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DETERMINATION OF REACTION TIME AND INTERVEHICLE SPACING AS IMPORTANT HUMAN BASED MICROSCOPIC TRAFFIC PARAMETERS IN URBAN ENVIRONMENT[1]

Aleksandar Kostik, PhD, assistant professor, ***Milan Kjosevski***, PhD, professor,
Ljupcho Kocarev, PhD professor

UDC: 629.01;519.2

Summary

This paper deals with the human aspects regarding enhanced traffic flow and capacity in urban environments. Based on experimental research and literature reviews, dominant microscopic traffic parameters which are primarily influenced by the driver behavior have been recognized. The focus is set on reaction time and especially on intervehicle spacing at urban intersections, a parameter which is rarely addressed in the appropriate literature. The results showed that reaction time and intervehicle spacing as random variables have lognormal distribution. This finding is proved with appropriate statistical tools like 3 sigma rule, probability-probability function and complementary cumulative distribution function.

Key words: Microscopic traffic parameters, driver behavior, reaction time, intervehicle spacing, lognormal distribution

ODREĐIVANJE VREMENA REAKCIJE VOZAČA I RASTOJANJA IZMEĐU VOZILA KAO ZNAČAJNIH MIKROSKOPSKIH PARAMETARA SAOBRAĆAJA U GRADSKIM USLOVIMA

UDC: 629.01;519.2

Rezime

Rad analizira sve sapekte čoveka u uslovima povećanog protoka i kapaciteta saobraćaja u gradskim uslovima. Na osnovu eksperimentalih istraživanja i pregleda literature, dominantni mikroskopski parametri saobraćaja koji su rezultat ponašanja vozača su determinisani. Fokus je stavljen na vreme reakcije i posebno uočavanje rastojanja između vozila na gradskim raskrsnicama, što je postojećoj literaturi retko korišćeno. Rezultati su pokazali da veme reakcije i rastojanje između vozila su slučajne promenljive koje imaju normalnu logaritamsku raspodlu. Rezultat je dokazan pravilom 3 sigma, funkcijom gustine verovatnoće i komplemetarnom kunulativnom funkcijom raspodele

Ključne reči: Mikroskopski parametri saobraćaja, ponašanje vozača, vreme reakcije, rastojanje između vozila, normalna logaritamska raspodela

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DETERMINATION OF REACTION TIME AND INTERVEHICLE SPACING AS IMPORTANT HUMAN BASED MICROSCOPIC TRAFFIC PARAMETERS IN URBAN ENVIRONMENT

Aleksandar Kostik [1], ***Milan Kjosevski***, ***Ljupcho Kocarev***

UDC:629.01;519.2

INTRODUCTION

Traffic stream primarily depends on individual driver and vehicle behaviour and on the way they react to each other. In order to understand traffic stream characteristics, certain parameters have been defined as a quantitative measure for their description. Traffic stream parameters are classified in two broad categories: macroscopic parameters which define the traffic stream as a whole, and microscopic parameters which characterize the behaviour of individual drivers and vehicles in the traffic stream with respect to each other [2]. Furthermore, microscopic parameters can be treated as mutually vehicle and human dependent (acceleration, speed, travelled distance, time, etc.) and purely or primarily human dependent.

The reaction time as a human factor is probably the most exploited microscopic traffic parameter regarding virtual modelling and simulation of traffic flow. It encounters the driver reaction time and reaction time of the vehicle, and is one of the main input parameters in general acceleration models which are implemented in microscopic traffic simulators. It is shown that the reaction time, as a random variable, has lognormal distribution with appropriate mean value and standard deviation [1, 3, 4].

Another important human based microscopic parameter is intervehicle spacing. This parameter depends on the reaction time and is primarily used as a control parameter in the modelling and simulation process, and in the spacing policy design regarding longitudinal vehicle control.

Based on experimental research in urban environment and literature reviews, the above mentioned parameters which are primarily influenced by the driver behaviour are recognized as dominant microscopic traffic parameters regarding enhanced traffic flow and capacity in urban environments.

In this paper the focus is set on the reaction time and intervehicle spacing at urban intersections i.e. when vehicles are stationary and start moving due to appearance of the green traffic light. As it is stated before, the reaction time is well exploited traffic parameter. On the other hand, intervehicle spacing as a random variable is rarely addressed in the appropriate literature. The importance of these parameters in terms of ITS is confirmed in wider research activities involving development of urban traffic simulator, focused on one boulevard in urban environment.

The remainder of the paper is organized as follows: The method for determining the reaction time and the intervehicle spacing at urban intersections is presented in section II. Also, the obtained results are presented in section II. The distribution of the reaction time and the intervehicle spacing as random variables is defined and proved with appropriate

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statistical tools in section III and IV respectively. Section V covers the methods regarding optimization of the subject traffic parameters. Finally, conclusions are drawn in the last section.

1. REACTION TIME AND INTERVEHICLE SPACING DETERMINING METHOD

The reaction time represents the time delay from the moment of appearance of stimulation till the moment of appearance of the response to the stimulation. It includes the perception time, arms and feet movement time and the response time of the vehicle. The perception time is the time when the driver recognizes the stimulation and decides how to react to it. The arms and feet movement time is the time required for the physical movements to be taken. The vehicle response time is the time the vehicular systems take to respond to the driver's inputs (brake, throttle and steering). The reaction time is affected by characteristics of the driver (e.g. age, alertness, physical condition) and the vehicle as well as by environmental conditions (e.g. weather conditions, visibility, road geometry) [4].

The reaction time and intervehicle spacing at urban intersections when vehicles are stationary and start moving are determined using digital video cameras and appropriate post processing of the recorded material. The records were made on several intersections on one of the main arterial roads in the city of Skopje. The reaction time is determined measuring the time from the moment of appearance of the green traffic light till the moment when the first vehicle in the certain traffic lain starts moving as well as measuring the time when the vehicle predecessor starts moving till the time when the vehicle successor starts moving. Considering the fact that the records were made with 14 frames per second, the results are obtained with accuracy of 0.07 seconds. In this way we have determined totally 1691 value of reaction time. These values are shown on fig.1 in ascending order. The determined reaction time is between 0s and 5.83s. It has a mean value of 1.267s and standard deviation of 0.84s. The obtained results correspond with the values in the appropriate literature [4].

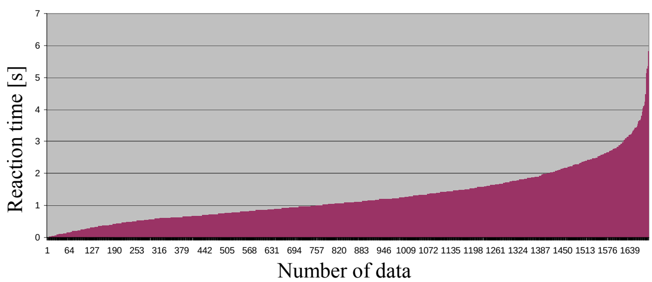

Figure 1: *Reaction time in ascending order*

In order to determine the intervehicle spacing at urban intersections, frames that represent stationary vehicles during the red traffic light were extracted and analyzed. Intervehicle spacing is determined in relation to already known distance on the subject frame (fig. 2). Aiming towards higher precision, each traffic lane is divided into three zones with different length ratios. In this way we have encountered different distances between the camera and the vehicles on the road, and also different camera positions with respect to the observed vehicles.

Figure 2: *Three central zones with different length ratios*

With this method we have determined totally 1413 values of intervehicle spacing which are shown on fig.3 in ascending order. The determined intervehicle spacing is between 0.27m and 11.26m. It has a mean value of 2.07m and standard deviation of 1m.

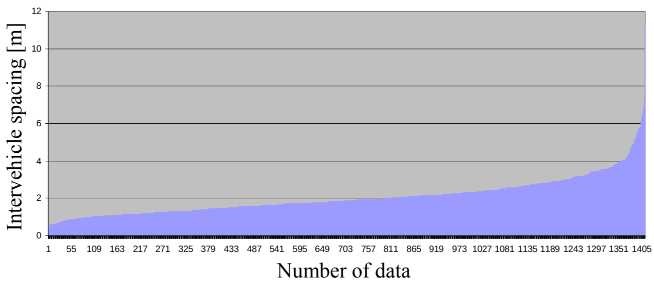

Figure 3: *Intervehicle spacing in ascending order*

2. REACTION TIME DISTRIBUTION

The reaction time is modeled as a random variable in order to cover the variability of the parameters that depends on. The lognormal probability function is widely accepted and used to describe the distribution of reaction times [1, 3, 4]. In the remainder of the section we have confirmed the lognormal distribution of the reaction time using several statistical tools.

