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To cite this article: A I Taratorkin *et al* 2020 *IOP Conf. Ser.: Mater. Sci. Eng.* **709** 022112

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Theoretical and experimental optimization of vibro-acoustic parameters of MKSM-800 loader

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Abstract. This paper presents the results of a theoretical and experimental study of the vibro-acoustic radiation of the mini loader (MKSM-800), and analyzes the variant of reducing the noise level both of the loader on the whole and of its specific structural components. It estimates the vibro-acoustic activity of an internal combustion engine (ICE), including the cooling system, and the left and right side reduction gear. On the basis of the dynamic analysis of the mechanical system structures, the most dangerous forms of oscillations are determined, they contribute to the integral level of acoustic radiation most of all. This waveshape damping is achieved by targeted ribbing of the structural components of the reduction gear, as well as by reducing the vibroactivity of the excitation source by increasing the precision degree in gear manufacturing.

1. Introduction

For the recent decades, designers of road construction and community service vehicles have been actively involved in noise reduction. In this class of vehicles, the main sources of noise are the processes of mechanical and hydrodynamic origin. First of all, it is the noise from the vibration of stationary base members, gas exchange and engine cooling systems, transmission units, as well as the noise arising during processing equipment operation. Some manufacturers claim outstanding achievements in this field. For example, Bucher-Schörling large refuse trucks operate with a noise level not exceeding 70 dBA. At the same time, Russian road construction and community service vehicles, for example, MKSM-800, are characterized by an increased level of acoustic radiation. In this regard, the problem of reducing the level of acoustic radiation of road-construction and community service vehicles is topical. The purpose of this study is to develop substantiated technical solutions to reduce the level of acoustic radiation of the MKSM-800 loader on the basis of theoretical and experimental research.

The works of many researchers from around the world are devoted to the analysis of variants for reducing a vehicle noise level on the whole and specific structural components of the vehicle. First of all, the vibro-acoustic activity of an internal combustion engine (ICE), including its cooling system, as well as reduction gear units and specific elements of the hydraulic drive system, is evaluated. The sources of noise from diesel engines are mainly due to periodic operation of its cylinders, to the inlet and exhaust manifolds, to the design of protective housings, to the mechanical dynamics of the fans,



etc. These factors are studied in the works [1-2]. The factors that form the vibro-acoustic activity of the reduction gear units are determined by the structural parameters of the gears and the modal characteristics of the reduction gear casing, which interact with each other through bearing supports. At the same time, the noise emitted by the reduction gear is one of the main sources that leads to a violation of ergonomic requirements, both for the driver and for the environment. In accordance with the standard classification, the acoustic radiation of gearboxes, reduction gears is divided into three types, i.e. gearwhine, rattle and clunk.

Rattle and clunk results from dynamic system imperfections, manifested in resonance excitation or from control system imperfections, also leading to their occurrence. These phenomena have been studied by many researchers, including the authors. The study of oscillations in the pre-torque-converter zone of the hydromechanical transmission, the conditions for exciting resonances in a nonlinear system, and development of the methods for their damping are given in [3]. The methods for eliminating dynamic loads using nonlinear energy absorbers (NonlinearEnergySinks (NESs)) in meshing of gear teeth to eliminate vibrations are described in [4].

Gearwhine results from imperfections of gear geometry and their supporting components. This type of radiation is one of the top priority manifestations of the NVH phenomenon and is characterized by the complexity of developing constructive and technological measures for its elimination [5-6]. This leads to the fact that the vehicle engineers have to rely on acoustic radiation testing, which requires a significant investment of time and money. At the same time, the mathematical and software tools describing this phenomenon is constantly being improved. A detailed study of the effects of microgeometry changes in meshing of gear teeth was studied in [6]. To forecast gear wheel dynamics and transmission errors, gear engagement designs are simulated based on the dynamics of several rigid bodies, and analyses are carried out along with gear wheel testing at the testing bench [6].

It is concluded that imperfection of the geometric parameters of the lateral surface of the tooth is the main prerequisite for gearwhine occurrence. In [7], the simulations are performed to decrease the noise of the reduction gear by means of transmission error reduction by modifying the gear profile. Many other works include experimental and analytical studies that attempt to reduce the level of excitation.

However, only few researchers suggest reducing the response level of a dynamic system (the transmission case) to excitation. At the same time, the acoustic radiation of the transmission arises directly on its vibrating surfaces, which exercises a function of membranes that transmit the effects of internal forces as an audible sound. These forces occur when transmitting torsional and bending moments in meshing of gear teeth, radial and axial forces in bearing supports in steady and transient modes of operation, as well as in the event of resonance excitation. Transmission vibration is directly transmitted to the vehicle structure. At the same time, it can become obvious and tangible in the form of tactile sensations on various elements of the vehicle structure (steering wheel, seats, glazing, etc.).

