. 2. The model of a gear transmission with the assumed denotations for walls and measurement surfaces which were assigned to them

Table 1. Parameters of the nets of the measurement surfaces surrounding the gear transmission during the measurements of acoustic power using the sound intensity method

Measurement surface dimensions [mm]	Measurement net dimensions [mm]	Distance between net elements and wall edges [mm]
A 440x320	20x20	20 from the left and right 10 from the bottom and top
B, C 240x320	40x40	20 from the left and right; 20 from the bottom and top

The results of the investigations are shown in the form of surface distributions of the global value and partial acoustic efficiencies for the individual third-octave bands. The surface distribution of the global acoustic efficiency η_G for measurement surfaces A, B and C is shown in figure 3.

Because of large sets of test results, figure 4 shows as an example only selected surface distributions of the values of partial acoustic efficiencies for the rotational speed of the motor of 1290 rpm, for three walls A, B and C of the cuboid surrounding the transmission, and for the third octave band with the mid-band frequency of 500 Hz.

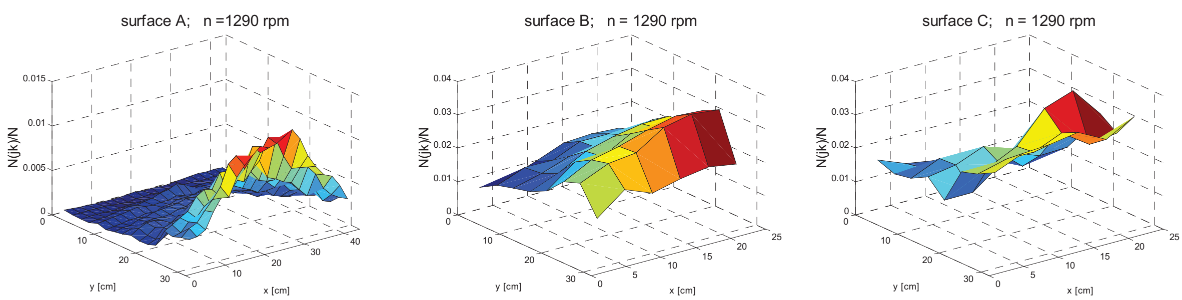

Fig. 3. Global acoustic efficiency η_G for the walls A, B and C of the gear transmission model

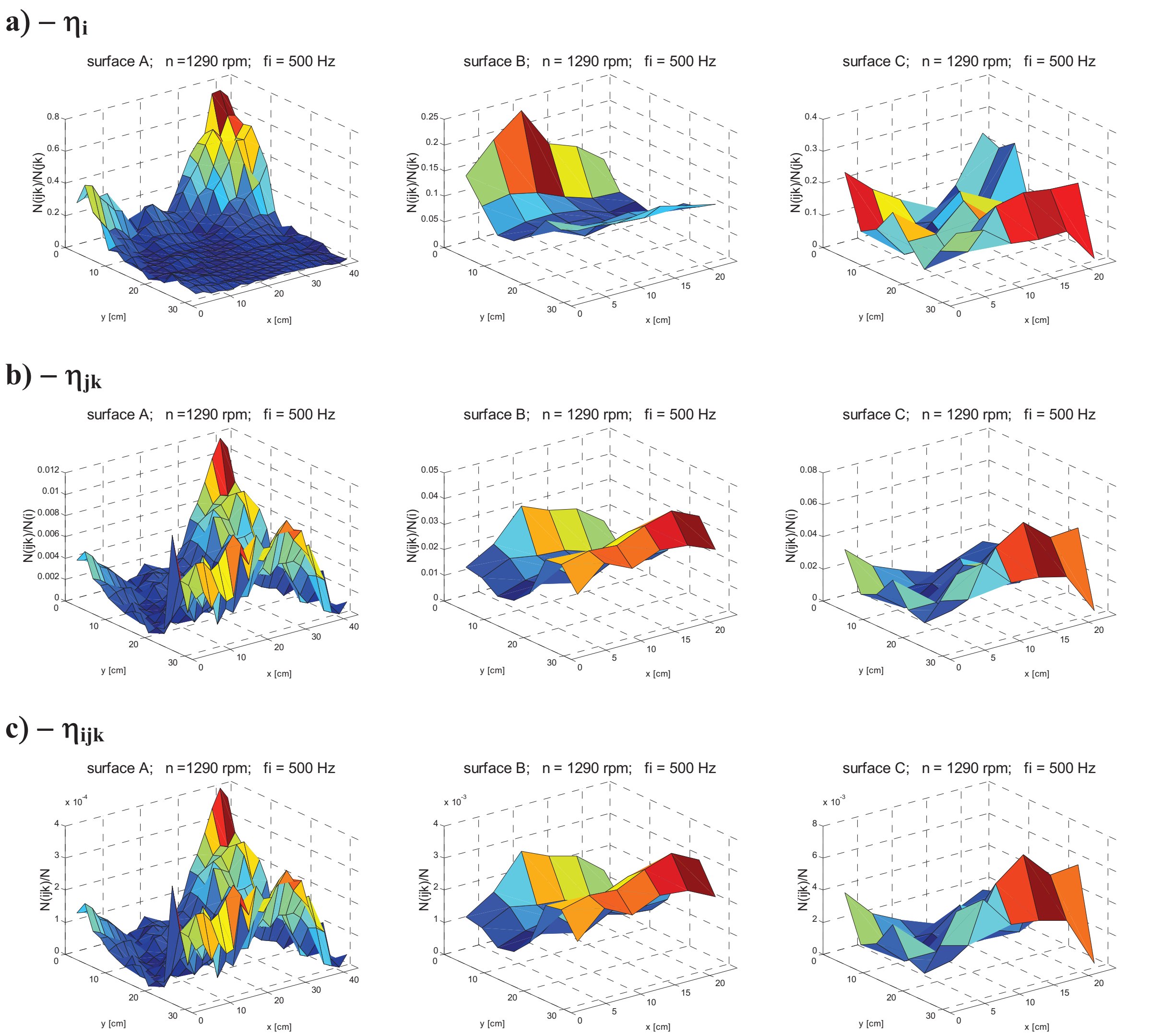

Fig. 4. Surface distributions of partial measure values of acoustic efficiency for the third octave band with mid-band frequency of 500 Hz; walls A, B and C of the gear transmission model; a) – η_i, b) – η_{jk}, c) – η_{ijk}

Both the observed diverse dynamics of the partial measure values of acoustic efficiency for the given third octave band (figure 4) and the measure of the global acoustic efficiency (figure 3) indicate unambiguously that the defined measures carry different diagnostic information about the investigated object.

4. SUMMARY

The analysis of the surface distributions of the acoustic efficiency values as a function of frequency enabled to identify the predominant sound sources in the investigated model of the gear transmission.

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