The histogram of the determined reaction time (fig.4) is the first indicator of its lognormal distribution.

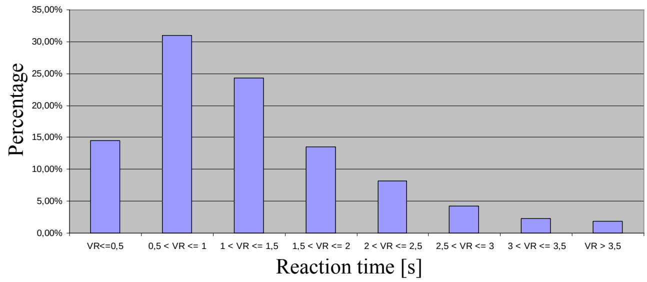

Figure 4: *Histogram of the reaction time*

The main indicator of lognormal distribution of the reaction time is the normal distribution of its logarithm which is proved with the 3 sigma rule, i.e. 68-95-99.7 rule for normal distribution. The percentage with which the logarithm of the reaction time is represented in the intervals $\mu\pm\sigma$, $\mu\pm2\sigma$ and $\mu\pm3\sigma$ is shown in table 1. The obtained results show that the 3 sigma rule is fulfilled in a 95 percent confidence interval i.e. the reaction time as random variable, at urban intersections when vehicles are stationary and start moving, has lognormal distribution with mean value of -0,0138 and standard deviation of 0,812807.

Table 1: *Percentage with which the logarithm of the reaction time is represented in the intervals $\mu\pm\sigma$, $\mu\pm2\sigma$ and $\mu\pm3\sigma$*

Interval	$\mu\pm\sigma$ (>68%)	$\mu\pm2\sigma$ (>95%)	$\mu\pm3\sigma$ (>99,7%)
Percentage	75,28%	95,32%	98,40%

The Probability Density Function (PDF) *f(x)* and the Cumulative Distribution Function (CDF) *F(x)* of the lognormal distribution of the reaction time are defined with the following equations and shown on fig. 5 and fig.6,

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where x is random variable with lognormal distribution, μ is mean value of ln(x) and σ is standard deviation of ln(x).

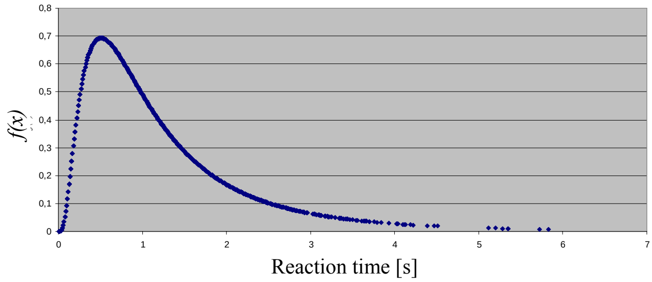

Figure 5: *Probability Density Function (PDF) f(x)*

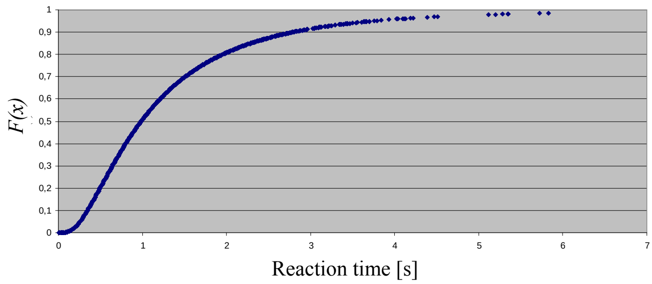

Figure 6: *Cumulative Distribution Function (CDF) F(x)*

The P-P (Probability-Probability) plot defining the CDF in relation to the Empirical Probability (EP) of the reaction time is shown on fig. 7.

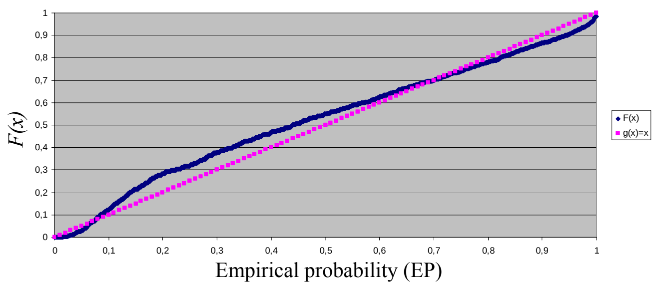

Figure 7: *P-P (probability-probability) plot*

The empirical probability is defined with the following equation:

$$EP=(i-\sigma)/n, \tag{3}$$

where $i=1...n$ is the index of the value in the data set and n is the total number of values in the data set.

Relatively good fit of the functions F(x) and g(x) is an additional indicator of the lognormal distribution of the reaction time.

The Complementary Cumulative Distribution Function (CCDF) *1-F(x)* of the lognormal distribution of the reaction time on log-log scale is shown on fig. 8. The straight part at the tail of the function is another indicator of the lognormal distribution of the reaction time.

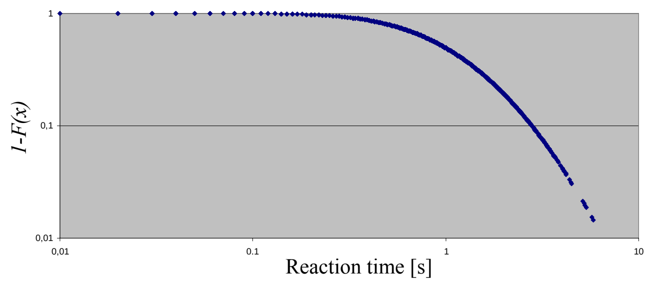

Figure 8. *Complementary Cumulative Distribution Function (CCDF) 1-F(x)*

3. INTERVEHICLE SPACING DISTRIBUTION

As it is stated before, the intervehicle spacing at urban intersections is random variable which is primarily influenced by drivers' current condition and his attitude towards driving. Considering the fact that it is used as a control parameter in the modelling and simulation process, and in the spacing policy design regarding longitudinal vehicle control, it is of great importance to determine its distribution. In order to do so, the above mentioned statistical tools are used.

The histogram of the determined intervehicle spacing (fig.9) is the first indicator of lognormal distribution.

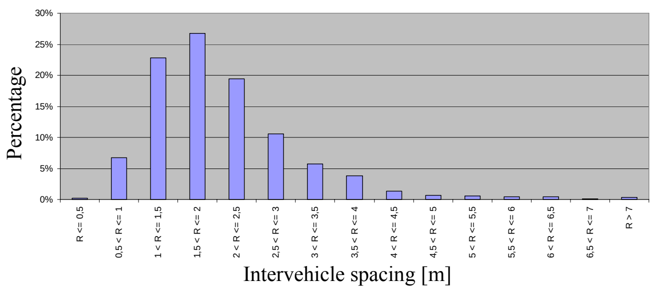

Figure.9. *Histogram of the intervehicle spacing*

The main indicator of lognormal distribution of the intervehicle spacing is the normal distribution of its logarithm which is proved with the 3 sigma rule, i.e. 68-95-99.7 rule for normal distribution. The percentage with which the logarithm of the intervehicle spacing is represented in the intervals $\mu\pm\sigma$, $\mu\pm2\sigma$ and $\mu\pm3\sigma$ is shown in table 2. The obtained results show that the 3 sigma rule is fulfilled in a 99 percent confidence interval i.e. the intervehicle spacing at urban intersections as random variable has lognormal distribution with mean value of 0.627277 and standard deviation of 0.444044.

Table 2: *Percentage with which the logarithm of the intervehicle spacing is represented in the intervals μ±σ, μ±2σ and μ±3σ*

Interval	μ±σ (>68%)	μ±2σ (>95%)	μ±3σ (>99,7%)
Percentage	69,07%	95,05%	99,58%

The Probability Density Function (PDF) *f(x)* and the Cumulative Distribution Function (CDF) *F(x)* of the lognormal distribution of the intervehicle spacing are shown on fig. 10 and fig.11.

