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Numerično modeliranje notranjih zvokov v železniških vozilih

Numerical Modelling of the Internal Sound in Railway Rolling Stock

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V prispevku je predstavljena analiza zvoka v železniških vozilih. S tehničnim razvojem železniških vozil se povečuje tudi hitrost vlakov, zato se povečuje tudi jakost zvoka, ki pri tem nastane. Hrup je ena izmed škodljivih posledic transporta. Ker je v bivalnih okoljih raven zvoka eden izmed pomembnejših pokazateljev bivalnega ugodja, se povečuje pozornost namenjena raziskavam poteka zvoka. V prispevku so predstavljeni zvok, ki ga povzročajo prevozna sredstva, njegovo širjenje in njegovo spreminjanje v notranjih prostorih vozil ter različni viri zvoka pri železniškem transportu. Obravnavani so tehnični, organizacijski in administrativni ukrepi za zmanjašnje nastanka zvoka. Prav tako pa so prikazani tudi škodljivi vplivi zvoka na zdravje ljudi. Za raziskavo smo izbrali dve vrsti ruskih lokomotiv M62 in TEP60. Z uporabo programske opreme ANSYS/Multiphysic smo simulirali zvok motorjev z različno močjo. Rezultate preračunov zvoka smo nato primerjali z dovoljenimi vrednostmi.

(Ključne besede: železniška vozila, generiranje zvoka, numerično modeliranje, parametri dušenja)

The paper presents an analysis of the sound in railway rolling stock. The speeds of trains tend to increase as railway rolling stock improves technically and becomes more sophisticated; however, the sound generated is intensified too. Noise is one of the harmful consequences of transport. As the sound level in domestic surroundings is one of the most important human-comfort indices, increasing attention is being devoted to an investigation of sound processes. The sound generated by transport vehicles, its distribution, and its variation in internal transport-vehicle spaces, and various sound sources in railway transport are considered in the paper. Engineering, organizational and administrative sound-reducing measures are discussed. The harmful effects of sound on human health are discussed as well. Two types of Russian production locomotives - the M62 and the TEP60 - were selected for this research. Using ANSYS/Multiphysic software the acoustic sound of the different power units in the engine sector was simulated. The results of the sound calculations were compared with the acceptable levels.

(Keywords: railway rolling stock, sound analysis, numerical modelling, damping parameters)

0 INTRODUCTION

Noise pollution is an increasing nuisance in the EU Member States. The EU Commission, in an attempt to pursue measures against sound problems, has in recent years intensified its activities relating to sound abatement. If sound limits are exceeded for railways, several major factors are considered: the condition of the rail and the wheels, the type of train, etc. But it is certainly also a question of the specific prediction method used. These prediction methods differ in many ways in various parts of Europe.

Noise transmission in locomotives can be difficult to study due to the complexity of the locomotive's structure, and the difficulties in characterizing the excitation or describing the response. Earlier work [1] has shown that structural vibration transmission can be predicted fairly accurately using a statistical energy analysis.

At high frequencies the engine's sound can be a problem in locomotives. Fan sound, combustion events, power-train vibration, road-wheel sound and aerodynamic sound can also input power to the vehicle structure at frequencies above 250 Hz. At these frequencies a statistical energy analysis can be used to study the transmission of sound ([1] to [6]).

The objective of the paper was to analyse the internal sound pollution in existing locomotives in Lithuania today. The speed of the trains is not high, and at low speeds the locomotive sound is the dominating source. We have chosen two types of locomotives – the freight locomotive M62 and the passenger locomotive TEP60. In this paper we present the results of modelling the internal sound of these locomotives.

1 THEORETICAL ANALYSIS

A functional scheme of the existing harmful factors affecting human health in railway transport is presented in Fig. 1.

The standards define the sound level of the rolling stock. The sound of the rolling stock is limited by requisition No 692 of the Minister of Health of the Lithuanian Republic. The equivalent sound levels in railway-transport working places are defined in this requisition (Table 1) ([7] and [8]).

The sound sources in a locomotive are grouped into three categories:

I – the sound arising from the motion devices – shoes, axle boxes with roller bearings, brake levers, traction engines, traction reducers and axle wheels.

