MOBILITY & VEHICLE MECHANICS

https://doi.org/10.24874/mvm.2024.50.01.03
UDC: 629.3.021

CONTRIBUTION TO THE DEVELOPMENT OF A METHOD FOR OPTIMAL DIMENSIONING OF THE FRAME IN THE INITIAL DESIGN PHASE OF A HEAVY MOTOR VEHICLE

Miroslav Demić [1]*

Received in December 2023 *Accepted in February 2024*

RESEARCH ARTICLE

ABSTRACT: As it is known, the parameters of a heavy motor vehicle, including the frame, are not known in the initial design phase of the vehicle. Therefore, in this paper, an attempt is made to define the necessary dimensions based on the minimization of its lateral vibrations, while simultaneously minimizing the mass and maximizing the moment of inertia of the cross-section of the frame.

Attention is given to the choice of the objective function in the optimization process, with a special focus on the interrelation of sub-objectives. The author's previously developed frame model and „stochastic parametric optimization" based on the Hooke-Jeeves method were used for the research.

The calculated value of the unknown frame parameter is intended to serve for the final selection of the frame structure in the later design phase.

KEY WORDS: *Heavy motor vehicle, frame, lateral vibrations, mass, moment of inertia, troparametric Fourier transformation*

[1]*Miroslav Demić, Faculty of Engineering, University of Kragujevac, Sestre Janjic 6, Kragujevac, Serbia, demic @kg.ac.rs, : 0000-0001-5068-5560 (*Corresponding author)*

PRILOG RAZVOJU METODE ZA OPTIMALNO DIMENZIONISANJE OKVIRA U FAZI IZRADE IDEJNOG PROJEKTA TERETNOG MOTORNOG VOZILA

Miroslav Demić

REZIME: Kao što je poznato, parametri teretnog vozila, a samim tim i okvira, nisu poznati u početnoj fazi projektovanja, pa je, u ovom radu učinjen pokušaj definisanja potrebnh dimenzija, na bazi minimizcije njegovih poprečnjih vibracija, uz istovremenu minimizaciju mase i maksimizaciju momenta inercije poprečnog preseka okvira.

Pažnja je posvećena izboru funkcije cilja u procesu optimizacije, sa posebnim osvrtom na međusobni odnos podciljeva. Za istraživanje je korišćen ranije razvijeni autorov model okvira i "stohastička parametarska optimizacija" zasnovana na metodi Hooke-Jeeves-a.

Izračunata veličina nepoznatog parametra okvira treba da posluži za konačan izbor structure okvira u kasnijoj fazi projektovanja.

KLJUČNE REČI: *Teretno motorno vozilo, okvir, poprečne vibracije, masa, moment inercije, troparametarska Furijeova transformacija*

CONTRIBUTION TO THE DEVELOPMENT OF A METHOD FOR OPTIMAL DIMENSIONING OF THE FRAME IN THE INITIAL DESIGN PHASE OF A HEAVY MOTOR VEHICLE

Miroslav Demić

INTRODUCTION

The initial design phase of a heavy motor vehicle, defined by the design task, is further developed in subsequent design phases, where creative and intuitive settings that played a significant role in the development of the design task give way to logical and objective factors, calculations, measurements, shaping, evaluations of production and technological capabilities, etc. [1,2].

The assumption is that the design task defines that a heavy motor vehicle with a total mass of 11000, kg and a payload capacity of 4,000, kg needs to be designed for the market, with dimensions (length * width * height, mm): 6400*2500*3600, with a short cab. The engine is positioned at the front, and it has all-wheel drive.

The frame of the newly designed vehicle must withstand rigorous exploitation conditions, and it will be dimensioned using optimization methods based on minimal transverse vibrations and mass, and maximum moment of inertia of the transverse section.

The previously developed frame model and the “stochastic parametric optimization” based on the Hooke-Jeeves method will be used. Special attention will be given to the choice of the objective function and the analysis of the influence of sub-objectives on the optimizing parameter.