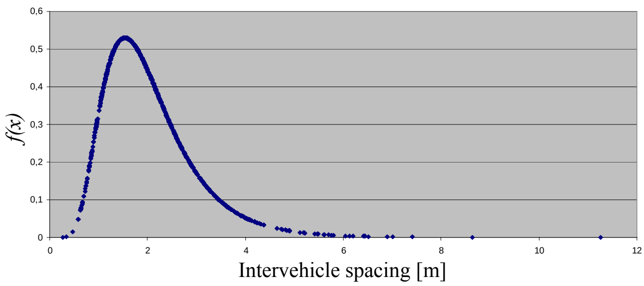

Figure 10: *Probability Density Function (PDF) f(x)*

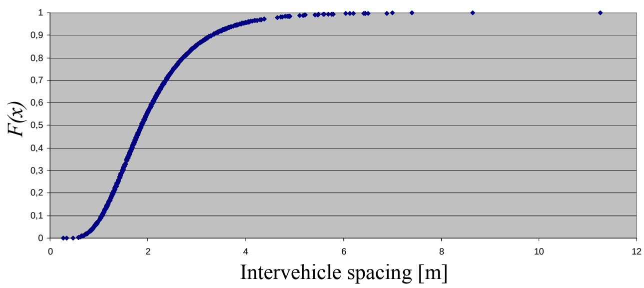

Figure 11: *Cumulative Distribution Function (CDF) F(x)*

The P-P (Probability-Probability) plot defining the CDF in relation to the Empirical Probability (EP) of the intervehicle spacing is shown on fig. 12.

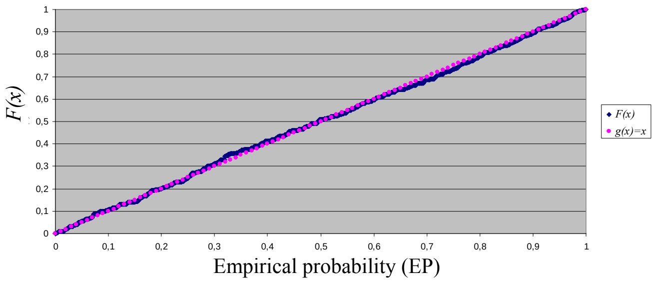

Figure 12: *P-P (probability-probability) plot*

Almost perfect fit of the functions F(x) and g(x) is an additional indicator of the lognormal distribution of the intervehicle spacing.

The Complementary Cumulative Distribution Function (CCDF) *1-F(x)* of the lognormal distribution of the intervehicle spacing on log-log scale is shown on fig. 13. The straight part at the tail of the function is another indicator of the lognormal distribution of the intervehicle spacing.

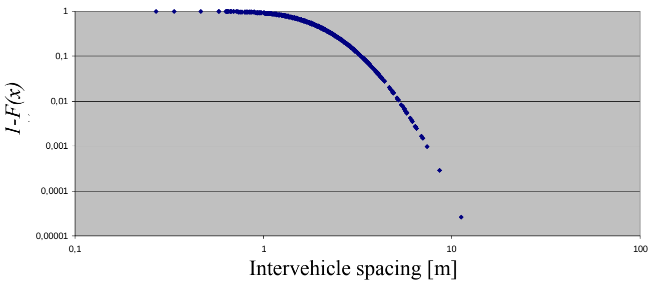

Figure 13: *Complementary Cumulative Distribution Function (CCDF) 1-F(x)*

4. REACTION TIME AND INTERVEHICLE SPACING OPTIMIZATION METHODS

The importance of the reaction time and the intervehicle spacing at urban intersections as microscopic parameters in terms of ITS is confirmed in wider research activities involving development of urban traffic simulator, focused on one boulevard in urban environment. It is found that lower values of reaction time and intervehicle spacing at urban intersections have certain positive influence on the traffic flow and capacity. In order to enhance the traffic flow and capacity, we have defined two methods in the course of optimization of the subject parameters.

The first one is focused on vehicles already involved in the traffic. According to it, these vehicles could be equipped with relatively simple and commercially accessible information and warning systems. Based on the information of these systems the drivers would be stimulated to react faster and obtain optimal intervehicle spacing at urban intersections. Additionally this information would keep the drivers attention towards the driving task. An example of such system is a distance sensing device. More sophisticated solutions anticipate implementation of ICT in vehicles.

The second method is focused on the educational process of the drivers about the benefit of the optimized reaction time and intervehicle spacing. Its goal is to change the drivers overall attitude towards the driving task through different forms of information dissemination.

5. CONCLUSIONS

The area of interest presented in this paper has been very attractive for researchers for the past few years. The expectations that these researches would lead to additional traffic benefit, regarding vehicle flow and traffic capacity, are justified.

Based on the results gained from the performed data acquisition and analyzes, it can be concluded that the presented methodology is appropriate for this kind of a challenge. The reaction time and the intervehicle spacing at urban intersections, when vehicles are stationary and start moving, are important microscopis human based traffic parameters. It is confirmed that the reaction time has lognormal distribution. Also, it is proved that the intervehicle spacing as a random variable has a lognormal distribution.

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[1]STATIC AND DYNAMIC ANALYSIS OF HYBRID METAL - COMPOSITE SHAFTS

Zorica Đorđević, PhD, Assistant professor, ***Nenad Kostić***, PhD student,
Saša Jovanović, MSc Assistant, ***Vesna Marjanović***, PhD, Assistant professor,
Mirko Blagojevic, PhD, Assistant professor,

UDC:621.8

Summary

Day after day, our needs are growing, so it is necessary to develop new technologies, including and using new "smart" materials that will replace the traditional ones. New materials are a combination of two or more materials and are collectively referred to as composite materials, having more superior performance. In this paper, a method for calculating the composite shaft and numerical model applied and analyzed in the programs FEMAP and NXNASTRAN, is developed. The advantages of the composite shaft in comparison to metal shaft are: light weight, high strength and hardness, increased resistance to fatigue, wear vibration and acoustic loads, greater reliability in use and corrosion resistance of parts,.. Based on the obtained results, the application of composite materials is recommended.

Key words: composite material, shaft, displacement, natural frequencies

STATIČKA I DINAMIČKA ANALIZA KOMBINOVANIH METAL - KOMPOZITNIH VRATILA

UDC:621.8

Rezime

Svakim danom naše potrebe rastu, pa je neophodno razvijati nove tehnologije, uključujući i korišćenje novih "modernih" materijala koji će zameniti tradicionalne. Novi materijali su kombinacija dva ili više različitih materijala koji zajedno, kao kompozitni materijali, imaju povoljnije karakteristike u odnosu na sastavne komponente.U ovom radu je razvijen numerički model za proračun kompozitnog vratila korišćenjem programa FEMAP i NXNASTRAN. Prednosti kompozitnih, u odnosu na metalna vratila, su: manja masa, velika čvrstoća i tvrdoća, povećana otpornost na zamor, habanje, vibraciona i akustička opterećenja, veća pouzdanost u eksploataciji i koroziona postojanost delova,… Na osnovu dobijenih rezultata preporučuje se primena kompozitnih materijala za izradu vratila.

Ključne reči: kompozitni materijali, vratila, pomeranje, sopstvene frekvencije

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STATIC AND DYNAMIC ANALYSIS OF HYBRID METAL - COMPOSITE SHAFTS

Zorica Đorđević[1], Nenad Kostić, Saša Jovanović, Vesna Marjanović, Mirko Blagojević

UDC:621.8

1. INTRODUCTION

Composite materials are obtained by combining two or more materials, all in order to make new materials with controlled and more favourable characteristics. As a result, these materials have improved mainly mechanical properties, such as strength, stiffness, toughness, than their constituent parts. The greatest advantage of composite materials is reflected in the fact that most of them use the best features of the component materials and characteristics that themselves do not possess individually. Mainly, the formation of composite materials improve the following characteristics: strength, stiffness, corrosion resistance, abrasion resistance, weight reduction, lifetime, thermal isolation, acoustic isolation and increase elasticity modulus. Replacement of steel composite shaft is a novelty all over the world, especially in our country. Composite materials have been applied, as mentioned earlier, in aviation with increased accent on their usage analysis in the automotive industry and all other vehicle types. The above paper review is relating to some, the most characteristic, work in this field. Naveen Rastogi in his scientific research [1] presented a method for calculating the drive shaft , used in a car. He gave two important aspects of the input shaft of the budget: the budget and budget composite shaft coupling between the joints and back, discussing about solutions. Gubran H.B.H and Gupta K. [2] carried out theoretical and experimental studies on deflection and cross-section deformation of tubular composite shafts subjected to point static loading.

