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INVESTIGATION OF TORSIONAL VIBRATIONS OF THE STEERING SHAFT FROM THE ASPECT OF MINIMAL DRIVER-HAND FATIGUE IN HEAVY MOTOR VEHICLES

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RESEARCH ARTICLE

ABSTRACT: Local hand vibrations can cause occupational diseases in drivers of heavy motor vehicles, such as white finger syndrome.

In this study, a method has been developed to check the diameter of the steering shaft from the aspect of hand fatigue due to the action of tangential vibrations of the steering wheel, as tangential accelerations occur in it due to torsional vibrations, as an elastic system. By applying the method of “stochastic parametric optimization”, the diameter of the steering shaft was calculated from the aspect of minimizing tangential accelerations. Based on this value, the tangential vibrations of the steering wheel were calculated and compared with the ISO 5349/DIS criteria. Since the obtained values of analyzed accelerations are lower than the mentioned criteria, the calculated diameter of the steering shaft can be used for further analysis. More precisely, it should be compared with the one on the derived steering mechanism and, as authoritative, adopt the larger one.

KEY WORDS: *Steering shaft, torsional vibrations, tangential accelerations of the steering wheel, optimization*

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PRILOG ISTRAŽIVANJU TORZIONIH VIBRACIJA OSOVINE UPRAVLJAČA SA ASPEKTA MINIMALNOG ZAMORA RUKU VOZAČA TERETNIH MOTORNIH VOZILA

REZIME Lokalne vibracije ruku mogu izazvati profesionalna oboljenja kod vozača teretnih motornih vozila, poput sindroma belih prstiju.U ovom radu je razvijena metoda za proveru prečnika osovine upravljača sa aspekta zamora ruku usled dejstva tangencijalnih vibracija točka upravljača, jer se kod nje usled torzionih vibracija, kao elstičnog sistema, javljaju tangencijalna ubrzanja. Primenom metode stohastičke parametarske optimizacije izračunat je prečnik osovine upravljača sa aspekta minimizacije tangencijalnih ubrzanja. Na osnovu te vrednosti izračunate su tangencijalne vibracije točka upravljača i upoređene sa kriterijumima ISO 5349. Kako su dobijene vrednosti analiziranih ubrzanja niže od pomentih kriterijuma, izračunati prečnik osovine upravljača se može koristiti tokom daljih analiza. Preciznije rečeno, isti treba uporediti sa onim na izvedenom upravljačkom mehanizmu i, kao merodavan, usvojiti veći.

KLJUČNE REČI: *Osovina upravljača, torzione vibracije, tangencijalna ubrzanja točka upravljača, optimizacija*

INVESTIGATION OF TORSIONAL VIBRATIONS OF THE STEERING SHAFT FROM THE ASPECT OF MINIMAL DRIVER-HAND FATIGUE IN HEAVY MOTOR VEHICLES

Miroslav Demić, Mikhail P. Malinovsky

INTRODUCTION

The heavy motor vehicle defined by the project task is elaborated in further design phases [1]. The conceptual design represents the first concretization of the project task, and its main goal is to define the parameters of the driver's ergo-sphere and the external dimensions, mass, and performance of the heavy vehicle, as well as its stylistic indicators necessary for further work on the project [1].

In this paper, there will be more words about the aspect of minimal driver hand fatigue during free driving of a heavy motor vehicle, and as a basis, the author uses the principles set in [2], for the automatic design of elastic systems in cargo motor vehicles.

The assumption is that the project task defines a cargo motor vehicle with a total mass of 11000, kg, with a useful load capacity of 4000, kg, dimensions (length * width * height, mm): 6400*2500*3600, with a short cab. The engine position is front, and the drive is on the rear wheels.

The steering axle (column, shaft-hereinafter referred to as the shaft), in addition to other elements of the steering system, must also provide minimal driver-hand fatigue due to local vibrations on the steering wheel.

For this purpose, a simplified axle model will be used, as well as a stochastic parametric optimization procedure based on the Hooke-Jeeves method.

It is pointed out that the choice of the steering shaft diameter can be made [1]:

- theoretically, using mathematical models and dynamic simulations,
- experimentally, and
- combined.

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 was accepted here, which required modeling of the steering shaft.

Considering the goal of this paper, it was considered appropriate to compare the tangential accelerations of the steering wheel, which are the result of torsional vibrations of the shaft, with international standards ISO 5349/DIS, which are, for illustration, shown in Figure 1 [3]. They define permissible hand vibration loads for exposure to vibrations lasting 4-8 hours. From Figure 1, it can be seen that the approximate permissible hand accelerations are constant up to 16 Hz, and with further increase in frequency, they linearly increase.

