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A contribution to the theoretical-experimental investigation of transverse vibrations of the controlling lever of a commercial motor vehicle

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ABSTRACT

During exploitation, commercial motor vehicles are exposed to various types of loads, among which vibrations are of particular importance. Vibrations lead to fatigue of vehicle users and materials of their components. Therefore, they must be studied in the earliest stages of design, using mathematical models, experiments, or their combinations.

In theoretical considerations, vibrations of concentrated masses are usually observed, although recently, with the development of numerical methods (especially finite element methods), attention has also been paid to the vibrations of elastic systems of these vehicles. In such cases, simplifications are usually made, especially regarding exploitation conditions and interconnections of vehicle components.

This paper attempts to develop a method for identifying vibrational loads on the controlling lever under exploitation conditions, using a two-parameter frequency analysis with the use of 2 D Fourier transform. The applicability of the procedure is illustrated on an idealized model of the controlling lever, and the conducted research has shown that the two-parameter frequency analysis can also be used to generate transverse vibrations in laboratory conditions.

KEYWORDS

Commercial motor vehicle, Controlling lever, Transverse vibrations, Two-parameter frequency analysis

1. INTRODUCTION

During exploitation, commercial motor vehicles are exposed to various types of loads, among which vibrations are of particular importance. Vibrations lead to fatigue of vehicle users and materials of their components. Therefore, they must be studied in the earliest stages of design, using mathematical models, experiments, or their combinations [1,2].

In theoretical considerations, vibrations of concentrated masses are usually observed, although recently, with the development of numerical methods (especially finite element methods [3-5]), attention has also been paid to the vibrations of elastic systems of vehicles. In such cases, simplifications are usually made, especially regarding exploitation conditions and interconnections of vehicle components.

The specificity of exploitation conditions for commercial motor vehicles is their random nature [6], which significantly complicates theoretical considerations using models, making experiments practically irreplaceable.

Despite significant progress in the development of software for automatic vehicle design and calculation [5], the final judgment on their characteristics is based on experimental research. Therefore, experimental methods remain important.

When it comes to the steering system of a commercial motor vehicle subject to random excitations causing random vibrations, the problem of identifying their parameters often arises. In this regard, methods for their identification have been developed, such as modal analysis [7,8]. In laboratory conditions, vibration modes are practically determined. However, a problem arises when real exploitation conditions are necessary to generate transverse vibrations of the controlling lever on test benches because modal analysis does not provide sufficient possibilities for generating signals in the time domain.

As known [9], during the calculation of the controlling lever, its bending is often checked. At the macro level, bending occurs due to the action of forces resulting from the torque at the steering wheel, and at the micro level, due to transverse vibrations of the controlling lever. Considering that a disturbing transverse force occurs in the lever of commercial motor vehicles, which originates from the variable torque at the steering wheel, it is necessary to study its transverse vibrations in more detail.

In this regard, an attempt has been made to develop a procedure for identifying parameters of transverse vibrations of the controlling lever, which will enable their generation in laboratory conditions based on frequency analysis [10]. Namely, if data obtained by Fourier transform are known, the application of inverse Fourier transform allows the generation of the original time-dependent signal, which is routinely performed in cases when the signal depends only on time [10].

However, vibrations of elastic systems depend on multiple parameters (dimensions and time) [11,12], which suggests that a multi-parameter Fourier transform should be used. In the case of an idealized model of the controlling lever, which is modeled as an elastic cantilever, transverse vibrations change along its length and depend on time, so-called two-parameter Fourier transform (2 D) should be applied [11].

This paper will analyze the possibilities of applying the two-parameter Fourier transform to create conditions for investigating transverse vibrations of the controlling lever in laboratory conditions. Since the expression for the Fourier transform in the case of multiple variables is given in [11,12], it will not be done here.

2. METHOD

As mentioned earlier, this paper aims to explore the possibility of using two-parameter frequency analysis (2 D Fourier transform) in identifying the parameters of transverse vibrations of the controlling lever of heavy motor vehicles. In the absence of experimental data of registered transverse vibrations of the controlling lever, the method is illustrated with data obtained from dynamic simulation using its mathematical model. It is considered appropriate to first analyze in detail the load that the controlling lever is subjected to during exploitation. For illustration purposes, Figure 1 shows a schematic representation of the position of the steering system on a heavy vehicle (the controlling lever is marked as A). Since a servo-hydraulic steering system is used in this type of vehicle, Figure 2 shows its scheme.