II – the sound inside the control cabin. This is the sound caused by the speedo meter, the engine driver's crane, the whistle and the watchfulness signal.

Table 1. *Equivalent sound levels in the railway-transport working place [7]*

Object	***Equivalent noise level, dB***
Cabins of the operators of steam and diesel locomotives	*80*
Cabins of the operators of suburban electric locomotives	*75*
Rooms of personnel in the wagons of long-distance communication trains	*60*
Service rooms in electric stations, cooling sections	*60*
Service rooms in luggage and post wagons	*70*
Relaxation rooms in luggage and post wagons	*60*
Service rooms in the restaurant wagons	*70*

III – the sound inside the machine section. The main sources of sound in the machine section are the diesel power aggregate, the sound caused by ventilators, the reducers of the auxiliary aggregates, the main electricity generator, the double machine aggregate, the breaking compressor, the turbo compressor and the exhaust system for the combustion products.

The axle-wheel sound is caused by the interaction between the rail irregularities and the bandage rolling on the rail head surface as well as by the sliding of the wheel along the rail in longitudinal and lateral directions. The vibrations of the bandage and the wheel centre can cause a wideband spectrum sound level up to 120 dB. The axle-wheel sound may be drowned out by the gear sound when movement speeds are low and loads are large.

The rolling sound largely depends on the speed of the rolling stock. Normally, the sound pressure rate increases by 9 dB with a doubling of the speed. However, this wheel-based sound may be different with regard to the type of rolling stock. The irregularities of the interacting surfaces agitate the vibrations of the wheel and rail under the influence of the masses inherent in the movement. The vibra-

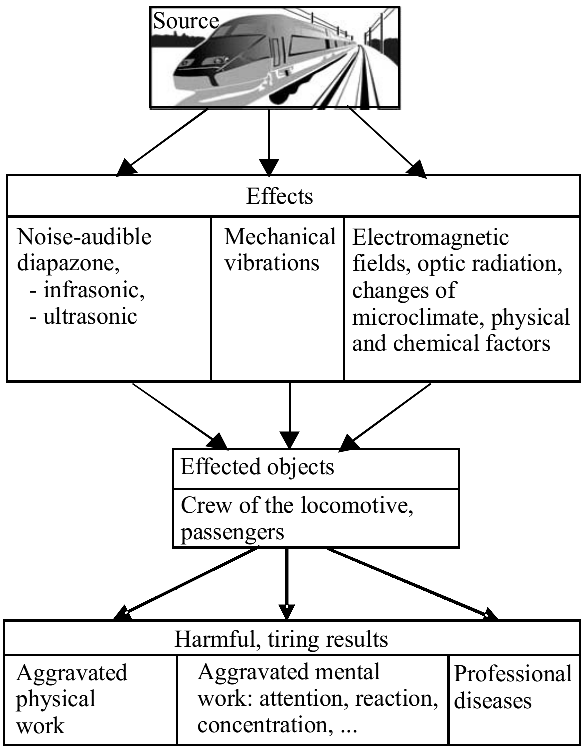

Fig. 1 *Scheme of factors harmful to human health*

tions of different amplitudes are agitated in the wheel and rail, depending on the properties of the materials and the geometries of the rail and the wheel. The motions of these surfaces cause the vibrations of the air, thus inducing the ambient air sound.

The sound in railway-transport facilities is measured using the procedure established in the following standards:

ISO 1996/1-1982 "Acoustics – measurement and sound description in ambience. Part I. The main parameters and measurement methods";

ISO 1996/2-1987 "Acoustics – measurement and sound description in ambience. Part II. Collection of data relevant to land usage";

ISO 1996/3-1987 "Acoustics – measurement and sound description in ambience. Part III. Application of sound limitation".

The general diagram can be conveniently represented by signal graphs, as shown in Fig. 2. The joints in the graph represent variable energy flows (sources), and the arcs represent the sound-energy-transmitting channels defined by the reduction indices of the sound-energy intensity ([9] to [11]).