2. The strategy of theoretical and experimental optimization of NVH parameters

In this work, to achieve the goal of the research, the strategy of theoretical and experimental optimization of MKSM-800 NVH parameters is used. Strategy implementation is carried out when conducting experimental studies with application of a set of data measuring equipment, including the SCADAS recorder of vibro-acoustic signals and acoustic array, LMS HD AcousticCamera. The elements modeling of the object of study is carried out in the LMS Virtual.Lab software package [8], which is a comprehensive set of software products for 3D finite element dynamic analysis of multimodular object structures.

3. Evaluation of NVH parameters when conducting experimental studies

Acoustic tests are carried out as follows. The loader is installed on the testing bench and is put in working condition, that is, a regular load is provided in the working range of revolutions and moments of resistance.

Table 1. Test Modes.

# mode	Acoustic array is installed at the side surface of the powertrain unit
1	Idling, $\alpha_f = 0$, $\beta = 0$, clutch disengaged.
2	Idling, $\alpha_f = 0$, $\beta = 0$, clutch engaged.
3	Idling, $\alpha_f = 2/3$, $\beta = 0$, clutch engaged.
4	Idling, $\alpha_f = 2/3$, $\beta = 0$, clutch disengaged.
5	Idling, $\alpha_f = 1$, $\beta = 0$, clutch engaged.
6	Idling, $\alpha_f = 1$, $\beta = 0$, clutch disengaged.
7	Running test, $\alpha_f = 2/3$, $\beta = 0,5$
8	Running test, $\alpha_f = 2/3$, $\beta = 1$
9	Running test, $\alpha_f = 1$, $\beta = 0,5$
10	Running test, $\alpha_f = 1$, $\beta = 1$
Acoustic array is installed in the rear part of the vehicle	
11	Idling, $\alpha_f = 0$, $\beta = 0$
12	Idling, $\alpha_f = 1$, $\beta = 0$
13	Running test, $\alpha_f = 1$, $\beta = 1$
14	Idling, $\alpha_f = 1$, $\beta = 0$, the air duct is removed from the muffler
15	Idling, $\alpha_f = 0$, $\beta = 0$, the air duct is removed from the muffler (the hood door is open)
16	Idling, $\alpha_f = 1$, $\beta = 0$, exhaust pipe removed (the hood door is open)

With the clutch engaged, the vibro-acoustic zones were determined on the side surface of the vehicle powertrain unit and in the rear part in three high-speed engine operating modes (n_i , $2/3n_{\max}$ and n_{\max}). It has been established that in all high-speed modes of the engine operation the main zone of acoustic radiation is concentrated in the area of location of the guide vanes of the engine cooling system fan (figure 1, below). In the same figure, the amplitude-frequency characteristic (Figure 1, above) shows three pronounced maxima, which unambiguously correlate with the rotation speed of the engine cooling system fan. At idle, the level of acoustic loading in the rear part is 71,73 dBA. At engine speed n_{\max} , the noise level was 88,35 dBA. When installing a measuring acoustic array at the side surface of the powertrain unit, the level of acoustic pressure is 66,06 dBA (n_i), 82,31 dBA ($2/3n_{\max}$) and 85,29 dBA (n_{\max}). It should be noted that the highest level of acoustic pressure is also observed in the rear part of the vehicle at the same frequencies. When disengaging the clutch (disengaging the transmission), the engine speed increases slightly by 6 ... 7%. This leads to a corresponding increase in the level of acoustic pressure by 1,5 ... 2 dBA.

When the MKSM loader operates under load (a braking device is connected to the front flange of the right side reduction gear), the nature of vibroacoustic loading changes significantly. In this case, the dominant source of acoustic radiation is still the fan of the engine cooling system. The spectral composition formed by this source remains the same with a slight increase in the amplitude of the acoustic pressure. When installing a measuring acoustic array at the side surface of the powertrain unit, the level of acoustic pressure is 84,38 dBA ($2/3n_{\max}$, $\beta = 0,5$), 91,14 dBA ($2/3n_{\max}$, $\beta = 1$), 86,99 dBA (n_{\max} , $\beta = 0,5$) and 87,11 dBA (n_{\max} , $\beta = 1$).