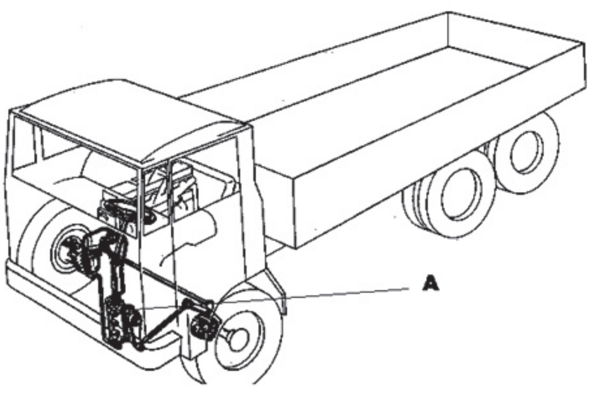

Figure 1: The installation scheme of the steering system on a heavy motor vehicle

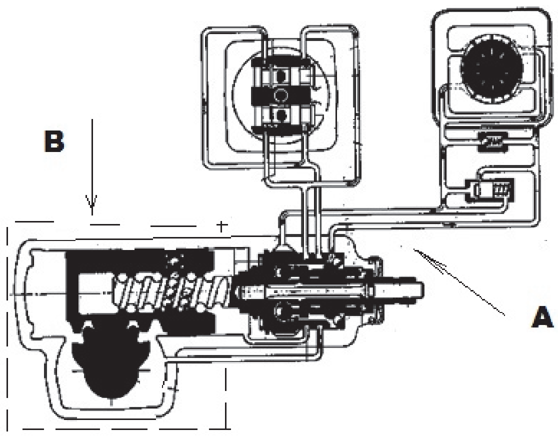

Figure 2. The scheme of the servo steering system

In Figure 2, the servo-hydraulic (A) and mechanical parts (B - highlighted separately by a rectangle with dashed lines) are marked. Its function is described in detail in [9], so it will not be discussed here. It should be noted that the servo-hydraulic and mechanical parts of the system work simultaneously when the entire servo-steering system is functioning properly.

In the event of a hydraulic failure, the system continues to operate but with poorer characteristics, so the driver must exert a greater torque on the steering wheel.

For further analysis, it is considered appropriate to show a simple chart diagram of the servo steering control system in Figure 3 [9].

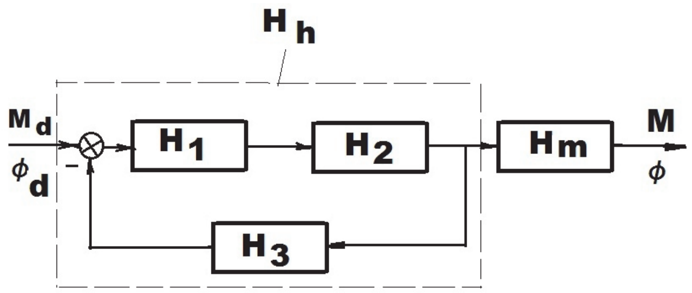

Figure 3. The chart diagram of the control system with servo steering

In Figure 3, the following notations are used: M_d - torque at the steering wheel, Φ_d - steering wheel angle, M - torque at the steering lever, Φ - steering lever angle, H_1 - transfer function of the servo valve, H_2 - transfer function of the executive system of the servo steering, H_3 - transfer function of the feedback, H_h - transfer function of the hydraulic part, H_m - transfer function of the mechanical part of the servo steering.

It should be noted that transfer functions depend on a large number of design parameters, the values of which are unknown because they are often protected by patents. Since the aforementioned transfer functions are described in detail in [13-17], it will not be done here.

In experimental measurement of the angle and torque at the steering wheel, the influence of the hydraulic part of the servo steering (H_h) is expressed through the registered values. If we go in reverse order so that the parameters at the output shaft are calculated based on the registered parameters at the steering wheel, then the influence of the hydraulic part is eliminated, so only the influence of the mechanical part of the system should be considered.

In this specific case, the torque and angle at the steering wheel were recorded while driving the FAP 1118 truck during driving on asphalt road, with a 4x4 wheel formula, equipped with the PPT 1145 power steering system. The results are shown in Figures 4 and 5 [18]. Since the procedure and testing conditions are described in detail in the mentioned reference, I will not go into further detail here.