D.G. Lee together presented in [3] explained the new way of manufacturing a single piece of combined aluminium / composite drive shafts for cars with rear wheel drive. The composite material is composed of several layers that are made by the inner surface of the aluminium tubes. T. Rangaswamy and co-authors in the work [4] gave a realistic budget and analysis of composite drive shafts for power transmission, the optimal budget single piece drive shaft cars with rear-wheel drive using a highly modular and E-glass/epoxy (NM) carbon / epoxy composite. Gubran H.B.H. in [5] analyzed the deformation of the cross section of the shaft made of metal (steel or aluminium), composites (CFRP and GFRP) and combined return obtained from metals and composites. Special attention is devoted to the analysis of combined back. It is concluded that deformation of the shaft, and thus the deformation of cross sections, can be reduced if, instead of pure composite shaft, combined shaft is used (composite + metal).

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2. TYPES AND PROPERTIES OF COMPOSITES

Composites can be classified in different ways and by different criteria. From the point of this paper, the corresponding division of the composite would be deployed into the three main groups:

- composites with discrete particles embedded in a matrix,
- fiber-reinforced composites,
- layered composite laminates.

The characteristics of composites depend largely on three factors:

- strength and chemical stability of the matrix,
- strength and elasticity of reinforcing fibers,
- bond strength between the matrix and fibers arming.

Glass, carbon, organic (kevlar-aramid etc.) and boron fibers are the ones that are most often used for construction composites [4, 6, 7,...].

Glass fiber/polyester or glass fiber/epoxy resin composites are widely used in practice. The advantages of glass fibers over other materials are the following: easy and cheap production, possibility to produce very long fibers, high resistance to impact, etc. The basic disadvantages of glass-fiber-reinforced composites lie in the fact that they have a low modulus of elasticity and that they lose resistance at elevated temperatures.

The need for stronger materials led to greater application of carbon fiber/epoxy resin composites. Carbon fibers have high strength, high modulus of elasticity, low density, excellent mach inability, resistance to elevated temperatures, low thermal expansion coefficient, etc. Their main disadvantages are low toughness, high anisotropy, which causes additional problems to the constructor, and a high production cost compared to glass fibers.

The best-known organic fibers are aramid fibers, such as Kevlar. Aramid fibers have extremely high tensile strength, low density, excellent impact resistance, excellent isolating and heat properties, they are stable at a wide temperature range, they neither melt nor shrink, they are easy to machine and they can be produced in the form of weave. Their disadvantages are low compression, rather low modulus of elasticity, they are difficult to obtain, and they have a high production cost.

Boron fibers are characterised by high compression strength and torsion resistance. They have a positive coefficient of linear expansion. Their disadvantages are difficulty to machine and shape due to their high hardness, and a high production cost.

3. THE DESIGN OF THE COMPOSITE SHAFT

The composite shaft often has a shape of laminated cylindrical shell. It is important to derive the expressions for stress calculation and deformations in case of such laminate composite shaft. Those equations were derived by *Chang* and *Chen* [6] using the *Hamilton's* principle.

The constitutive relations for a lamina (see Figure 1) in the principal material directions (indicated by 1, 2 and 3) are given by:

$$\begin{Bmatrix} \sigma_{11} \\ \sigma_{22} \\ \sigma_{33} \\ \tau_{23} \\ \tau_{31} \\ \tau_{12} \end{Bmatrix} = \begin{bmatrix} Q_{11} & Q_{12} & Q_{13} & 0 & 0 & 0 \\ Q_{12} & Q_{22} & Q_{23} & 0 & 0 & 0 \\ Q_{13} & Q_{23} & Q_{33} & 0 & 0 & 0 \\ 0 & 0 & 0 & Q_{44} & 0 & 0 \\ 0 & 0 & 0 & 0 & Q_{55} & 0 \\ 0 & 0 & 0 & 0 & 0 & Q_{66} \end{bmatrix} \begin{Bmatrix} \varepsilon_{11} \\ \varepsilon_{22} \\ \varepsilon_{33} \\ \gamma_{23} \\ \gamma_{31} \\ \gamma_{12} \end{Bmatrix} \tag{1}$$

The above equation will abbreviated as:

$$\{\sigma\} = [Q]\{\varepsilon\} \tag{2}$$

where [Q] is the stiffness matrix.

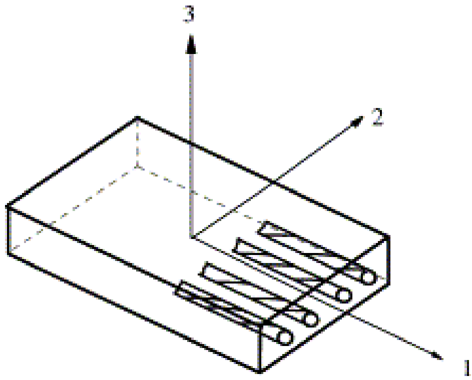


Figure 1: *A typical composite lamina and its principal material axes*

Applying the Galerkin procedure the following matrix equations of motion of the spinning shaft system can be found:

$$[M]\{\ddot{q}\} + \left(\Omega[G] + [C]\right)\{\dot{q}\} + [K]\{q\} = \{F\} \tag{3}$$

where [M] represents the mass matrix, [G] the gyroscopic matrix, [C] the damping matrix, [K] the stiffness matrix, {F} the external force vector and {q} the displacement vector.

The lamina is thin it is considered as the plane stress problem. Hence, it is possible to reduce the 3-D problem into 2-D problem. For unidirectional 2-D lamina, the stress-strain relation ship in terms of physical material direction is given by:

$$\begin{Bmatrix} \sigma_1 \\ \sigma_2 \\ \tau_{12} \end{Bmatrix} = \begin{bmatrix} Q_{11} & Q_{12} & 0 \\ Q_{12} & Q_{22} & 0 \\ 0 & 0 & Q_{66} \end{bmatrix} \begin{Bmatrix} \varepsilon_1 \\ \varepsilon_2 \\ \gamma_{12} \end{Bmatrix} \tag{4}$$

The matrix [Q] is referred as the reduced stiffness matrix for the layer and its terms are given by:

$$Q_{11}=\frac{E_{11}}{1-\nu_{12}\nu_{21}}; \qquad Q_{12}=\frac{\nu_{12}E_{22}}{1-\nu_{12}\nu_{21}}$$

$$Q_{22}=\frac{E_{22}}{1-\nu_{12}\nu_{21}}; \qquad Q_{66}=G_{12}$$

For an angle-ply lamina, where fibers are oriented at an angle with the positive x-axis (Longitudinal axis of shaft), the stress strain relationship is given by:

$$\begin{Bmatrix}\sigma_x\\ \sigma_y\\ \tau_{xy}\end{Bmatrix}=\begin{bmatrix}\bar{Q}_{11} & \bar{Q}_{12} & \bar{Q}_{16}\\ \bar{Q}_{12} & \bar{Q}_{22} & \bar{Q}_{26}\\ \bar{Q}_{16} & \bar{Q}_{26} & \bar{Q}_{66}\end{bmatrix}\begin{Bmatrix}\varepsilon_x\\ \varepsilon_y\\ \gamma_{xy}\end{Bmatrix} \tag{5}$$

4. APPLICATION OF NUMERICAL METHOD OF STATIC ANALYSIS OF COMPOSITE SHAFTS

Differential equations that describe the state of stress and state of deformity, especially when it comes to complex systems, often cannot be solved by analytic methods. In these cases a solution can be reached by applying numerical methods. The main feature of the numerical method consists of the fact that the fundamental equations of Theory of elasticity are solved in a similar, numerical way. One of the most used numerical methods is the final elements one. In the discretization of the continuum, one type of final elements or combination of more types can be used. For hollow shaft analysis, as the case with composite shaft is, commonly used elements are in the shell form, that is, in the form of multilayer shells. For layered shells is typical that, by using suitable fiber orientation along with small thicknesses, it is possible to achieve very high stiffness of the shell structure. In formulating a final element multi-layered shell calculation of all sizes is done in layers [7-10].