It is pointed out that optimization can be done:

•theoretically, using mathematical models and dynamic simulation,

•experimentally, and

•in a combined way.

It should be noted that experimental research is expensive and often not applicable in the initial phase of vehicle design. Therefore, the procedure of theoretical optimization is accepted here, which required modeling of the frame. It is noted that the author has previously developed a frame model, and it will be presented here with slight abbreviations that will not compromise the understanding of the method of the optimal selection process of its unknown parameter.

1. METHOD

As already mentioned, this paper aims to explore the possibility of applying optimal dimensioning of the frame in the initial design phase of a vehicle, using optimization methods. It was deemed appropriate that the objective function should enable the minimization of transverse frame vibrations and its mass, while maximizing the moment of inertia of the transverse section.

For further considerations, a ladder structure of the frame was adopted, as in [3], along with its length and width. Now, based on the required torsional stiffness of the frame, it is necessary to define the dimensions of the longitudinal and transverse profiles using some of the calculation methods, most commonly finite element methods [4]. It should be noted that in this design phase, the bending and torsional stiffness, as well as precise external loads, are not known, so the application of the mentioned method is not possible with satisfactory reliability.

Considering that in this design phase, a large number of frame parameters are unknown and some of them must be obtained through the study of simpler models, it was deemed appropriate to idealize and observe the frame as a homogeneous plate of adopted length and width, with unknown thickness [3], Figure 1a). The plate undergoes transverse vibrations under the influence of disturbing forces at the connection points of aggregates, systems, and the frame...

For motor vehicles [2], there are statistical data on the percentage participation of aggregates and systems in the vehicle's own mass. Based on the design requirement, a frame length of 6100, mm and a width of 800, mm were adopted, so based on the data from [2], the thickness of the equivalent plate (idealized steel frame) was calculated to be 23, mm [3].

In order to analyze the transverse vibrations of the frame model, it was necessary to define dynamic excitations. Considering that not all excitations are known in this project phase (uneven engine operation, road irregularities, tire non-uniformity, etc.), it was deemed appropriate to analyze vibrations under conditions of short-term intensive braking [3,5]. Based on experience, an impulse decceleration shape was chosen with the presence of random changes, as illustrated in Figure 1b) [3].

For further analysis, it was adopted that the engine is supported by the frame at four mounts, the cab also at four points [1,3]. The cargo box is supported at eight mounts, but for the sake of problem simplification, it was assumed that it is attached to the frame at four points. As for the springs, each of them is connected at two points, but for the same reasons as with the cargo box, it was assumed that they are connected at one point each [1,3].

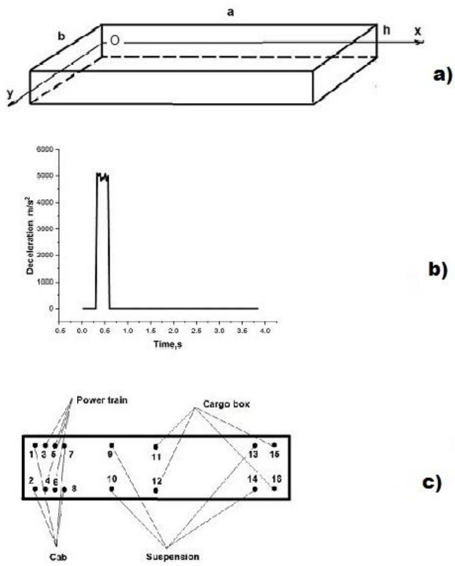

Figure 1. Idealized model of the vehicle frame 1a), vehicle deceleration during short-term braking 1b), and the position of the connection points of the engine, cabin, and cargo box 1c).

Illustration of the conection points is shown in Figure 1c), and the empirical coordinates of the points are given in Table 1.

Table 1. Coordinates of connection points

	Coordinate X, mm	Coordinate Y, mm
1	365	40
2	365	760
3	560	40
4	560	760
5	1375	40
6	1375	760
7	1496	40
8	1496	760
9	1600	40
10	1600	760
11	3400	40
12	3400	760
13	5015	40
14	5015	760
15	5050	40
16	5050	760

In order to determine the excitation forces, it was necessary to calculate the characteristic masses of the aggregates. This was done using statistical data on the percentage of aggregate masses in the vehicle mass, as well as based on recommendations for the size of the dead mass [2,3].