Thus, the level of acoustic pressure in the frequency range of the suction apparatus during the vehicle operation under load has changed slightly. At the same time, the frequency spectrum of acoustic radiation with a significant level of amplitude has significantly expanded due to the waveshapes of the reduction gear casing, excited by the meshing of gear teeth of the reduction gear under load. Additional frequencies are in the range of 200 to 5500 Hz with amplitude levels of 70 dBA and higher. In this case, the amplitudes of acoustic radiation of specific oscillation modes reach 100 dBA (for example, at a frequency of 5470 Hz, the amplitude is 96,92 dBA, figure 2).

To fulfill the requirements of regulatory documents (CH 2.2.4 / 2.1.8.562-96 and GOST R 50631-91 - the sound radiation level should not exceed 83 dBA), according to the results of the conducted

experimental studies, it is established that there is a necessity to perform the following experimental design works aimed at reducing the level of acoustic radiation, a follows:

- to develop measures to optimize the parameters of the guide vanes of the engine cooling system;
- to investigate the influence of the gear teeth characteristics of the reduction gear on the level of external acoustic pressure;
- to perform a dynamic analysis of the reduction gear structural design in order to optimize the modal characteristics of its body.

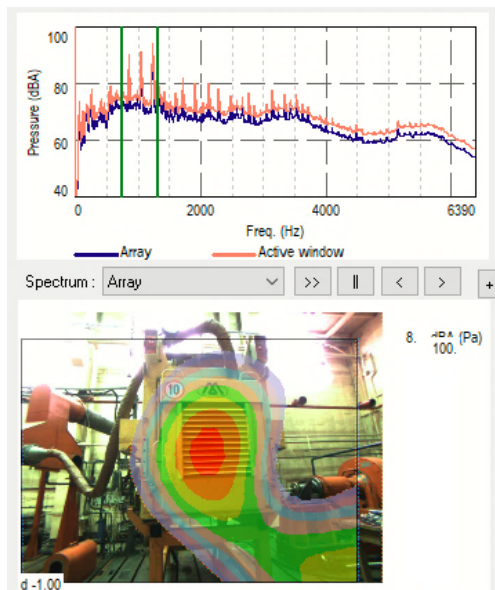


Figure 1. Illustration of the sequence of acoustic fields localization based on the obtained data (file 15.bdd, $\alpha_f = 1$ (n_{max}), $\beta = 0$ (the moment is zero), the clutch is engaged, the hood is closed).

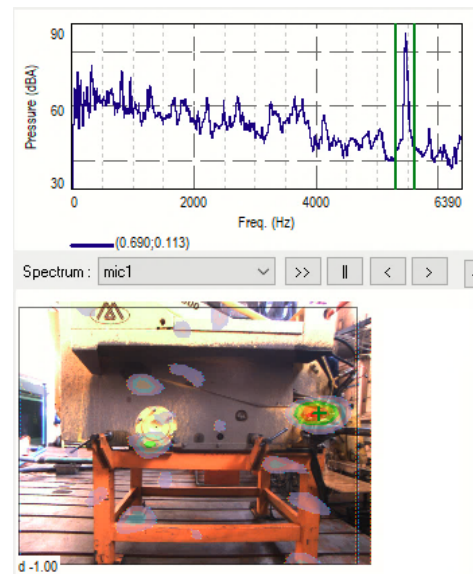


Figure 2. Illustration of the sequence of acoustic fields localization based on the obtained data (file 8.bdd, $\alpha_f = 2/3$ ($2/3n_{max}$), $\beta = 0,5$ (partial load), the clutch is engaged).

4. Vibroacoustic modeling

The process of noise and vibrations generation and propagation can be represented in the form of a block diagram presented in figure 3

To achieve the goal of the research, in accordance with the diagram, the modeling (optimization) strategy for NVH parameters of the reduction gear under consideration is applied. The strategy is implemented in the LMS Virtual.Lab software package, which is a comprehensive set of software products for 3D finite element analysis and design of multi-module objects, for modeling and optimization of mechanical systems when analyzing the structure of vibro-acoustics indicators.

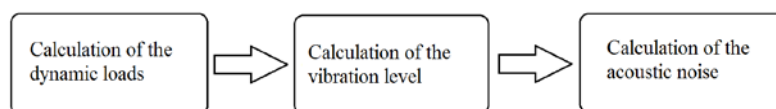


Figure 3. Block diagram of the generation and propagation of noise and vibration.

The calculation was carried out in three stages:

- 1) Calculation of the dynamic loads transmitted to the gearbox casing through bearing supports. A variable response in meshing of gear teeth was adopted as the main source of loads. At this stage, gearing parameters are varied in order to reduce the intensity of the generated excitation.

2) Calculation of the vibration level of the gearbox casing based on the findings of the modal analysis and determination of the oscillation propagation paths.