Based on the data presented in Figures 4 and 5, it can be concluded that the recorded values of the torque and angle at the steering wheel belong to the group of random processes [10].

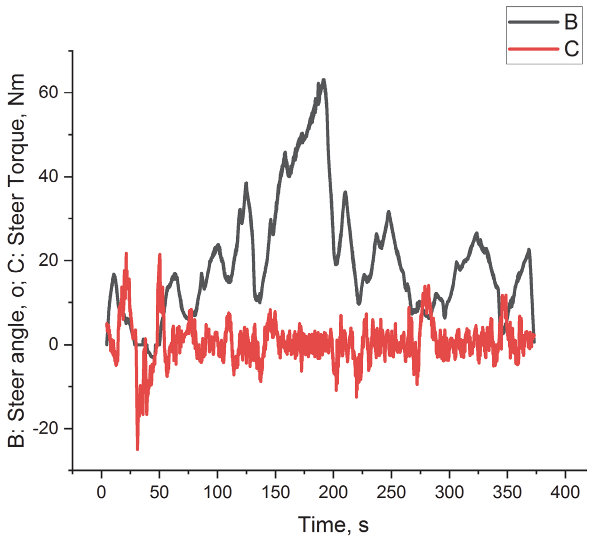

Figure 4: The time series of the angle and torque at the steering wheel during free driving on an asphalt road

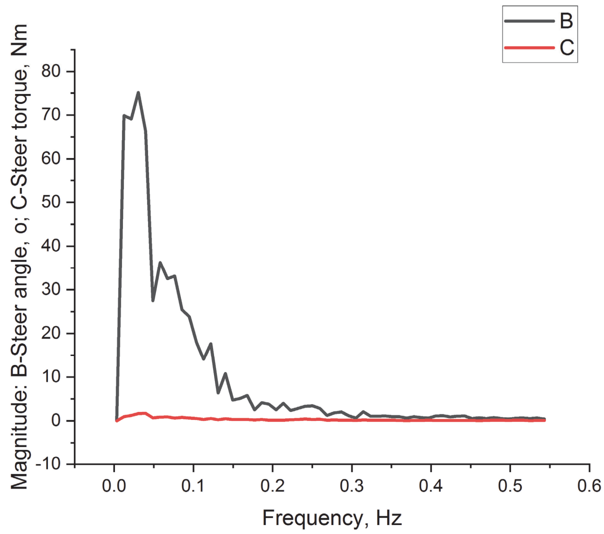

Figure 5: The spectra magnitudes of the angle and torque at the steering wheel during free driving on an asphalt road

For further analysis, Figure 6 will be considered, where 6a) shows the geometric model with important data, 6b) shows the physical model, and 6c) shows the static diagrams of the controlling lever.

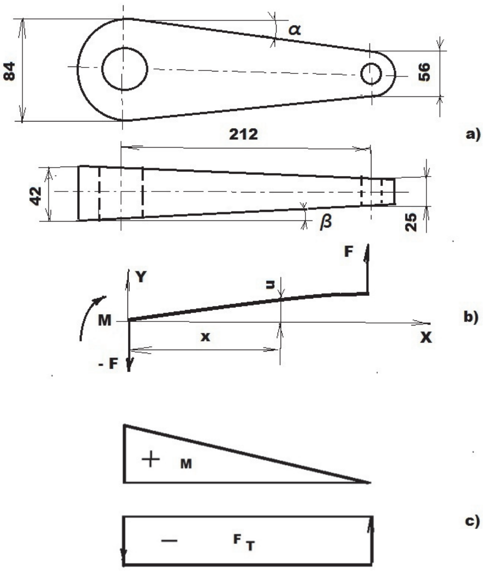

Figure 6: The geometric model of the controlling lever a), mechanical model b), and static diagrams c)

Taking into account the notes regarding the influence of the hydraulic part of the servo steering system to torque M and the controlling lever angle, we can express:

$$M(t) = M_d(t) i_u \eta_u$$
$$\Phi(t) = \frac{\Phi_d(t)}{i_u} \quad (1)$$

where: M_d - the torque at the steering wheel, Φ_d - the steering wheel angle, i_u – the steering system ratio, η_u- steering efficiency, and t - time.

The force F(t) is calculated based on the torque M(t):

$$F(t) = \frac{M(t)}{L} \quad (2)$$

where L - is the length of the controlling lever.