Referring to the signal graph represented in Fig. 2 a sound power flow J_{is} in the analysed direction of the sound field of the rolling stock is written as follows:

$$J_{is} = W_1C_1 + W_2C_2 + ... + W_kC_k = \sum_{i=1}^{i=k} W_iC_i \qquad i = 1,2,3....,k \qquad (1)$$

The value of the intensity rate L_{is} is equal to the sound pressure rate expressed in dB, as follows:

$$L_{is} = 10\lg {J_{is}}/{J_0} = 10\lg \sum_{i=1}^{i=k} {W_iC_i}/{J_0} \le [L_{is}] \qquad (2),$$

where W_1, W_2 ..., W_k are the sound capacities of the sound sources; C_1, C_2, ..., C_k are the indices involving intensity reduction with the increase of the distance from the sound sources; J_0 is the limiting value of the sound intensity, $J_0 = 10^{-12}$ W/m^2; $[L_{is}]$ is the rate of permissible external sound.

Two methods for determining the sound energy in the rolling-stock cabin are used (Fig. 2 b and c): from each source via all the elements of the cabin, or from all sources via each element of the cabin. For the first calculation method the sound energy in the cabin is expressed as follows:

$$W'_{Ki} = W_i\left(K'_1\tau'_1 + K'_2\tau'_2 + ... + K'_m\tau'_m\right) = W_i\sum_{i=1}^{i=m} K'_m\tau'_m \qquad (3).$$

For the second method of calculation the expression is:

$$W''_{Km} = \left(W_1K''_1 + W_2K''_2 + ... + W_kK''_k\right)\tau''_m = \tau''_m\sum_{i=1}^{i=k} W_kK''_k \qquad (4),$$

where K'_1, K'_2,... and K''_1, K''_2,... are the indices evaluating the transfer of sound energy to the surfaces of the relevant partitions; τ'_1, τ'_2,... and τ''_1, τ''_2,... are the indices evaluating the transfer of sound energy through the relevant partitions.

The number of signal graphs in the diagram being calculated is defined by the number of sound sources being evaluated, as well as by the number of elements that are homogeneous according to the sound permeability for all the surfaces of the cabin.