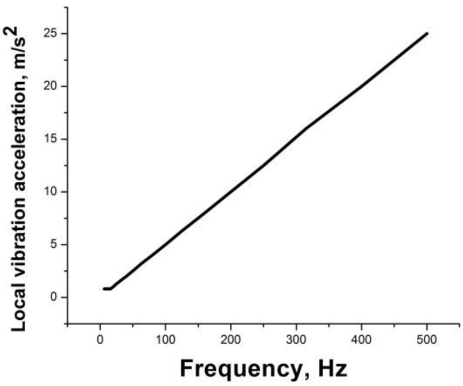

Figure 1. ISO 5349/DIS (1979) Local hand vibrations for exposure to local vibrations lasting 4-8 hours

In practice, there are several characteristic concepts of servo steering systems, which will not be further discussed here, as they are described in detail in [4].

Considering that the goal of this paper is the development of a method for analyzing torsional vibrations of the shaft, from the aspect of minimizing tangential accelerations of the steering wheel, it was considered appropriate to illustrate it on the concept shown in Figure 2a).

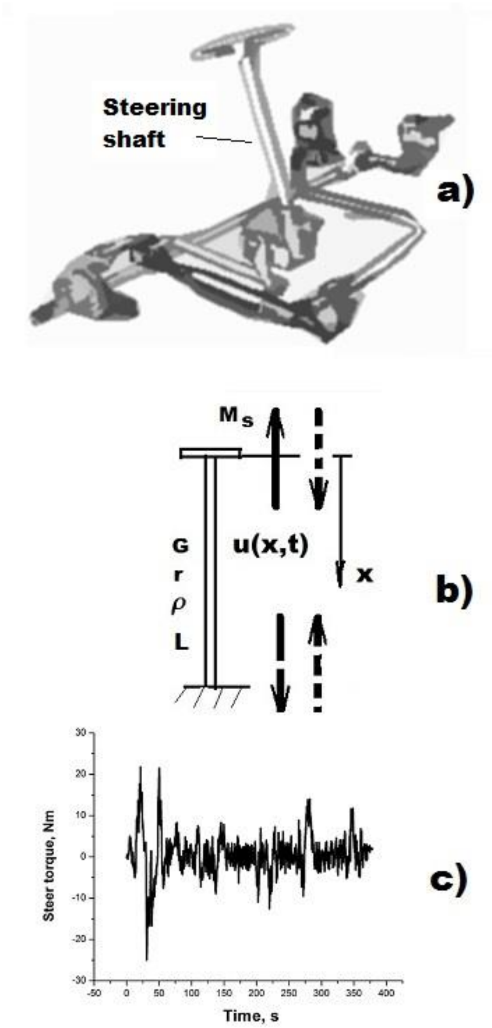

Figure 2. Observed concept of servo steering a), idealized model of the steering shaft b) and experimental torque on the steering wheel c)

It should be noted that the delivered servo steering system has a defined shaft diameter, but it was considered appropriate to calculate it from the conditions of minimal tangential accelerations and compare it with the existing one. The larger one should be adopted as the relevant one.

1 METHOD

As already mentioned, this paper aims to explore the torsional vibrations of the steering shaft from the aspect of minimal driver fatigue due to tangential accelerations of the steering wheel. In doing so, the influences of other oscillatory loads of the vehicle as a whole, or its systems, are neglected.

It should be noted that in the conceptual design phase of the steering system of a commercial motor vehicle, its parameters are usually defined by calculation from the conditions of the moment of rotation in place, or movement at low speeds [4]. Based on this, the selection of a power steering system is made, which already has a defined shaft diameter.

For the research of torsional vibrations of the shaft, it was considered appropriate to idealize it and observe it as a homogeneous rod of adopted length (from ergonomic conditions [1], and unknown diameter [4], (Figure 2b). The shaft undergoes torsional vibrations under the action of a disturbing torque. The torque at the steering wheel point registered in operational conditions [2] was used in the work. For illustration, a depiction of its temporal change during the free ride of an analog vehicle on an asphalt road is shown in Figure 2c.

By applying the method of stochastic parametric optimization, the diameter of the steering shaft was calculated from the aspect of minimizing tangential accelerations. Based on this value, the tangential vibrations of the steering wheel were calculated and compared with ISO 5349/DIS criteria.