All numerical methods involve the usage of computers and appropriate software packages. Nowadays everywhere in the world as well as in our country, there are many software packages developed, such as NX Nastran, SAP, PAK, I-DEASm SESAM, COSSMOS / M, PAL2, ANSYS ...

5. DEVELOPMENT OF MODELS FOR THE NUMERICAL ANALYSIS OF COMPOSITE SHAFTS

Basic characteristics of metal materials (steel and aluminum) and composite materials, that are commonly used for making shafts (carbon fiber / epoxy resin, glass fiber / epoxy resin, aramid fiber / epoxy resin, boron fiber / epoxy resin) are given in Table 1 [7].

Table 1: *Basic characteristic of material*

Material	E_1, MPa	E_2, MPa	G_{12}, MPa	ν	ρ, kg/m^3
Steel	210000	210000	83000	0,3	7830
Aluminium	70000	70000	28000	0,28	2600
Carbon fiber / Epoxy resin	131600	8200	4500	0,281	1550

Glass fiber / Epoxy resin	43300	14700	4400	0,3	2100
Aramid fiber / Epoxy resin	81800	51000	1510	0,31	1380
Born fiber / Epoxy resin	211000	24100	6900	0,36	1967

The labels in the table are: E_1-modulus in the longitudinal direction; E_2 - modulus of elasticity in the transverse direction; G_{12}-modulus, ν - Poisson's ratio, ρ-density of material. Dimensions of the shaft, analyzed in this paper are: length of the shaft is 1000 mm, mean radius is 50 mm, wall thickness of the ring cross section is 4 mm. The shaft is supported at the ends and the middle of the span is subject to static load of 1000N.

Model of analyzed composite shaft, obtained in the program package FEMAP v9.3 is shown in the Fig.2 [10]. The analysis used rectangular isoparametric final elements of multilayer shells form so that the shaft is divided into 8 elements in the axial and 12 elements in the circular direction.

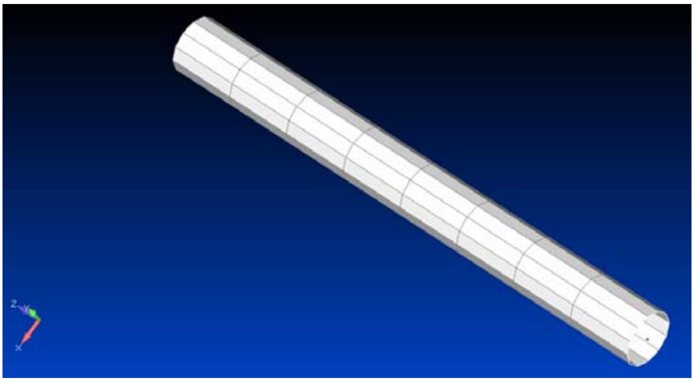


Figure 2: *Composite shaft model*

The process of analysis has began with metal shaft example (steel and aluminum). Figures 3 and 4 show the deformed shape and value displacement of these shaft materials.

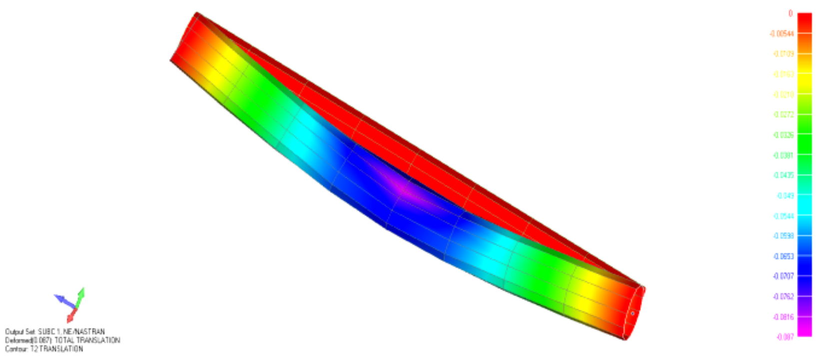


Figure 3: *Value displacement of the steel shaft*

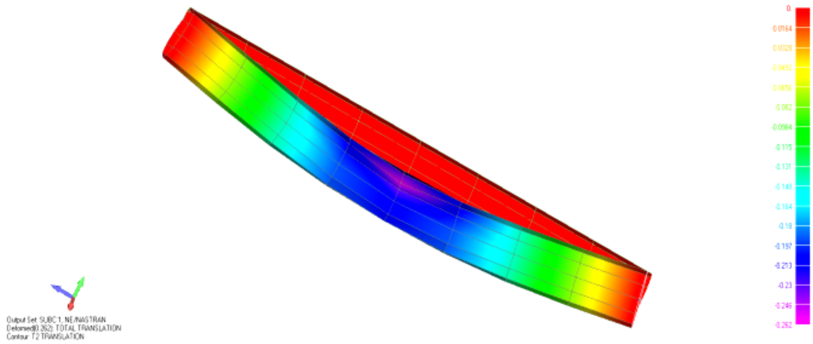
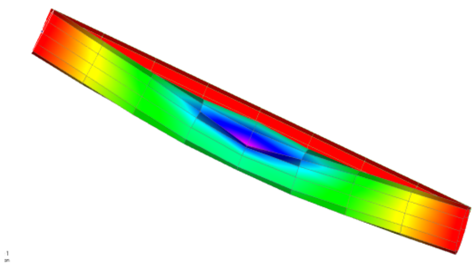
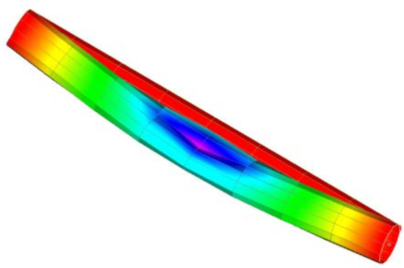


Figure 4: *Value displacement of the aluminum shaft*

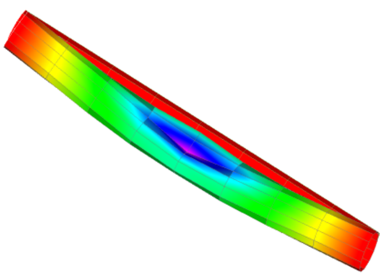
According to figures 5 and 6, the appearance of deformed shafts and value displacement is given in case of steel and composite materials combination with fiber orientation by 0º and 30º. Results which are presented in figure 6 and figure 7 are created on the same way like for orientation fiber at 0º.



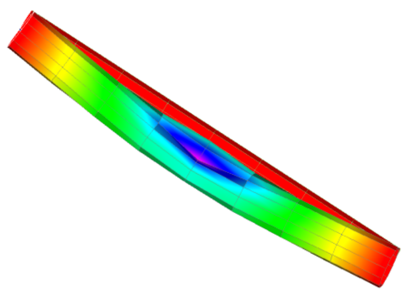
Steel-carbon fiber / epoxy resin $[0_2]$



Steel and glass fiber / epoxy resin $[0_2]$



Steel-aramid fiber / epoxy resin $[0_2]$



Steel-boron fiber / epoxy resin $[0_2]$

Figure 5: *Value displacement of the shaft in the case of hybrid fiber orientation by 0º*

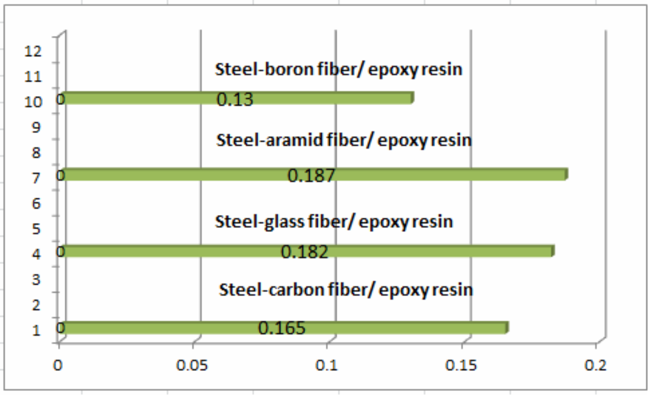


Figure 6: *Value displacement of the shaft in the case of hybrid fiber orientation of 30°*

In the 90 degree fiber orientation, value displacement obtained for the case of hybrid shafts are shown in figure 7.