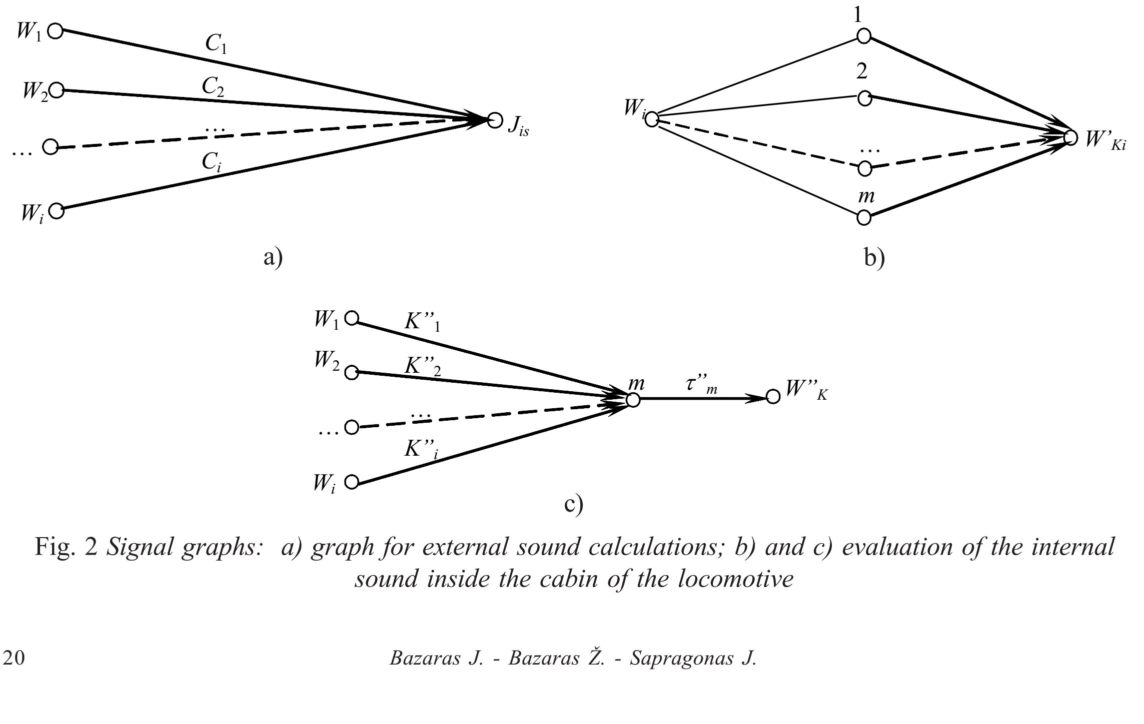

Fig. 2 *Signal graphs: a) graph for external sound calculations; b) and c) evaluation of the internal sound inside the cabin of the locomotive*

The calculations are carried out according to the corrected and octave sound capacity values by evaluating the relevant values of the transfer indices τ – the sound permeability through partitions. Evaluating the total sound intensity in the cabin gives:

$$J_k = \frac{\sum W_k}{\bar{\alpha} S_\Sigma} \quad (5),$$

where $\sum W_K$ is the total sound energy of the cabin, calculated using the following formula

$$\sum W_K = \sum_{i=1}^{i} W'_{Ki} = \sum_{i=1}^{m} W''_{Km} \quad (6),$$

where $\bar{\alpha}$ is the average sound-absorption coefficient in the cabin and the body; S_Σ is the area of the internal surface of the cabin and the body.

The sound rate (sound-pressure ratio) in the cabin of the rolling-stock is calculated using the formula:

$$L_K = 10\lg \sum W_K \Big/ \left(\alpha \bar{S_\Sigma} J_0\right) \le [L_K] \quad (7),$$

where $[L_K]$ is the allowed sound rate in the cabin.

The transmission indices C_i, which evaluate the reduction of the sound intensity with the increase of the distance from the point of the sound source, are determined using the relationship:

$$C_i = \frac{1}{\Omega r_i^2} \quad (8),$$

where r_i is the distance of the i-th sound source from the field-sound source point of the rolling-stock cabin under consideration; $\Omega = 4\pi$ for spherical sound radiation, and $\Omega = 2\pi$ for semi-spherical sound radiation (semi-spherical sound radiation will be for $H \le r_i/3$, here H is the agreed point-source height above the railroad surface).

The transmission indices K_i in Equations (3) and (4) are calculated using the formula:

$$K_i = \frac{S_i}{\Omega r_i^2} \quad (9),$$

where S_i is the surface area of the partition.

The index of the sound-energy transmission through the partitions (sound permeability coefficients) is calculated as:

$$\tau_i = 10^{-0.1 R_i} \quad (10),$$

where R_i is the sound isolation of the partition.

The presented acoustic-calculation model of a rolling-stock cabin allows an evaluation of the structural solutions and, in the case of an emergency, taking extra measures in the process of rolling-stock design.

2 METHODS OF EVALUATION

2.1 Complex evaluation of the impact of sound on the environment

For the modelling of the internal sound in the locomotive we used ANSYS software to create a 2D model of the internal space of the locomotive body. The geometry of the model consists of four different parts:

1. the internal space of the front control cabin;
2. the internal space of the back control cabin;
3. communicating tambours to the internal space of the machine section;
4. the internal space of the machine sector, which, for convenience, when forming the finite-elements grid, was divided into four areas

In the ANSYS/Multiphysic software the problems of acoustics are solved on the basis of a harmonic response analysis by providing the harmonic pressure agitation (sinus type) at some points of fluid structure and obtaining the pressure distribution in the fluid. By changing the agitation frequency, a variable sound distribution in the interval of different frequencies is obtained.