2 STEERING SHAFT MODEL

When defining a model that describes the torsional vibrations of an elastic steering shaft, the following assumptions were introduced:

- the shaft is homogeneous and of constant diameter,
- the influence of clearance in the joints of the power steering system is neglected,
- the influence of tangential elasticity of the steering wheel and other elements in the steering system is neglected, and
- the existence of other vibrational excitations originating from the vehicle itself is neglected.

Since the partial differential equation describing the torsional vibrations of an elastic rod, which also applies to the steering shaft, is detailed in [5,6], it will not be done here, but its final form will be shown. Forced torsional vibrations of the steering shaft [5,6] are described by a partial differential equation:

$$\rho \frac{\partial^2 u}{\partial t^2} = G \frac{\partial^2 u}{\partial x^2} + f\left(x,t\right), \tag{1}$$

where: $u(x, t)$ - torsional vibrations of the shaft, x - coordinate along the length of the shaft, $f(x, t)$ - the forced torque transmitted from the steering wheel to the shaft, t - time, G - shear modulus, and ρ - material density of the shaft.

The excitation function from the moment at the steering wheel is given by the expression:

$$f\left(x,t\right) = m_s\left(t\right), \tag{2}$$

where: $m_s(t)$ - the experimentally registered torque at the steering wheel.

As known [5-7], to determine the general integral of the partial differential equation (1), it is necessary to know the boundary and initial conditions.

The torsional torque due to shaft vibrations can be expressed [5,8]:

$$M = GIo \frac{\partial u\left(x,t\right)}{\partial x}, \tag{3}$$

where:

- I_0 - polar moment of inertia of the cross-section of the shaft, given by the expression [8]: $I_o = \frac{\pi r^4}{2}$
- r - shaft radius.

Defining boundary conditions in the analysis of vibrations of elastic bodies, in general, represents an idealization of the real state. In this specific case, it is assumed that one end is quasi-free (subject to a steering wheel torque), and the other end is clamped. Additionally, it is assumed that torsional vibrations and their velocities are initially equal to zero, i.e.:

$$\begin{aligned} & GI_o \frac{\partial u(0,t)}{\partial x} = m_s(0,t) \\ & u(0,L) = 0 \\ & u(x,L) = 0 \\ & u`(x,0) = 0 \end{aligned} \quad , \tag{4}$$

The partial differential equation (1), with boundary and initial conditions (4), can only be solved in closed form in the case of harmonic excitation [5-7]. Attempts to solve it using Wolfram Mathematica 13.2 [7] encountered difficulties in presenting numerical data, so it was decided to solve the problem numerically using the finite difference method [9].

The author developed a program in Pascal to numerically solve the partial differential equation (1) with the excitation function (2) and boundary and initial conditions (4). It is noted that in the case of numerical solution of partial differential equations, additional boundary and initial conditions may sometimes need to be introduced [7].

Dynamic simulation was performed for a steel steering shaft structure with the following data: G = 8•104, N/mm^2; ρ= 8•10^{-6}, kg/mm^3; n_x = 4086 h_x = 0.1, mm; n_t = 4096; h_t = 0.001, s.

The values of the number of points and discretization steps used during the dynamic simulation ensured the reliability of the results for parameters in the x-axis direction: 0.0024 to 5, 1/mm and t: 0.244 to 500, Hz [13].

3 DYNAMIC SIMULATION

In the following text, there will be more words about the analysis of torsional vibrations of the steering shaft, using optimization methods. It should be noted that in practice, various procedures are used for these purposes, and here the method of "stochastic parametric optimization" will be applied. As is known, it is used in optimizing the oscillatory parameters of motor vehicles and is based on nonlinear programming methods. Since there are constraints on the 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 used for selecting the diameter of the shaft is based on the Hooke Jeeves method and external penalty functions (the length was accepted from the ergonomic aspect). The optimization method is detailed in reference [11], so it will not be discussed in detail here. The block diagram of the method will be shown in Figure 3 for illustration purposes.