In addition to the mass of the aggregates and systems, it was necessary to calculate the distance between the supports along the length of the vehicle frame (which was done using data from Table 1) and define the height of the aggregate's center of gravity relative to the upper edge of the frame [1,3]. Approximate data is given in Table 2.

Table 2. Longitudinal distance of connection points, height of center of gravity relative to frame, and mass of aggregates and systems

	Connection point distance, mm	Center of gravity height, mm	Mass, kg
Power train	1010	200	962
Cab	1040	800	578
Cargo box	1915	850	890
Suspended mass	3600	1200	8070

The inertial force due to braking is given by the expression [2,3,5]:

$$F_i = m_i a \,, \tag{1}$$

where:

- a - decceleration defined by Figure 1b), and
- m_i - corresponding mass (power train, cab, cargo box, suspended mass).

We will assume that the center of mass of the aggregate and system is located in the middle of the longitudinal distance, so in that case, the static load (Fst) is equal on all mounts to a quarter of the gravitational force. The static force increases on the front mounts and decreases on the rear during vehicle braking. The magnitude of the force change due to braking is given by the expression [3,5]:

$$\Delta Z = \frac{F_i h_{ti}}{4L_i}, \quad (2)$$

where:

- F_i - inertial force due to braking, defined by equation (1),
- h_{ti} - height of the center of mass of the corresponding mass from Table 2,
- L_i - longitudinal distance between mounts from Table 2,
- + refers to the front, and - to the rear mounts.

Based on equations (1 and 2), the force on each mount is calculated [3]:

$$F = F_{st} \pm \Delta Z, \quad (3)$$

Transverse vibrations of the elastic plate are described by a partial differential equation.The evalution of the partial differential equation that describes the transverse vibrations of the elastic plate is detailed in [3,6,7], it will not be done here, but only its final form will be presented [6]:

$$D\left(\frac{\partial^4 u}{\partial x^4} + 2\frac{\partial^4 u}{\partial x^2 \partial y^2} + \frac{\partial^4 u}{\partial y^4}\right) + \rho h \frac{\partial^2 u}{\partial t^4} = f(x, \mathrm{y}, \mathrm{t}), \quad (4)$$

where:

- u = u(x, y, t) - transverse vibrations of the frame,
- x - coordinate along the length of the frame,
- y - coordinate along the width of the frame,
- F(x, y, t) - disturbance transverse force (excitation function),
- t - time.

The value of D is given by the expression [6]:

$$D = \frac{Eh^3}{12\left(1-\upsilon^2\right)}, \quad (5)$$

where:

- E - Young's modulus,
- υ -Poisson's ratio, and
- h - plate thickness.

As you know [6,7,8], in order to find the general integral of a partial differential equation (4), it is necessary to know the boundary and initial conditions.

In this specific case, all edges are free (torques and shear forces are equal to zero), and vibrations and their velocities are equal to zero at the initial time [3].

Mathematically, these conditions are defined by equations [3]:

$$M_x = -D\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\upsilon \partial y^2}\right) = 0 : x = 0$$

$$V_x = Q_y - D\left[\frac{\partial^3 u}{\partial x^3} + (2-\upsilon)\frac{\partial^3 u}{\partial x \partial y^2}\right] : x = 0; Q_y = 0$$

$$M_x = -D\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\upsilon \partial y^2}\right) = 0 : x = a$$

$$V_x = Q_y - D\left[\frac{\partial^3 u}{\partial x^3} + (2-\upsilon)\frac{\partial^3 u}{\partial x \partial y^2}\right] : x = a; Q_y = 0$$

$$M_y = -D\left(\frac{\partial^2 u}{\partial y^2} + \upsilon\frac{\partial^2 u}{\partial x^2}\right) = 0 : y = 0 \qquad (6)$$