The stages of the harmonic acoustic analysis are as follows:

1. Formation of the model
2. Identification of the boundary conditions and the acoustic loads as well as a solution of the finite-element model
3. Review of the results

2.2 Limit conditions of the model and loads

When designing the calculation diagram for the front locomotive a planar structure of finite elements was used. The whole structure was described by FLUID29 2D finite elements designed for a specified acoustic analysis. These acoustic elements have the following degrees of freedom: displacements UX, UY and pressure PRES. For the acoustic FLUID29 finite elements the following characteristics of the material need to be specified: the air density, DENS; the sound velocity in the air, SONC; and the damping index, MU. Also, the real constant is to be indicated, i.e., the sound pressure value taken as an audible limit – $p_0 = 2 \cdot 10^{-5}$ Pa. During the creation of the

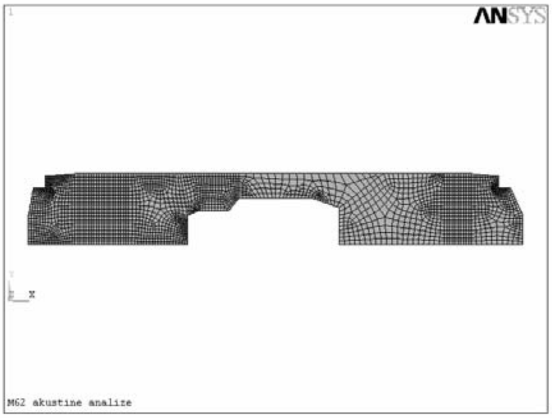

Fig. 3. *Finite-element grid of the M62 locomotive*

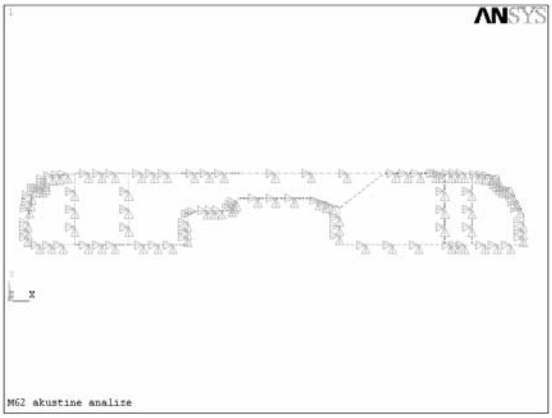

Fig. 4. *Boundary conditions of the M62 locomotive model*

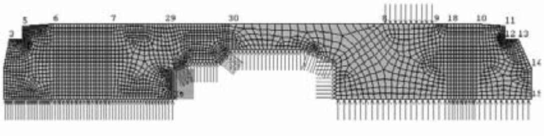

Fig. 5. *Excitation places of the fluid in the M62 locomotive model*

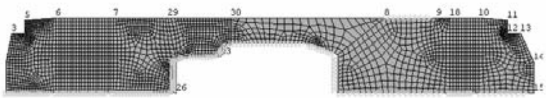

Fig. 6. *Acoustic loads of the M62 locomotive*

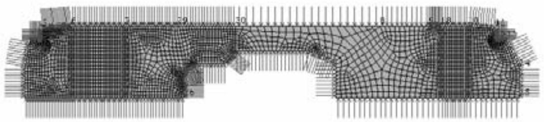

Fig. 7. *Energy-damping surfaces of the M62 locomotive model*

finite elements' grid (Fig. 3) and the tightening of the body walls, ceilings and floor junctions as well as in the interior body-space linear displacements (Fig. 4), the interaction surfaces of the fluid and a solid structure are indicated.

These surfaces are the floor surface of the locomotive (loaded by outside sound, caused by the rolling of the axle-wheels) as well as the agreed surfaces of the diesel engine and the fan (Fig. 5). On the surfaces of the model, the interaction between the vibrating structures and fluid particles occurs. Also, the acoustic load, i.e., the harmoniously varying pressure (Fig. 6), corresponding to the sound pressure levels, existing on these surfaces and represented in Table 2, is also indicated on these surfaces ([7] to [11]).

Table 2. *Noise pressure at various locations of the M62 locomotive [7]*

Surface	Noise level dB	Pressure Pa
Floor of vehicle, when speed v is km/h:		
30	*90*	*0.632*
60	*97*	*1.420*
90	*103*	*2.830*
120	*111*	*7.100*
Engine	*120*	*20*
Fan	*104*	*3.17*

2.3 The solution of the model formed

Depending on the sound frequency, the isolation materials as well as the interior elements' damping characteristics vary as shown in Table 3. Therefore, the calculations were performed by varying both the agitation frequency in terms of the internal geometric frequencies of the octave bands and by accordingly changing the damping coefficients of the parameters defining the sound-energy damping surfaces of the model.