3) Calculation of acoustic noise. Vibrational waveshapes of the gearbox casing walls, calculated at the previous stage, were taken as the main source of external noise.

The calculation was carried out for two variants of the pair of gear wheels - A and C, and two variants of the reduction gear casing – of the original and ribbed ones. The second variant differs from the original one by introducing reinforcement ribs to damp vibration waveshapes with the greatest acoustic contribution. The identification of the most significant wave shapes was carried out on the basis of integral coefficient calculation of the waveform contribution to the total sound pressure level taking into account the considered modes.

To determine the objective function (of an integral sound pressure level), the significance of vibration waveshapes was estimated according to a ten-point scale for different rotational speeds of the reduction input shaft. The maximum magnitude of the summation of points for the n^{th} waveshapes determined the necessity for making structural changes to damp this waveshape contribution into the overall integral sound pressure level. To damp the most dangerous vibrational waveshapes, the ribbing of the covers, side surfaces and the bottom of the reduction gear is introduced. Visualizing simulation results in the form of waterfall diagrams (dependences of the acoustic response at the desired points for each rotational speed of the input shaft) for the four calculation variants is shown in figures 4 - 7.

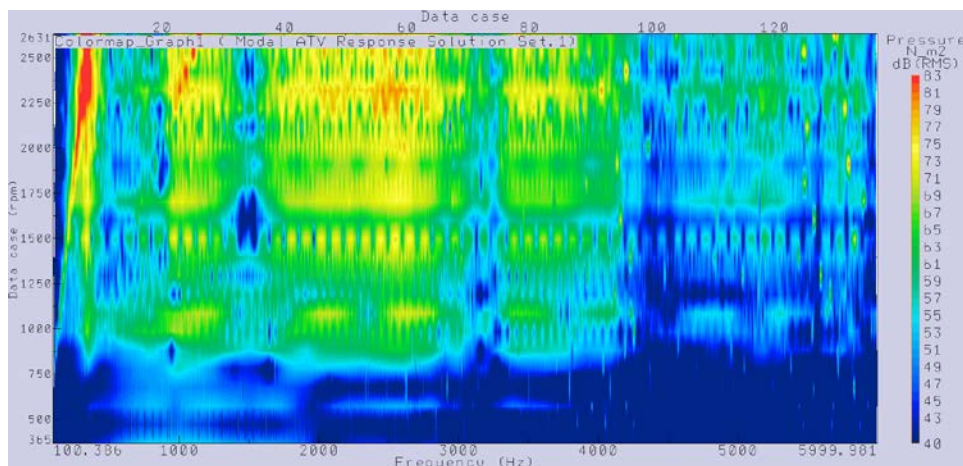


Figure 4. Waterfall diagram, pairing type - A, reduction gear casing – original.

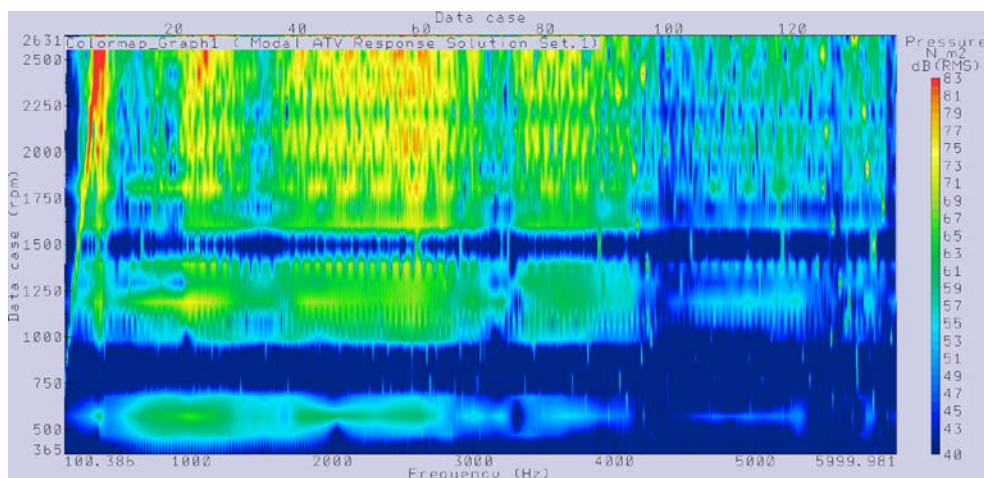


Figure 5. Waterfall diagram, pairing type - C, reduction gear casing – original.