$$V_y = Q_x - D\left[\frac{\partial^3 u}{\partial x^2 \partial y} + (2-\upsilon)\frac{\partial^3 u}{\partial y^3}\right] : y = 0; Q_x = 0$$

$$M_y = -D\left(\frac{\partial^2 u}{\partial y^2} + \upsilon\frac{\partial^2 u}{\partial x^2}\right) = 0 : y = b$$

$$V_y = Q_x - D\left[\frac{\partial^3 u}{\partial x^2 \partial y} + (2-\upsilon)\frac{\partial^3 u}{\partial y^3}\right] : y = b; Q_x = 0$$

$$u(x,y,0) = 0 ,$$

$$u`(x,y,0) = 0$$

The disturbance force represents the sum of dynamic forces at the mounts [3]:

$$f(x,\mathrm{y},\mathrm{t}) = \sum_{i=1}^{16} F_i(t), \qquad (7)$$

where the force Fi(t) is defined by expression (3) calculated at each mount.

The integral of the partial differential equation (4), with boundary, initial conditions (6), and disturbance force (7), can only be sought in the case of harmonic excitation (and not without difficulties), so an attempt was made to solve it using the Wolfram Mathematica 13.2 software [8]. However, this software allows solving partial differential equations up to the second order, so the problem had to be numerically solved [9] using the finite difference method.

The author developed software for solving the partial differential equation (4) using the finite difference method, with boundary, initial conditions (6), and disturbance force (7), in Pascal. It should be noted that in the case of numerical solving of partial differential equations, sometimes it is necessary to introduce additional boundary and initial conditions [8]..

2. VEHICLE FRAME DIMENSIONING

In the following text, there will be more words about optimal frame dimensioning, using optimization methods. It should be noted that various procedures are used in practice for this purpose, and here the method of "stochastic parametric optimization" will be applied. As known, the method of "stochastic parametric optimization" is used in the optimization of oscillatory parameters of motor vehicles and is based on nonlinear programming methods. Since there are constraints on design parameters in the optimization process, the problem is solved by introducing "external" or "internal" penalty functions [10,11].

In this specific case, the method of "stochastic parametric optimization" based on the Hooke-Jeeves method and "external" penalty functions was used for the selection of plate thickness (length and width are adopted). Considering that this optimization method is described in detail [10,11], it will not be done here. For illustration purposes, its block diagram will be shown in Figure 2, and the software is implemented in Pascal.

It was deemed appropriate to make the optimal choice of plate thickness (idealized frame) based on the conditions of minimal frame vibrations as an elastic system, its minimal mass, and maximum moment of inertia of the cross-section.

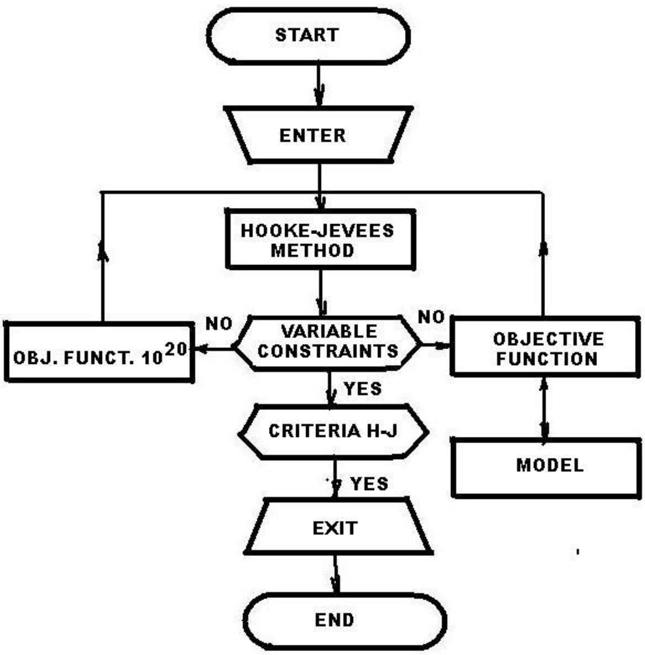

Figure 7. Block diagram of the used optimization method

Bearing this in mind, the objective function is used:

$$Z = r_1 u_{RMS} + r_2 m - r_3 I_x , \quad (8)$$

where:

- r_1, r_2, and r_3 - weighting factors that define the influence rank of sub-objectives in the objective function and allow the conversion of sizes defining sub-objectives into the same units (more words about their choice will be mentioned later),

- u_{RMS}- RMS values of frame vibrations obtained by solving the partial differential equation (4), and
- I_x - moment of inertia of the cross-section of the idealized frame.