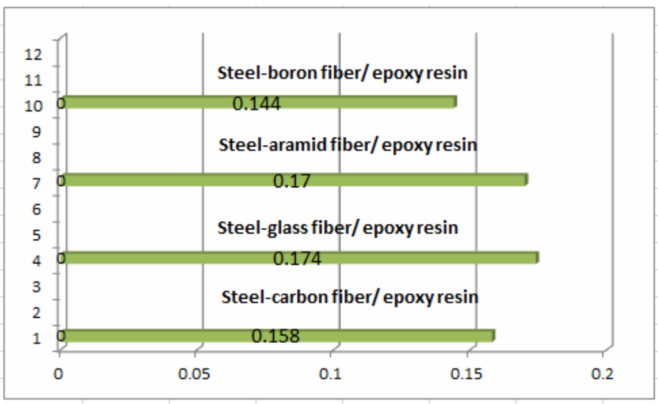


Figure 7: *Value displacement in the case of hybrid fiber orientation by 90°*

It can be concluded that in the case of the shaft obtained from a steel and composite materials combination, the best results, that is, the minimum deflection value boron fibers have, whereas, the worst still are aramid fiber / epoxy resin.

It can also be concluded that all have shafts have approximately similar displacement values combined with steel and composite materials with fiber orientation angles by 90º. This is explained by the fact that in the case of fiber orientation at 90º, elasticity modules in the transverse direction of the composite material is very small compared to steel. Therefore, in that case, displacement value, as well as shaft deformation depends a lot on the content of the steel part.

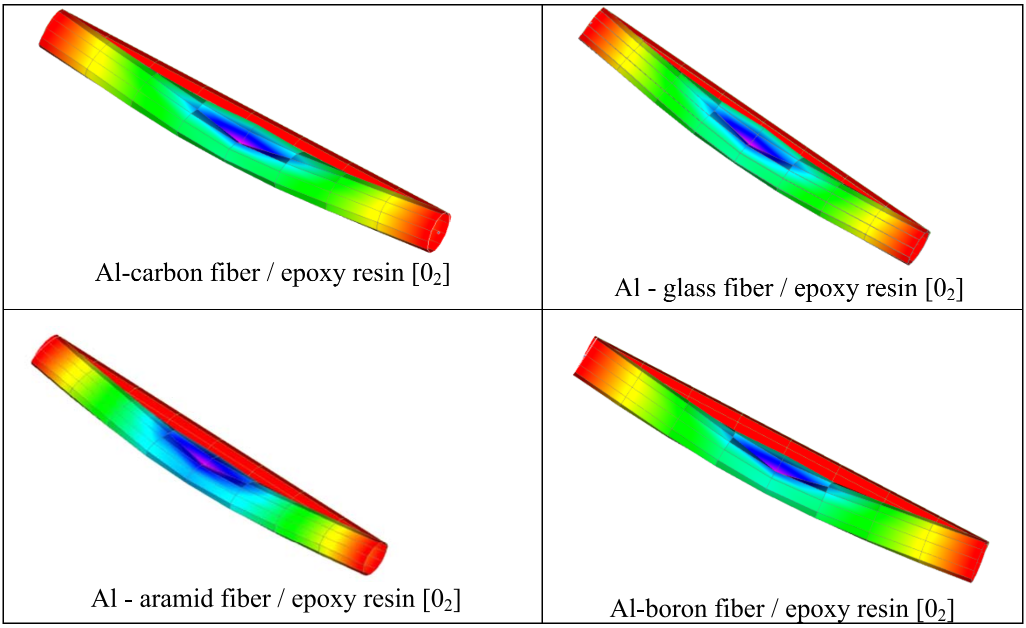


Figure 8: *Displacement value of shaft in the case of hybrid fiber orientation at 0°*

In figures 8 and 9, the appearance of deformed shaft and value displacement are given in case of aluminium and composite materials combination with fiber orientation at 0° and 30°. Results which are presented in figures 9 and 10 are created on the same way like for orientation fiber at 0°.

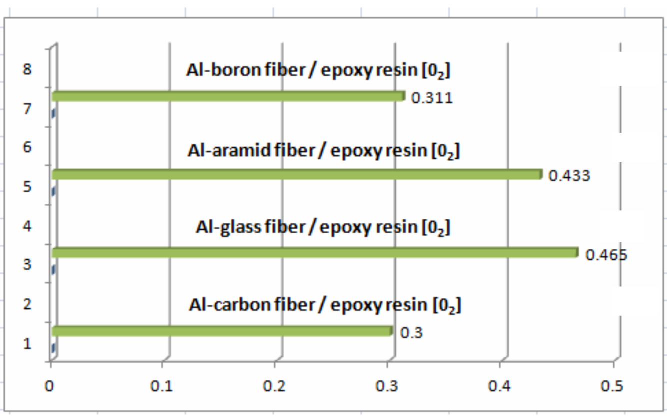


Figure 9: *Displacement value of the shaft in the case of hybrid fiber orientation of 30°*

In the fiber orientation at 90° degrees in case of combination of aluminum and composite materials obtained values of displacement are shown in Figure 10.

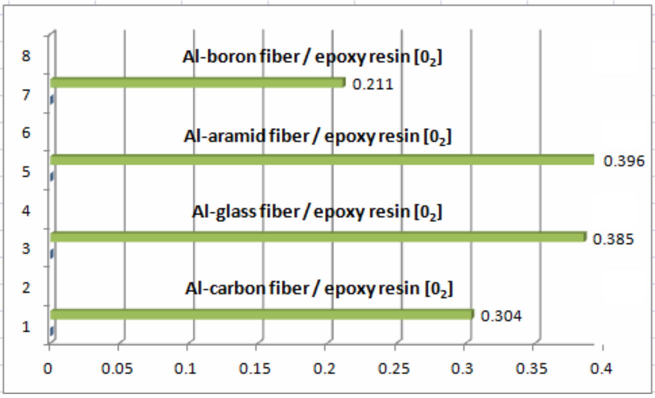


Figure 10 : *Displacement value of the shaft in the case of hybrid fiber orientation at 90^o*

The conclusions drawn for the case of combining steel and composite materials are also valid here. So, here is a recommendation to use a combination of aluminum and boron fiber orientation at angle of 0^o. It should be noted that the density of aluminum is much lower than the density of steel, so this type of combined shaft is of much less weight and high durability, having for the result the combination more common in practice.

When it comes to laminate composites, as is this case here, it should be known that their strength depends on the strength of each layer which exists in it. Failure occurs in the layer with the most critical stress state. In order to eliminate the occurrence of failures in all laminate layers, it is necessary to define the permitted area for each layer. Resistance of laminated shell depends on the ratio of the initial failure FI; the smaller the fracture coefficient, the higher the resistance of the shell during the load impact. According to *Tsai-Wu* failure criterion,to avoid fracture in any layer, the condition is the FI <1. This criterion is met for all types of analyzed materials and all the angles of fiber orientation.

6. FUNDAMENTAL NATURAL FREQUENCY

For the shaft, which has been subjected to static analysis, considered were also natural frequencies. It is known that the values of natural frequencies depend on the ratio E_1/ρ [5], [7]; the ratio is almost the same for the shaft made of steel or aluminum, hence the value of natural frequencies of the shaft of steel or aluminum are almost the same as it can be seen from Figure 11. However, in the case of the composites, ratio E_1/ρ varies, and depends on the orientation of the fibers, the maximum is for fibers angle of 0 ^{o}, and decreases when the fiber orientation angle approaches 90 °.

By analysis of Figure 11 can also be concluded that the worst characteristics, in terms of the value of natural frequencies, a hybrid composite shaft made of Steel-glass and Al-glass materials have. Given the small value of the E1/ρ ratio, shaft natural frequencies of glass are for all values of fiber orientation angles less than metal ones.

Therefore, it can be concluded that the angle of fibers orientation has a major impact on the value of natural frequencies and the corresponding choice of the optimal fiber orientation angle can significantly affect the dynamic characteristics of the shaft [7], [11].