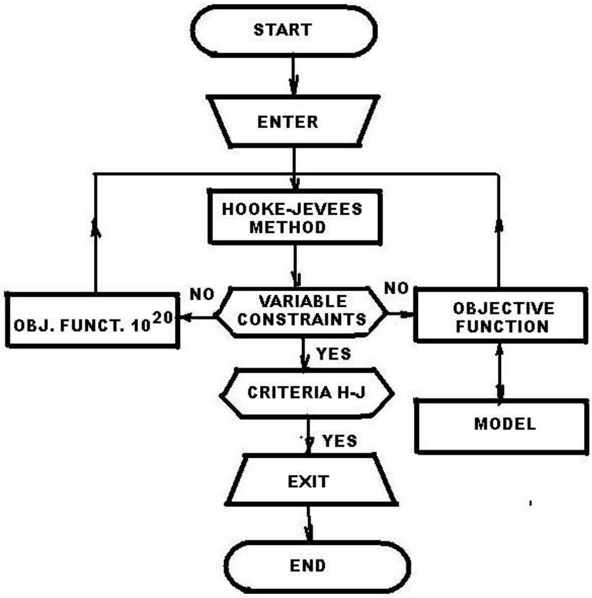

Figure 3. Flow block diagram of the optimization method

The analysis of torsional vibrations of the shaft is focused on minimizing the tangential accelerations of the steering wheel, by using the objective function.

$$Z = Ra_{RMS} \, , \tag{5}$$

where: RMS values of the tangential acceleration of the steering wheel due to torsional vibrations of the axle obtained by solving the partial differential equation (1), R - radius of the steering wheel.

RMS of the tangential vibrations of the steering wheel using the formula:

$$a_{RMS}^2 = \frac{1}{n_x n_t} \sum_{i=1}^{nx} \sum_{j=1}^{nt} a(i,j)^2 \, , \tag{6}$$

where: a(i,j) - tangential accelerations of the steering wheel due to torsional vibrations of the shaft, n_x - number of points along the x-axis, and n_t - number of points along the t-axis.

During the optimal selection process of the shaft unknown dimension, boundary values for the shaft radius were defined: $5 \le r \le 15$

By introducing the optimizing parameter x[i], i=1, instead of r, with corresponding adopted boundary values x_u[i], x_l[i], i=1 (u - upper, l - lower boundary value), the objective function depends on one optimizing parameter and it has more local minima and only one global minimum [10].

Considering that, in practice, finding the global minimum is achieved by starting the optimization process with more initial values of optimizing parameters [2,10,11], it was deemed appropriate in this case to start with three initial parameter values, namely:

x=0.5 x_u[1] x=0.5 x_u[2]

$x=0.8\ x_u[1] \qquad x=0.8\ x_u[2]$

$x=1.2\ x_l[1] \qquad x=1.2\ x_l[2]$

The optimization was performed on a Pentium 4 computer (Intel 2.4, GHz, 9, GB RAM), and the iterative process was automatically stopped when the difference between two adjacent values of the objective function was 10^{-30}. The optimization time for one combination was around 20 minutes, and the calculated parameters are shown in Table 1.

Table 1. Key parameters of the optimal selection of the steering shaft diameter

Initial values	Optimal parameter, r, mm	Objective function, Z, rads^{-2}	No of iter., N,-
$0.5(x_l+x_u)$	5.000	9.814976100393929E-004	574
$0.8\ x_u$	5.000	9.814976100393929E-004	576
$1.2\ x_l$	5.000	9.814976100393929E-004	560

4 DATA ANALYSIS

By analyzing the data from Table 1, it can be concluded that the optimal value of the shaft diameter and the objective function do not depend on the initial value of the optimizing parameter, as identical values are obtained (5, mm).

Considering that the integration of the partial differential equation was performed at 4096 points, due to Excel limitations, torsional vibrations of the steering shaft for the optimal diameter value at 256 points are shown in Figure 4.

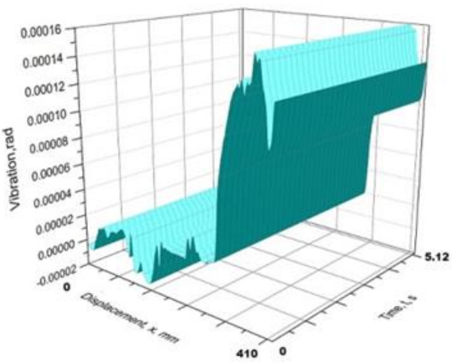

Figure 4. Torsional vibrations of the steering shaft

By analyzing the data from Figure 4, it can be determined that they stochastically change along the length of the shaft. The random nature of the vibrations can be explained by the random nature of the excitation function used, which is in agreement with [5].