The RMS of transverse frame vibrations is calculated using the expression:

$$u_{RMS}^2 = \frac{1}{n_x n_y n_t} \sum_{i=1}^{n_x} \sum_{j=1}^{n_y} \sum_{k=1}^{n_z} u\left(i, j, k\right)^2 , \quad (9)$$

where:

- $u(i,j,k)$ - transverse vibrations of the idealized frame,
- n_x - number of points along the x-axis,
- n_y - number of points along the y-axis, and
- n_t - number of points along the t-axis.

The mass of the frame is calculated based on the dimensions of the cross-section and length, using the expression:

$$m = \rho bhl , \quad (10)$$

where:

- b - width of the frame,
- h - thickness of the idealized frame,
- l - length of the frame, and
- ρ - density of the material.

The moment of inertia of the cross-section of the frame is calculated using the expression [12]:

$$I_x = \frac{bh^3}{12} , \quad (11)$$

During the process of optimal selection of the thickness of the idealized frame, its boundary values are defined:

$$5 \le h \le 23 .$$

By introducing the optimizing parameter x[i], i=1(u-maximum, l-minimum), instead of h, with the corresponding adopted boundary values $x_u[i]$, $x_l[i]$, i=1, the objective function depends on one optimizing parameter, and it has multiple local minima and only one global minimum.

Considering that, in practice, the problem of finding the global minimum is solved by starting the optimization process with multiple initial values of the optimizing parameters [10,11], it was deemed appropriate to start with three initial values of these parameters, namely:

$$x = 0{,}5 x_u \left[1\right]$$

$$x = 0{,}8 x_u \left[1\right]$$

$$x = 1{,}2 x_l \left[1\right]$$

A dynamic simulation was performed for a steel frame structure with the following data: E=2.1*105, N/mm^2; ρ=8*10^{-6}, kg/mm^3; ν=0.3; n_x=128; n_y=128; n_t=128; h_x=47.65, mm; h_y=6.25, mm; h_t=0.08, s. The values of the number of points and discretization steps used during dynamic simulation ensured the reliability of the results for the parameter x: 0.00016 to 1.05, 1/mm, y: 0.0012 to 0.08, and t: 0.19 to 12.5 Hz [13].

Table 3. The data of the optimal selection of the thickness of the plate (idealized frame)

Initial values	Optimal parameter, h, mm	Objective function, Z, -	No of iter., N,-
0.5xu[i]	2.300000000000000E+001	-8.102355310777008E+005	1175
0.8(xu[i]) *	2.299999999999998E+001	-8.102355310776414E+005	863
1.2(xl[i]) *	2.299999999999988E+001	-8.102355310777E+005	567
0.5xu[i]**	5.000000000000192E+000	1.951759259703150E+002	484
0.8(xu[i]) **	5.000000209339827E+000	1.951759353878179E+002	746
1.2(xl[i]) **	5.000000000000021E+000	1.951759216328079E+002	273
0.5xu[i] ***	2.291000000000019E+001	1.576522170159383E-005	441
0.8(xu[i]) ***	1.853112842559817E+001	2.989550736588539E-005	206
1.2(xl[i]) ***	2.292999999999970E+001	1.572400545690352E-005	441
0.5xu[i] ****	2.298000000000023E+001	5.566501146807580E+005	443
0.8(xu[i]) ****	2.299773124575687E+001	5.543849553365553E+005	398
1.2(xl[i]) ****	2.299878906249978E+001	5.542498338388450E+005	436

Legend: *r_1=1; r_2=1; r_3=1; ** r_1=1; r_2=1; r_3=0; *** r_1=1; r_2=0; r_3=0; *** *r_1=3*1010; r_2=103; r_3=1;

The optimization was performed on a Pentium 4 computer (Intel 2.4 GHz, 9 GB RAM), and the iterative process was automatically terminated when the difference between two adjacent values of the objective function was 10-15. The optimization time for each combination was about 25, minutes, and the calculated parameters are shown in Table 3.