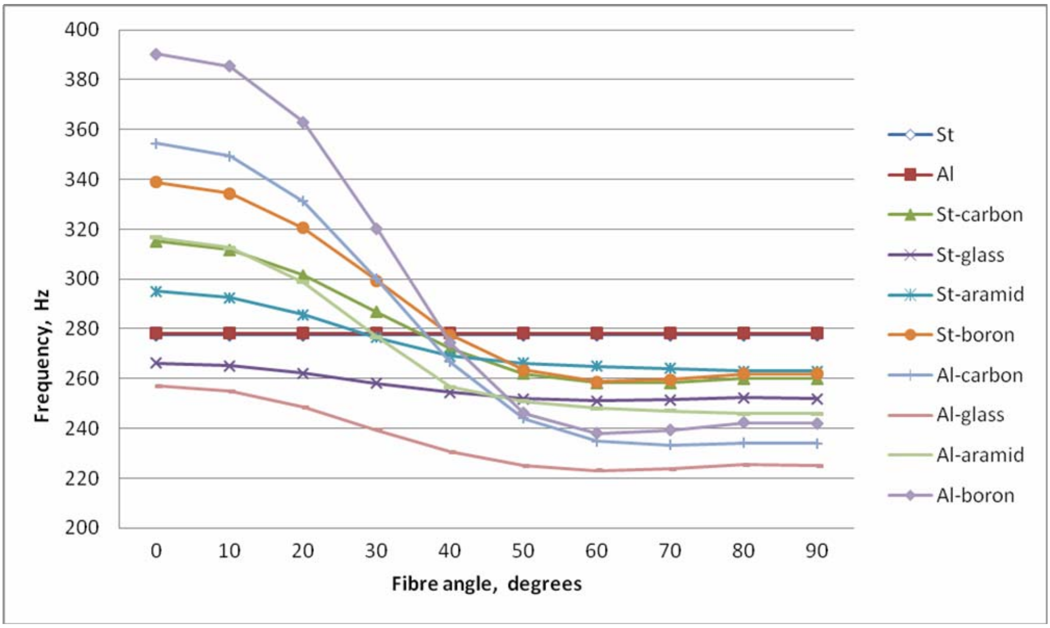


Figure 11: *Natural frequencies of shafts made of different materials*

7. CONCLUSIONS

The aim of this paper was to perform a complete analysis of the deformation state, as well as estimate of its own frequencies of composite shafts (made of carbon, glass, aramid and boron materials), and hybrid metal-composite shafts and compare their behavior with metal shafts (made of steel or aluminum).

Thanks to all the advantages that applying of composite material provide, like light weight, high strength and stiffness, good striking tenacity, better fatigue resistance, abrasion, vibration and acoustic resistance, it is so common nowadays for shafts to be made of composite material. It should be noted that the density of aluminum is much lower than the density of steel, so this type of combined shaft is of much less weight and high durability, having for the result the combination more common in practice.

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[1]NOISE OF ROAD VEHICLES IN THE VICINITY OF THE EXHAUST SYSTEMS

Valentina Golubović-Bugarski, PhD Assistant professor
Snežana Petković, PhD Associate professor

UDC: 534.6; 681.8

Summary

Measurement methodology of stationary vehicle noise has been developed for utilization in the engineering evaluation of the sound pressure level performance of road vehicles in the vicinity of the exhaust systems. The purpose of this method is to check the vehicles that are in use and determine changes in noise level of exhaust system occurred due to wear and modification of certain components of the exhaust system or removing devices that are used for noise reduction. Basic requirements relating to measuring instruments, measurement conditions and measurement methodology of stationary vehicle noise are presented in this paper. Measurement procedure was demonstrated by measuring the stationary noise of vehicles of L i M category (two passenger cars and one motorcycle) using two different instruments, with the aim to indicate the possible uncertainty. Same vehicles have been tested on three technical inspection stations. Author's intention is to point out the failures that are happening when stationary vehicle noise is measured within the regular roadworthiness test in Bosnia and Herzegovina.

Key words: vehicle in use, stationary noise, measurement methodology, national regulations

BUKA VOZILA MERENA U BLIZINI IZDUVNOG SISTEMA

UDC: 534.6; 681.8

Rezime

Razvijena metodologija merenja stacionarne buke vozila je namenjena za ocenu nivoa zvučnog pritiska u blizini izduvne grane. Cilj rada je provera vozila u eksploataciji i određivanje promene bivoa buke izduvnog sistema koja nastaje kao posledica habanja i promena karakteristika pojedinih komponenata izduvnog sistema ili izgradnje elemenata koji su namenjeni snižavanju nivoa buke. Prikaz zahteva u pogledu poreme, uslova merenja i primenjene metodologije prikazani su u radi. Postupak merenja je demonstriran merenjem buke vozila kategorije L i M (dva putnička vozila i motorcikl) pomoću dva različita instrumenta kako bi se ukazalo na nesigrnost merenja. Neka vozila su ispitivana u tri stanice tehičkog pregleda. Namera autora je bila da ukaže na probleme koji se javljaju kod merenja stacionarne buke predvođenim standardom Bosne i Hercegovine

Ključne reči: korišćena vozila, stacionarna buka, metodologija merenja, nacionalna regulativa

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NOISE OF ROAD VEHICLES IN THE VICINITY OF THE EXHAUST SYSTEMS

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UDC:534.6; 681.8

1. INTRODUCTION

Approximately one in five Europeans are affected by road traffic noise, which is the most pervasive and widespread noise in towns and cities. Vehicle noise includes noise from exhaust systems, horns, brakes and sound systems. External vehicle noise originating from mechanical parts and exhaust system is standardized by the EU Directive 70/157/EEC, and it's amending 92/97 EEC, [1]. International regulations ECE R41 and ECE R51 and national regulations which are consistent with these rules, prescribe maximum permissible noise levels for each type of vehicle (e.g. 74 dB(A) for passenger cars and 80 dB(A) for motorcycles). Vehicle manufacturers are required to examine compliance of products and obtain ECE approval for vehicle type with regard to noise emission. The noise of the vehicle type submitted for approval should be measured by two methods described in regulations ECE R41 and ECE R51: for the vehicle in motion and for the vehicle when stationary (in the vicinity of the exhaust systems*)*, in accordance with ISO 362 and ISO 5130, [2]. A noise emitted by vehicle in motion is subjected to limitation prescribed by maximum permissible noise levels for each type of vehicle. A test made on a stationary vehicle provides a reference value for administrations which use this method to check vehicles in use. This reference noise value of stationary vehicle is not limited, according to regulations ECE R41 and ECE R51. However, levels of vehicles exhaust noise are prescribed by national regulation in some countries, for example in Australia, where authorised officers can issue penalty notices if noise limits are exceeded.

2. EXHAUST NOISE FROM VEHICLES

Regarding the noise of vehicles in use, regulations ECE R41 and ECE R51 prescribe that noise measurement procedure should be conducted according to the methodology of noise measurement for stationary vehicles. That sound pressure level measurement procedure has been developed for use in the engineering evaluation of the sound pressure level performance of road vehicles in the vicinity of the exhaust systems. The method is intended to check vehicles in use and also to determine variations in the exhaust sound pressure level that can result from:

- the wear, maladjustment or modification of particular components, when the defect does not appear by visual inspection;
- the partial or complete removal of devices reducing the emission of certain sound pressure levels.

The result of testing of the vehicle in use may be interpreted by comparison with the result of the reference test in which the vehicle was tested using the same method, for

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instance during type approval. The vehicles manufacturers are required to specify the stationary noise reference value and engine operating speed (rpm) which referring this level on the identification label placed on the vehicle. However, it was observed that the identification label containing reference noise level is set on only on vehicles of category L. For vehicles of categories M and N manufacturers do not set the identification label with reference noise level on the vehicle, but this information can be found in the approval certificate for the vehicle type (Certificate of Conformity), [3]. Figure 1 shows a part of EC Certificate of Conformity document for the vehicle of N3 category, containing information about sound level- stationary at engine speed - and drive-by sound level, inscribed in line number 46.