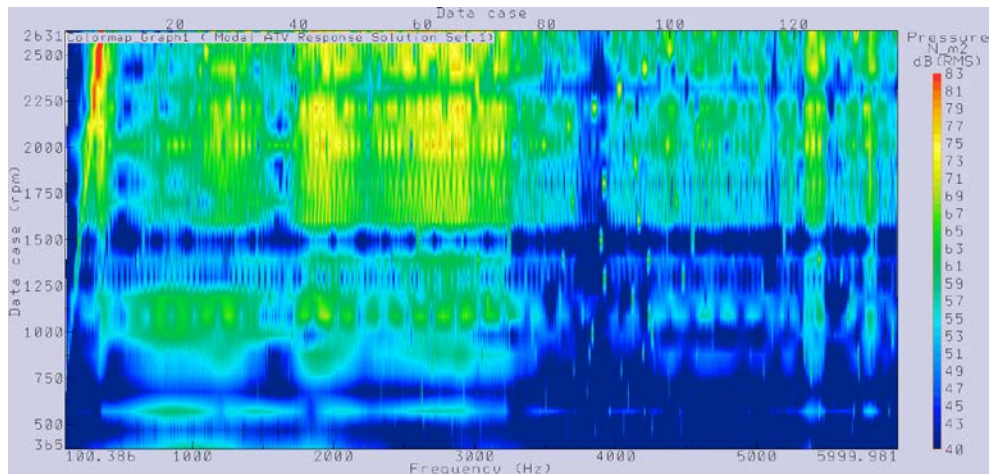


Figure 6. Waterfall diagram, pairing type - A, reduction gear casing – ribbed.

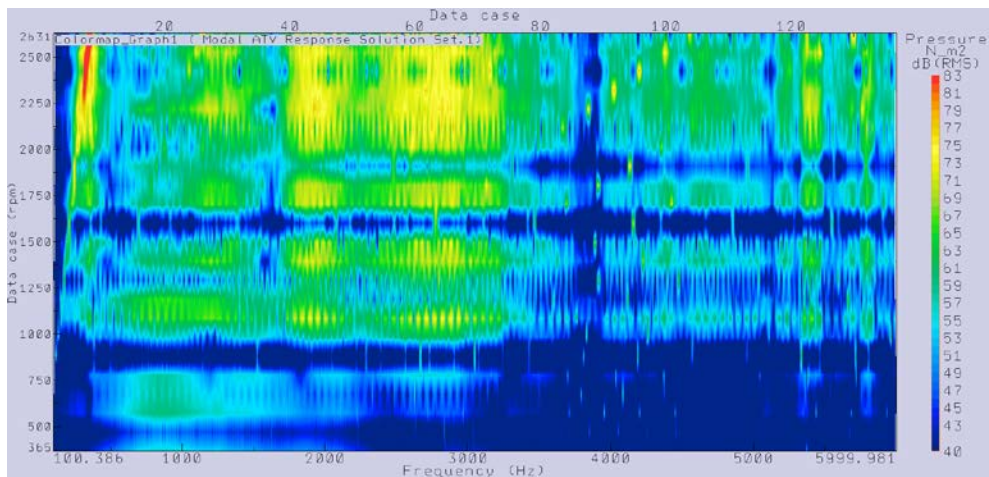


Figure 7. Waterfall diagram, pairing type - C, reduction gear casing – ribbed.

As follows from figures 4 - 7, each of the proposed changes significantly reduces the noise level. The introduction of a pairing type with a lower guaranteed side backlash and targeted ribbing of the reduction gear configuration made it possible to reduce the sound pressure level to 83 dBA in the basic modes of operation.

Conclusions

In the course of the experimental and computational studies it is established:

- the highest level of vibroacoustic radiation is created by the cooling system fan of the internal combustion engine. The frequency range varies from 400 to 1300 Hz, the amplitudes depending on the rotation frequency of the impeller (engine speed) reach 94 dBA;
- when the shafts are loaded with a moment, the frequency spectrum of acoustic radiation with a significant amplitude level expands significantly due to occurrence of vibration waveshapes of the reduction gear casing excited by meshing of gear teeth under load. Additional frequencies are in the range from 200 to 5500 Hz with an amplitude level of 70 dBA. In this case, the amplitudes of acoustic radiation of individual vibration waveshapes reach 100 dBA (for example, at a frequency of 5470 Hz, the amplitude is 96,92 dBA);

- Introduction of the pairing type with a lower guaranteed side backlash and targeted ribbing of the reduction gear configuration allowed for reducing the sound pressure level to 83 dBA in the basic modes of operation.

Acknowledgements

This study was funded by the program of the President of the Russian Federation for the support of young scientists - Grant MK-5809.2018.8, and also with partial support under the state task IES UB RAS No. 0391 - 2014 - 007.

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