When defining a model to describe the transverse vibrations of the controlling lever, the following assumptions were made:

- the curvature of the controlling lever is small during transverse vibrations,
- the influence of longitudinal and other dynamic loads is neglected,
- the transverse cross-section of the controlling lever is rectangular and linearly decreases with displacement along its length, and
- the controlling lever material is homogeneous.

Considering that the transverse cross-section changes along the length of the controlling lever, based on Figure 6a) and the aforementioned, expressions for auxiliary variables a(x), b(x), cross-section area A(x), and moment of inertia Ix(x), were given:

$$tg(\alpha) = \frac{84-56}{2 \cdot 212}$$
$$tg(\beta) = \frac{42-25}{2 \cdot 212}$$
$$a(x) = 84 - 2\,x\,tg(\alpha) \quad (3)$$
$$b(x) = 42 - 2\,x\,tg(\beta)$$
$$A(x) = a(x)b(x)$$
$$I_x(x) = \frac{a(x)^3\, b(x)}{12}$$

Transverse vibrations of the elastic controlling lever are described by a partial differential equation. Since the evaluation of partial differential equations describing transverse vibrations of an elastic beam is extensively given in [19-21], it will not be done here, but only its final form will be presented. Based on the introduced assumptions, forced transverse vibrations of the controlling lever with a variable cross-section along its length are described by a partial differential equation [19]:

$$\frac{\partial^2 u(x,t)}{\partial t^2} + \frac{\partial^2}{\partial x^2}\left[\frac{EI_x(x)}{\rho A(x)}\frac{\partial^2 u(x,t)}{\partial x^2}\right] = F(x,t) \tag{4}$$

where: u=u(x,t) - transverse vibrations of the controlling lever, x - coordinate along the length of the controlling lever, F(x,t) - disturbance transversal force (excitation function) originating from the driver's action on the steering wheel and the random nature of micro-unevenness of the road, t - time, A(x) - cross-sectional area of the controlling lever, expression (3), Ix(x) - moment of inertia of the controlling lever's cross-section, expression (3), ρ - material density of the controlling lever, E - elastic modulus of the controlling lever.

As known [19-21], to find the general integral of the partial differential equation (4), it is necessary to know the boundary and initial conditions. In this specific case, it is assumed that the left end of the controlling lever is clamped and subjected to a torque M(t) (Figure 6c), while the right end is free (with zero torque). The transverse force is equal to -F(t) and acts along the entire length of the lever (see Figure 6c). At the initial time, both the deflection and inclination of controlling lever (vibration and velocity) are zero. Keeping that in mind, we can write the boundary and initial conditions:

$$\begin{aligned}
&M(x=0,t) = M(t) \rightarrow EI_x(x)\frac{\partial^2 u(x=0,t)}{\partial x^2} = M(t) \\
&u(x=0,t) = 0 \\
&u'(x=0,t) = 0 \\
&M(x=L,t) = 0 \\
&u(x,t=0) = 0 \\
&u'(x,t=0) = 0
\end{aligned} \tag{5}$$

It is obvious that the disturbance force originates from the transverse force on the lever (Figure 6c), so we have:

$$F(x,t) = -F(t) \tag{6}$$

where the force F(t) is defined by expression (2).

It should be noted that in the case of numerical solving of the partial differential equation (4), sometimes it is necessary to introduce additional boundary and initial conditions [11]. The integral of the partial differential equation (4), with boundary, initial conditions (5), and disturbance force (6), can only be sought in the case of harmonic excitation, so an attempt was made to solve it using the Wolfram Mathematica 13.2 software [11]. However, this software allows solving partial differential equations up to the second order, so the problem had to be solved numerically [22] using the finite difference method. Since this procedure is known from [22], there will be no further discussion about it here, and the problem was solved using a developed program that the author created in Pascal. A dynamic simulation was performed for a steel controlling lever with a variable cross-section, whose dimensions are given in Figure 6a), using the following data: $E=2.1\cdot10^5$, N/mm^2; $\rho= 8\cdot10^{-6}$, kg/mm^3; $n_x=424$; $h_x=0.5$, mm; $n_t=500$; $h_t=0.02$, s, $i_u=22,7$; $\eta_u=0.999$. Bearing in mind that the transverse vibrations of the controlling lever depend on two parameters, it is necessary to apply 3D graphics for their graphical representation. The integration of the partial differential equation (4) was done numerically, and the results are shown in Figure 7.