3. DATA ANALYSIS

In order to select the ranks of influence on the objective function (Table 3), the optimization process was implemented with four groups. The first combination (*) introduced equal influence of sub-goals. However, the sub-goals did not have the same numerical values, so the moment of inertia had the greatest influence on the goal function, while the transverse vibrations had the smallest influence, which resulted in a negative sign in front of the numerical value of the objective function. Taking this into account, it was deemed appropriate to eliminate the influence of the moment of inertia from the objective function (combination **). Now, smaller values of the goal function have been obtained.

To determine how vibrations affect the goal function, research was conducted for combination (***). In this case, larger differences in the optimal thickness values were obtained, as well as differences in the minimum of the objective function.

Based on the aforementioned, the influence of the rank of influence and the type of sub-goal used on the process of optimal selection of plate thickness in the initial design phase of the frame is obvious and logical. Therefore, it was decided to give approximately equal influence to each individual sub-goal in expression (8). In this regard, a preliminary analysis was performed, and certain ranks of influence were marked with (****). Their application led to a change in the values of the goal function.

It can be concluded that the optimal plate thickness values are approximately equal, except in the case of (***). Namely, the optimal plate thickness is very close to its upper limit value. This can be explained by the fact that the used plate thickness interval is relatively small, and a larger thickness leads to a higher moment of inertia.

Therefore, it was considered appropriate to adopt a plate thickness of 22.99, mm as the optimal size (combination ****). It should be used to define the structure and dimensions of the ladder frame in the subsequent stages of vehicle design.

Having in mind what was previously said, it is appropriate in future research to start the optimization process with only one group of influence ranks, and that is in the middle of the interval.

In this case, the RMS vibration was $1.69381*10^{-5}$ mm, the mass was 897.41, kg, and the moment of inertia of cross-section was 8.10075*105, $kgmm^4$.

It was deemed appropriate to examine the contribution of the calculated mass to the total mass of the newly designed vehicle (11000 kg). The obtained ratio was 8.15%, which can be considered acceptable for further project development [1,2,3].

It was deemed appropriate to perform an analysis of the transverse vibrations of the idealized frame for the optimal thickness. Considering that they depend on three parameters (*x*, *y*, *t*), which require the use of 4D graphics that do not exist in commercial form, it was deemed useful to use 3D graphics and only display vibrations for the centre of gravity planes, as illustrated in Figures 3 and 4.

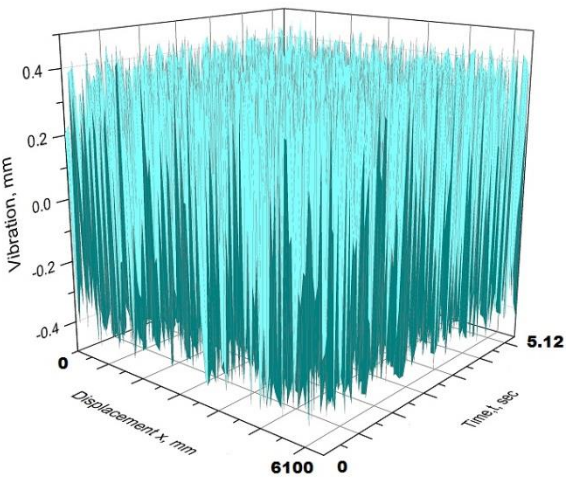

Figure 3. Transverse vibrations of the frame in the longitudinal center of gravity plane