2.4 Boundary conditions of the model, conditions of loading and the solution

By forming the grid of finite elements, and restricting the linear displacements in the junctions

Table 3. *Noise pressure at various places in the TEP60 locomotive [7]*

Surface	Noise level dB	Pressure Pa
Floor of the vehicle, when speed v, km/h:		
≤120	*96*	*1.260*
120≤v≤160	*105*	*3.560*
Engine	*120*	*20*
Ventilator	*109*	*5.64*

Table 4. *Noise damping parameters [8]*

Frequency Hz	Engine room	Cabin
31.5	*0.04*	*0.17*
63	*0.04*	*0.17*
125	*0.04*	*0.17*
250	*0.04*	*0.26*
500	*0.04*	*0.64*
1000	*0.04*	*0.89*
2000	*0.04*	*0.75*
4000	*0.04*	*0.78*

of the body walls, ceilings and floors as well as in the internal space of the body, by indicating the interaction surfaces of the fluid and solid structure and by marking the damping surfaces, the model of the finite elements of the passenger locomotive, shown in Fig. 7, is obtained. The variations of the damping parameters and the loads are presented in Tables 3 and 4.

The calculations were carried out when changing the agitation frequency and damping parameters in an order analogous to the one presented in Table 4.

One of the essential indices of human comfort is the level of sound in the working and domestic environment. The permissible sound level in thermal trucks, according to OSZhD recommendations, is the following ([10] to [12]):

- long-term sound, N80 = 80 dB,
- interrupted sound, N85 = 85 dB.

3 RESULTS

The sound levels in the control cabins of the M62 and TEP60 locomotives are not the same when driving at the maximum allowed speed of 120 km/h due to the difference in power (for the TEP60, N_e =2237 kW; for the M62, N_e=1421 kW) (Fig. 8). The internal sound in the cabins of these locomotives exceeds the permissible values.

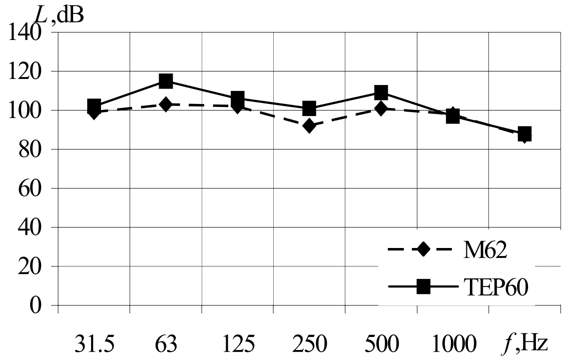

Fig. 8. *Comparison of the sound levels in the cabins of the M62 and TEP60 locomotives when the velocity is 120 km/h*

Fig. 9 shows the change of the sound level *L* in the cabin of the M62 locomotive when the sound of the riding wheels increases. The sound increase in the cabin is the result of the increasing velocity of the train.

In Fig. 10 the change of the sound inside the body of the M62 locomotive is presented in the sound frequency range 125 to 1000 Hz. In the machine section, where the damping of sound waves is minimal, the maximum sound pressure is formed, and it reached 122 dB. Such a working environment is harmful to human health. Even a short time spent in such an environment can cause damage to the ear. The sound spreading from the machine sector and the external sound coming into the cabin are damped by the wall, floor and ceiling sound isolation.

4 CONCLUSIONS

1. The sound in the high-frequency range of the rolling stock used in Lithuania exceeds the leg-

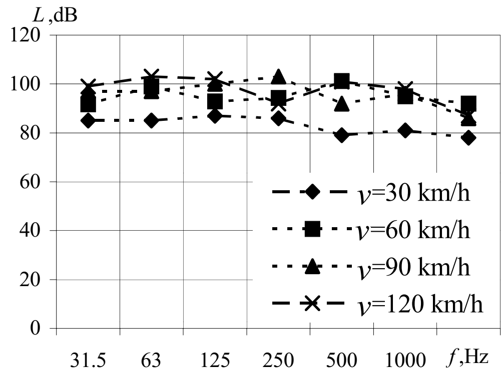

Fig. 9. *Change of sound level in the cabin of the M62 locomotive driving at 30 to 120 km/h*