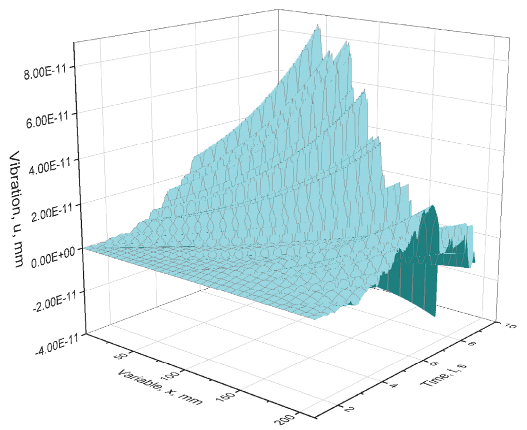

Figure 7: The transverse vibration of controlling lever

From Figure 7, it can be seen that the transverse vibrations of the controlling lever depend on the position along its length and time. When it comes to changes in the length of the controlling lever, it is obvious that the transverse vibrations are greatest around its center. As can be noticed, they have a stochastic character, which is agreed by [20].

Considering the aforementioned, it is necessary to apply 2 D Fourier transformation for frequency analysis. For its implementation, the author developed software in Pascal. However, considering the available commercial software on the market, it was deemed appropriate to use Origin 8.5 [23] for further analysis, as potential users will have easier access to this software.

Using the mentioned software, the magnitudes and phase angles of the two-parameter Fourier transform were calculated, and the results, for illustration purposes, are shown in Figures 8 and 9.

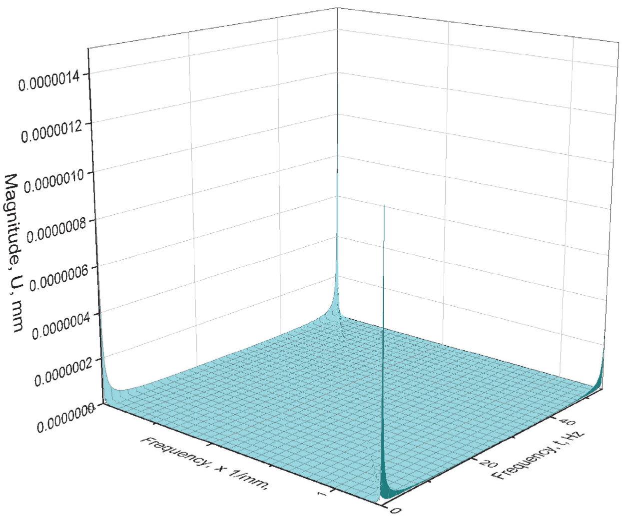

Figure 8: The spectra magnitude of transverse vibrations of the control lever

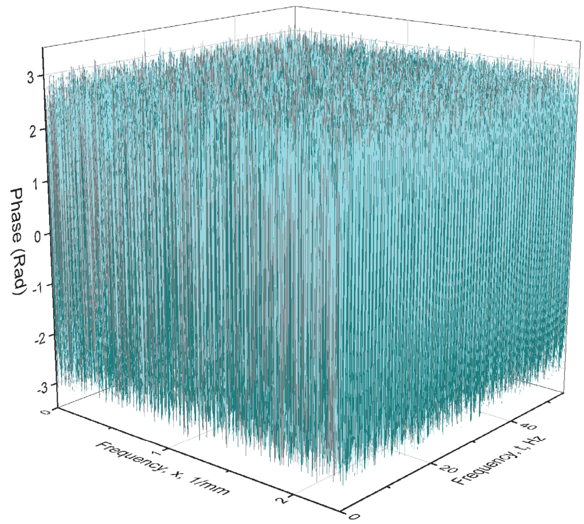

Figure 9: The spectra phase angles of transverse vibrations of the controlling lever

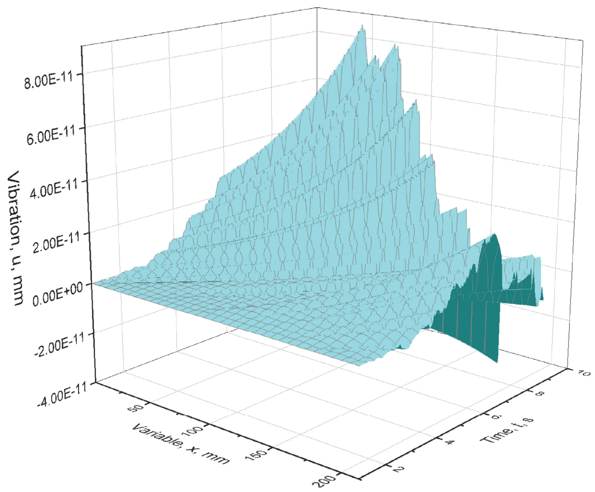

Figure 10: The transverse vibrations of the controlling lever obtained by inverse 2 D Fourier transform