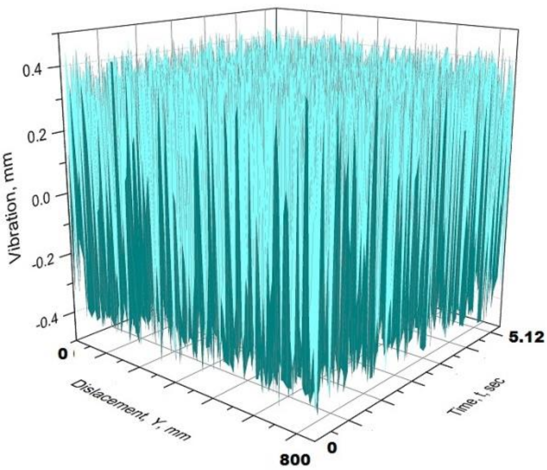

Figure 4. Transverse vibrations of the frame in the lateral center of gravity plane

From the analysis of the data from Figures 3 and 4, it can be determined that they vary stochastically along the length and width of the frame. The random nature of the vibrations can be explained by the random nature of the excitation force used, and the change in vibrations across the surface of the plate is in accordance with [6,7,14].

It was deemed appropriate to perform a frequency analysis of the transverse vibrations using 3D Fourier transform, using a program developed in Pascal. For the same reasons as in the previous case, the results are only shown for the center of gravitty planes, in Figures 5-8.

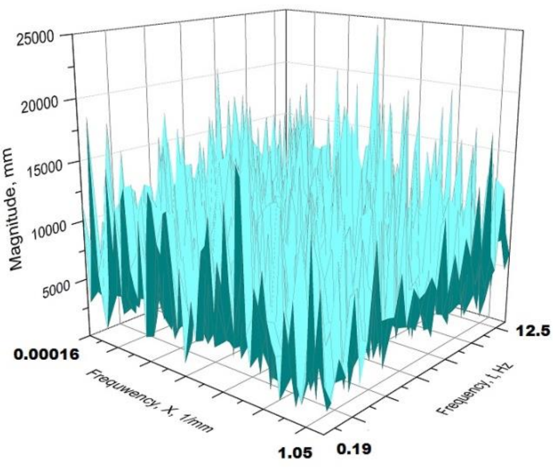

Figure 5. Spectrum magnitude of frame vibrations in the longitudinal center of gravity plane

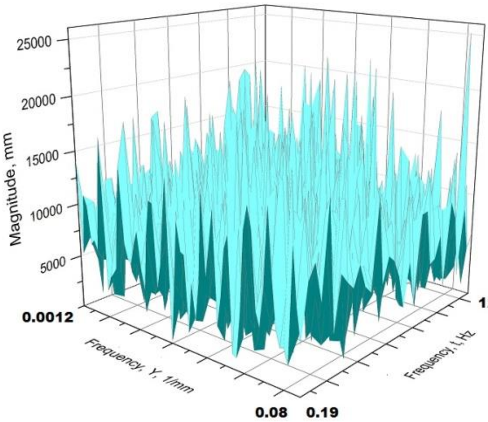

Figure 6. Spectrum magnitude of the frame in the lateral center of gravity plane

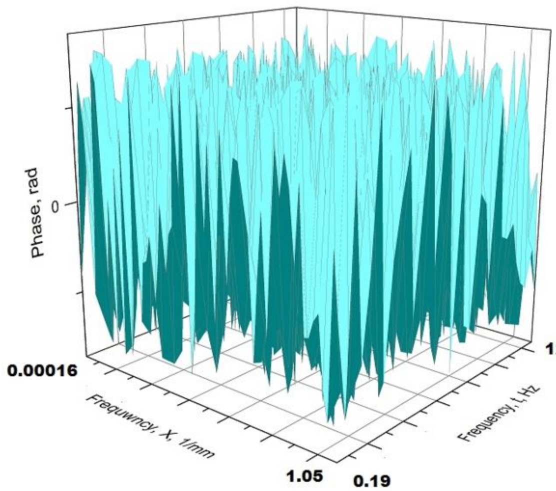

Figure 7. Phase angles of the vibration spectrum of the frame in the longitudinal center of gravity plane