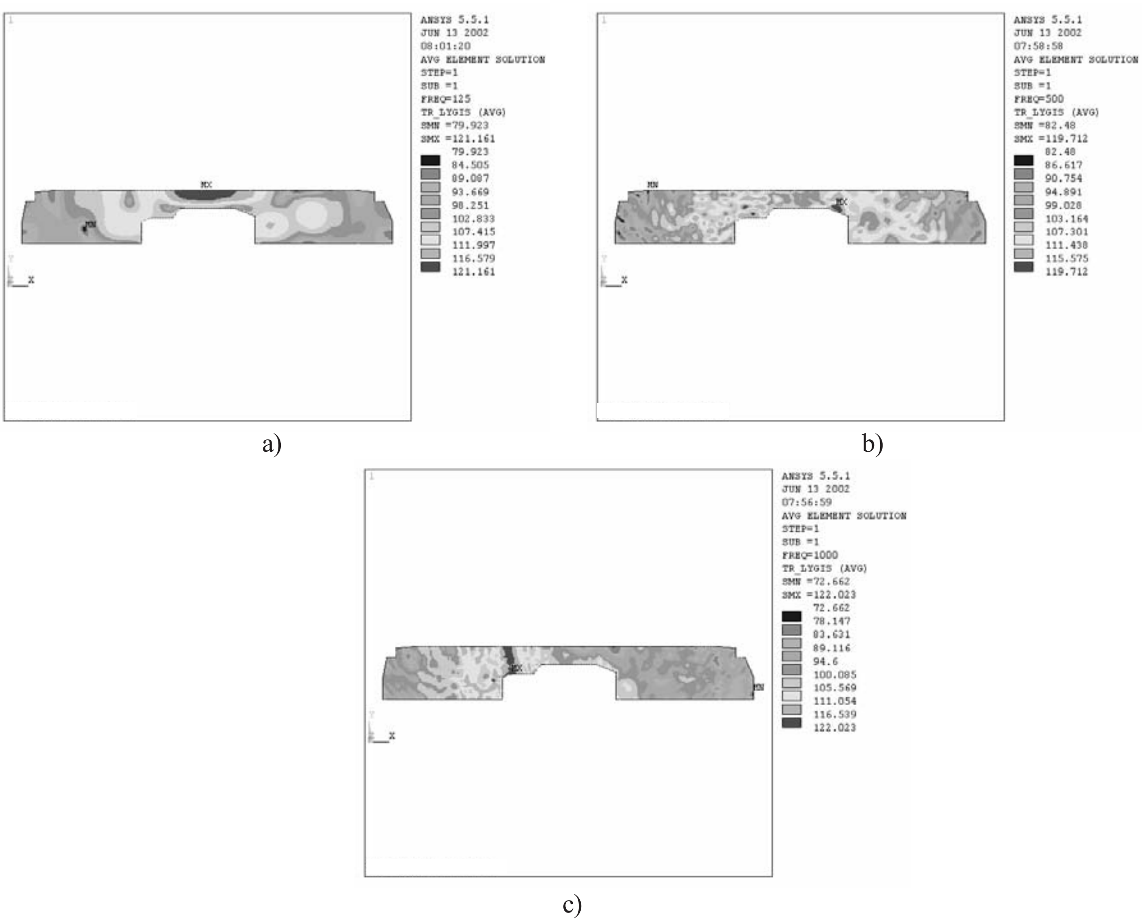

Fig. 10. *Change of the sound level [db], when the excitation frequency is: a) 125 Hz; b) 500 Hz; c) 1000 Hz*

islated levels by 5 to 25 dB. The main sources of sound pollution, such as the main and additional power aggregates, the road-wheel interaction, the braking equipment, and the sound-isolation equipment, have a mechanical character.

2. The highest sound level in the control cabin of the TEP60 passenger locomotive is 106 dB; in the M62 locomotive it is 106 dB, in the machine section of the TEP60 locomotive it is 120 dB, and in the M60 locomotive it is 125 dB in the 31.5 to 250 Hz range of sound frequencies.

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