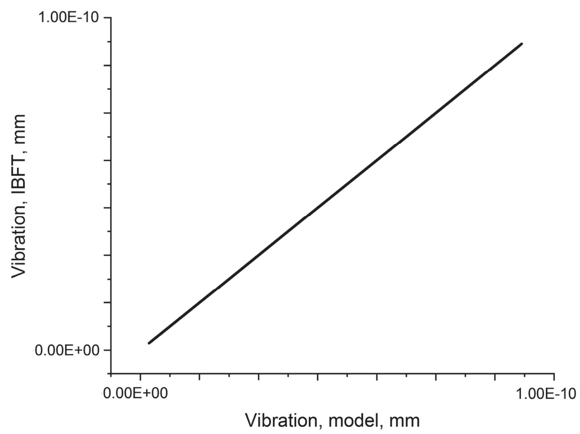

Figure 11: The comparison of transverse vibrations of the controlling lever obtained based on the model and inverse 2 D Fourier transform

3. DATA ANALYSIS

By analyzing the data from Figure 8, it can be determined that the vibrations (magnitude spectrum) vary along the length of the lever and depend on time. It is evident that the vibrations propagate in the form of random waves and that the magnitudes are higher near the ends of the controlling lever, as well as at lower and higher frequencies. This fact confirms the theoretical knowledge about the transverse vibrations of the lever [20,21]. The amplitude and frequency of harmonics depend on the design parameters of the controlling lever and the time excitation.

Based on Figure 9, it can be concluded that the phase angles of transverse vibrations of the controlling lever vary randomly along the length of the lever and depend on frequency (time), which confirms the random nature of the observed vibrations.

It should be noted that this study aims to investigate the possibilities of applying two-factor frequency analysis in the testing of controlling levers in the laboratory. Namely, it often requires generating data that correspond to those in exploitation, so it is justified to use the 2 D Inverse Fourier transform, which allows calculating experimental vibration values based on the magnitudes and phase angles of the spectrum. The Inverse Fourier transform can be implemented using the mentioned software Origin 8.5 [23]. By using this software, the Inverse 2 D Fourier transform was calculated, and the results are shown in Figure 10.

To determine the reliability of the data obtained based on the Inverse Fourier transform, numerical data from Figures 7 and 10 were compared. The result of the comparison is shown in Figure 11. Since their dependence is a straight line, it is evident that the results match, which is also by mathematical laws [11,12]. Of course, slight differences may occur at the micro level as a result of numerical operations [22].

Considering the high agreement between the data obtained by Inverse 2 D Fourier transform and experimental data (in this case, calculated based on the mathematical model (4)), the procedure have been created for its practical application in laboratory conditions [6].

During the performance of operational tests, it is necessary to record the parameters of transverse vibrations of the controlling lever (stresses, displacements, velocities, or accelerations of selected points) along its length, over a longer period. The sampling parameters of the recorded signals are described in detail in [12], so there will be no further discussion on that here.

It must be pointed out that in the case of two-parameter Fourier transforms, there are no explicit procedures for calculating errors in spectral analysis, as in the case of the 1 D Fourier transform [10]. Bearing that in mind, as well as the fact that this study aims to illustrate the possibilities of applying two-parameter frequency analysis in the investigation of transverse vibrations of the controlling lever, the analysis of statistical errors was not specifically performed.

Finally, it should be emphasized that the developed procedure has created conditions for analyzing the influence of the integration step on the accuracy and stability of the solution of the partial differential equation (4), the influence of design parameters on transverse vibrations of the controlling lever, the influence of disturbance forces, etc. However, considering that the results of dynamic simulation in this study served as a substitute for missing experimental results, it was assessed that a more detailed analysis is not necessary.

4. CONCLUSION

The conclusion of the research indicates that the two-parameter Fourier transform reliably enables the analysis of data on transverse vibrations of the controlling lever of commercial motor vehicles. Calculated magnitudes and phase angles of the spectra, using the inverse 2 D Fourier transform, allow for the generation of identical vibrations in the laboratory as well as in exploitation conditions.

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