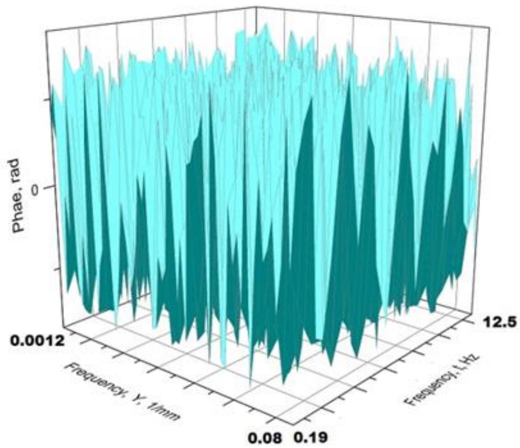

Figure 8. Phase angles of the vibration spectrum of the frame in the lateral center of gravity plane

Note that the calculated plate thickness is approximate and may be subject to change during the structural verification using finite element method and experimentation, which will not be discussed here [4].

It should also be noted that there are no explicit procedures for calculating errors in spectral analysis for 3D Fourier transform, as in the case of 1D Fourier transform [10]. Considering this, as well as the fact that the goal of this study is to illustrate the potential application of 3D Fourier transform in the analysis of transverse vibrations of the vehicle frame, statistical errors were not calculated...

4. CONCLUSIONS

In conclusion, the developed procedure, based on the analysis of transverse vibrations, mass, and moment of inertia cross-section of the vehicle frame, allows for the definition of its dimensions in the initial phase of vehicle design.

In further development of the project, based on these defined parameters, more detailed calculations can be performed, potentially using the finite element method.

The conducted analyses have shown that the use of 3D Fourier transform is desirable for the analysis of transverse vibrations of the vehicle frame..

REFERENCES

[1] Demić, M.: Basics of designing heavy motor vehicles, Faculty of Mechanical Engineering in Kragujevac, 1994 (in Serbian).

[2] Simić, D.: Motor vehicles, Scientific book, Belgrade, 1988 (in Serbian).

[3] Demić, M. Development of a method for defining the torsional stiffness of the frame in the initial phase of designing a heavy motor vehicle, Mobility and Vehicle Mechanics, Vol 49, No 3, pp. 15-29, https://doi.org/10.24874/mvm.2023.49.03.02 (2023).

[4] Bathe, K.J. : Finite Element Procedures in Engineering Analyses, Prentice Hall, Englewood Cliffs, N.J, 1982.

[5] Gillespie, T.: Fundamementals of Vehicle Dynamics, SAE,1992.

[6] Singeresy, S.R. : Vibration of continuous systems, John Wiley and Sons, Inc. New Jersey, 2007.

[7] Sobolev, S. L.: Partial differential equations of Mathematical Physics, Pergamon Student Edition, Elsevier, 2016.

[8] Wolfram Mathematica: https://Reference.Wolfram.com/language/tutorial/NDSolverPDE.html

[9] Stanton, R. G.L: Numerical methods for Science and Engineering, Prentice Hall, Englewood Cliffs, N.J,1961.

[10] Bunday P.: Basic optimization methods. Spottiswoode Ballantyne, Colchester and London, 1984.

[11] Demić M.: Optimization of vehicles elastodamping elements Characteristics from the aspect of ride comfort, Vehicle System Dynamics, Vol. 23 , No 5, pp 351-370, 1994.

[12] Timoshenko, S.: Strength of materials, Part II, Advanced Theory and Problems, Paolo Atto, California, 1941.

[13] Bendat, J.S., Piersol, A.G.: Random Data-Analysis and measurement procedures, John Wiley and Sons, 2000

[14] Demić, M.: Application of multi-parameter frequency analysis in experimental identification of vibration parameters in motor vehicles, Mobility and Vehicle Mechanics, Vol. 49, No. 1, pp 25-37, https://doi.org/10.24874/mvm.2023.49.01.03 (2023).