It was deemed appropriate to perform a frequency analysis of torsional vibrations using 2D Fourier transform. The author developed software for calculating the parameters of 2D Fourier transform [12], but it was deemed appropriate to use commercial software Origin [13]. The calculated values of the magnitude and phase angles of the 2D Fourier transform are shown in Figures 5 and 6 (for 256 points).

By analyzing the data from Figures 5 and 6, it can be observed that the magnitude is highest at the ends of the axle, and the phase angles change stochastically along its length.

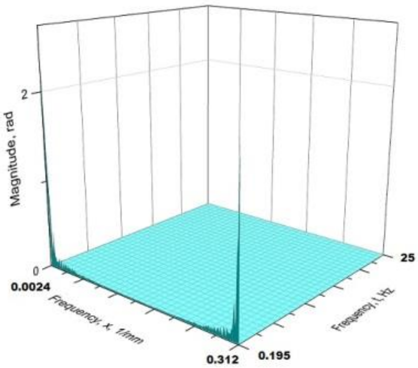

Figure 5. Spectrum magnitude of torsional vibrations of the steering shaft

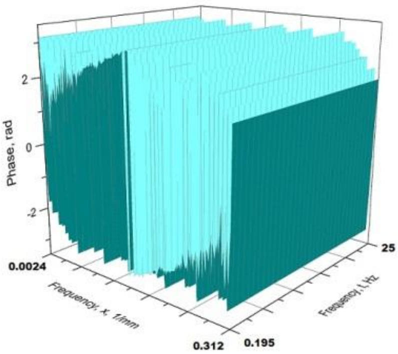

Figure 6. Phase angles of the spectrum of torsional vibrations of the steering shaft

It is noted that the optimal diameter of the steering shaft is calculated based on the conditions of minimal tangential vibrations of the steering wheel, so it can be changed during the later design phase through finite element method verification and experimentation, or the one supplied with the power steering can be adopted, which will not be discussed further here [14].

It is emphasized that based on the adopted standard dimensions, an analysis of the torsional resonance of the steering shaft must be performed and compared with the harmonic frequencies obtained by 2D Fourier transform. The goal is to avoid overlap of axle resonance and wave frequencies, which will not be discussed further here [15]...

It was deemed appropriate to compare the tangential accelerations of the steering wheel with the ISO 5349/DIS criteria [3]. Therefore, tangential accelerations of the steering wheel were calculated and a third-octave frequency analysis was performed, with the results shown in Figure 7.

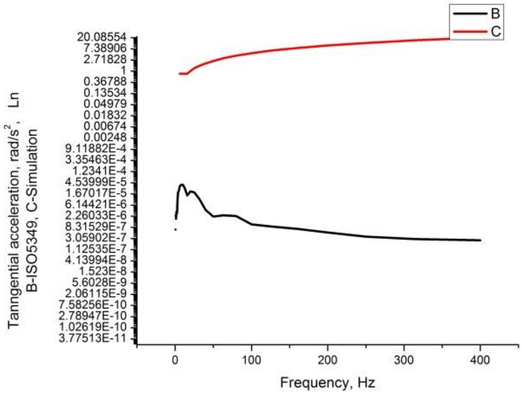

Figure 7. The comparison of ISO 5349/DIS criteria and calculated tangential accelerations (the ordinate axis is logarithmic due to small values of tangential accelerations).

By analyzing the data from Figure 7, it can be observed that the ISO 5349/DIS criterion is satisfied, so the calculated axle radius can be accepted for further analysis.

Since the obtained values of analyzed accelerations are lower than the mentioned criteria, the calculated steering shaft diameter can be used for further analysis. More precisely, it should be compared with the one on the derived steering mechanism and, as a reliable measure, a larger one should be adopted.

It is also noted that in 2D Fourier transform, there are no explicit procedures for calculating errors in spectral analysis, as in the case of 1D Fourier transform [15]. Taking this into account, and considering that this work aims to illustrate the potential application of 2D Fourier transform in the analysis of steering axle torsional vibrations, statistical errors were not calculated.

5 CONCLUSION

The developed procedure, based on the analysis of tangential accelerations of the steering wheel due to torsional vibrations of the steering shaft, allows for the definition of its diameter, which will be compared to the one supplied with the power steering, and for further findings, a larger one is adopted.

In further project development, based on the defined diameter, more detailed calculations can be approached, possibly using the finite element method.

The analyses performed have shown that the use of 2D Fourier transform is desirable for analyzing the torsional vibrations of the steering shaft, as they allow for checking the matching of the excitation frequency and the resonance of the shaft.

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