EC CERTIFICATE OF CONFORMITY
complete vehicles
MODEL A1 - SIDE 2
Vehicle Category N3

	Or maximum continuous rated power (electric motor) [kW]			N/A			
28.	Gearbox (type)	29.	Maximum speed [km/h]	Manual		90	
31.	Position of retractable axle(s)	32.	Position of loadable axle(s)	N/A		N/A	
33.	Drive axle(s) fitted with air suspension or equivalent			Yes			
35.	Tyre/wheel combination		Tyre-size (axle 1, 2)	295/80R22.5		295/80R22.5	
			Load Index, Speed symbol, Wheel-size (axle 1, 2)	152 M 8,25 ET157		148 M 8,25 ET157	
	Tyre/Wheel combination		Tyre size (axle 3, 4)	N/A		N/A	
			(load index, speed symbol, wheel size) (Axle 3,4)	N/A		N/A	
36.	Trailer brake connections (Mechanical / Electric / Pneumatic / Hydraulic)			Pneumatic / Electric			
37.	Pressure in feed line for trailer braking system [bar]			8.5			
38.	Code for bodywork			BC			
41.	Number and configuration of doors			2 doors, 1 left 1 right			
42.	Number of seating positions (including the driver)			2 seats, 1 driver and 1 co-driver			
44.	Approval number or approval mark of coupling device (if fitted)			Installation certificate e4*94/20*94/20*0613*			
45.1.	Characteristic values (D/V/S/U) [kN]			152	N/A	N/A	20
46.	Sound level [db(A)] - Stationary at engine speed [min-1] - Drive-by [db(A)]			91	1425	80	
47.	Exhaust emission level: Euro			Euro V			

***Figure 1**. A part of EC Certificate of Conformity containing information about sound level*

Within the regular roadworthiness in Bosnia and Herzegovina, the test of noise level of stationary vehicle should be done. Due to the lack of precise definition of certain norms in the national legislation of B&H relating to vehicle technical requirements, [4], it is noted that stationary noise of vehicle in use is compared with the permissible limits of noise emitted by motor vehicles prescribed by EU Directive 70/157 / EEC and regulations ECE R41 and ECE R51. This is definitely wrong, because these international regulations determine upper limits of exterior vehicle noise that vehicle shouldn't exceed in order to acquire ECE type approval, which is determined by the methodology of measuring noise for a vehicle in motion, according to ISO 362. In addition, the vehicle noise measured near the exhaust system is higher than noise emitted by the vehicle in motion. So, estimation of the stationary vehicle noise regarding to permissible upper limits of external vehicle noise definitely results that tested vehicle doesn't satisfy.

Some countries have normalized the upper limit of noise emitted near the exhaust system of vehicle in use (e.g. the Ministry of Infrastructure and Transport of Australia prescribes that the maximum permitted levels of noise an exhaust system for cars in use is 90 dB (A) and 94 dB (A) for motorcycles; this authority has conducted its own noise measuring and ensure systematized data base about the stationary noise level for each type of vehicle that passed the national certification and approval according their national regulations), [8]. In several countries, for example the Member States of the European Union and Norway, a system has been introduced such that the stationary level of noise (measured during type approval or when imported as a used vehicle) is labelled in the vehicle-registration documents, which are kept with the vehicle, [2]. This concept provides a more efficient basis for spot checks of the performance of vehicles using a stationary test. Comparison of results of the level of noise obtained from a roadside, or periodic technical inspection, to the baseline level of noise obtained during type approval gives a more accurate measure of the performance of any given vehicle.

It would be reasonable that vehicle inspection authorities in B&H prescribe measurement of vehicle noise only if information about reference noise level is labeled on the vehicle, as long as the corresponding database of stationary noise for different types of vehicles isn't formed. Almost all motorcycles manufactured after 1999 possess an identification label, while vehicles of categories M and N don't have it. If this label does not exist on the vehicle, the noise should not be measured because the measured result cannot be compared with anything. In this case, the controller can only have subjective judgment about the emitted noise.

3. REQUIREMENTS REGARDING EXHAUST NOISE MEASUREMENT

Vehicle noise measurement methodology is prescribed by ISO 362 and ISO 5130, as well as by ECE R51 and ECE R41 regulations. These regulations prescribe requirements to be fulfilled regarding measuring instruments, measurement conditions (appearance of a test site, weather conditions, preparing vehicles to be tested), and methods for vehicle noise testing (noise of the vehicle in motion and noise of the vehicle when stationary), [6, 7]. Specially, for measurement of stationary vehicle noise, that is exhaust noise, following requirements are to be meet:

- **Acustic measurement** is performed by the sound level meter meeting the requirements of Type 1 instrument in accordance with IEC 651. The measurement is done using the frequency weighting „ A “, and the time weighting „ F “. The maximum sound level expressed in „ A “weighted decibels must be measured during the operating period, L_{AFmax} dB(A). At the beginning and at the end of every measurement session the entire measurement system should be checked by means of a sound calibrator that fulfils the requirements for sound calibrators of at least precision Class 1 according to IEC 942:1988.
- **Test site** should be rectangular in shape with edges located at least 3 m from external dimensions of the vehicle. Test site surfaces should be dry, clean and covered with asphalt, concrete or some other hard material. Test site should be in an open space free from major obstacles and reflecting surfaces such as parked vehicles, buildings, billboards, trees and shrubs, or people within a radius of 3 m to microphone position and any point of the tested vehicles. During the test nobody shall be in the measurement area, except the observer and the driver whose presence must have no influence on the meter reading.
- **Meteorological conditions** during the measurements must be such as not to disturb the measurement result. Wind speed must not be greater than 5 m/s, and the air temperature should be $5^0C \div 45^0C$ according to R 41 ($0^0C \div 40^0C$ for R 51). It is

necessary to record the values of temperature, wind speed and direction, relative humidity and barometric pressure measured during the noise measurements.

- **Background noise** should be checked during the 10 s before and after testing vehicle noise. Measurement of background noise is made by the same microphone and at the same position as the measurement of vehicle noise. The maximum sound pressure level using the "A" weighting curve is to be measured. Background noise, including any wind noise, must be at least 10 dB(A) below the sound level produced by the vehicle.
- **Positioning and preparation of the vehicle**: The vehicle shall be located in the centre part of the test area with the gear level in neutral position and the clutch engaged, and the parking brake applied for safety. If the vehicle is fitted with fan(s) having an automatic actuating mechanism, this system shall not be interfered with during the sound level measurements. The engine hood or compartment cover shall be closed. Before each series of measurements, the engine must be brought to its normal operating condition, as specified by the manufacturer. In case of a two-wheeled motor-driven vehicle having no neutral gear position, measurements shall be carried out with the rear wheel raised off the ground so that the wheel can rotate freely.
- **Microphone orientation** is defined regarding the reference point of the exhaust pipe. The microphone shall be located at a distance of 0.5 m ± 0.01 m from the reference point of the exhaust pipe and at an angle of 45º (±5º) to the vertical plane containing the flow axis of the pipe termination. The microphone shall be at the height of the reference point, but not less than 0.2 m from the ground surface. The reference axis of the microphone shall lie in a plane parallel to the ground surface and shall be directed towards the reference point on the exhaust outlet.
- **Operating conditions of the engine** are defined by the target engine speed, as follows:

  A) for vehicles of L category**:**

  75 % of the engine speed n if n does not exceed 5,000 min^{-1};

  50 % of the engine speed n if n exceeds 5,000 min^{-1};

  B) for vehicles of M and N category:

  75 % of the engine speed n for vehicles with a rated engine speed ≤ 5,000 min^{-1};

  3,750 min^{-1} for vehicles with a rated engine speed above 5,000 min^{-1} and below 7,500 min^{-1};

  50 % of the engine speed n for vehicles with a rated engine speed ≥ 7,500 min^{-1};

  (n is the engine speed in min^{-1} (rpm) at which the engine develops its rated maximum net power).
- **Test procedure**: The engine speed shall be gradually increased from idle to the target engine speed, not exceeding the tolerance band of ±5 per cent of the target engine speed, and held constant. Then the throttle control shall be rapidly released and the engine speed shall be returned to idle. The sound pressure level shall be measured during a period consisting of constant engine speed of at least one second and throughout the entire deceleration period, Figure 2. The maximum sound level meter reading shall be taken as the test value. The test shall be repeated until three consecutive measurements at each outlet are obtained, which are within 2 dB of each other, allowing for deletion of non valid results.

- **Measurement result** is the arithmetic average of the three valid measurements, mathematically rounded to the first significant figure before the decimal place.

Due to the uncertainty influence, differences between the sound pressure level of the vehicle in-use and that in corresponding reference tests should not be considered significant unless they are equal to or larger than 5 dB.

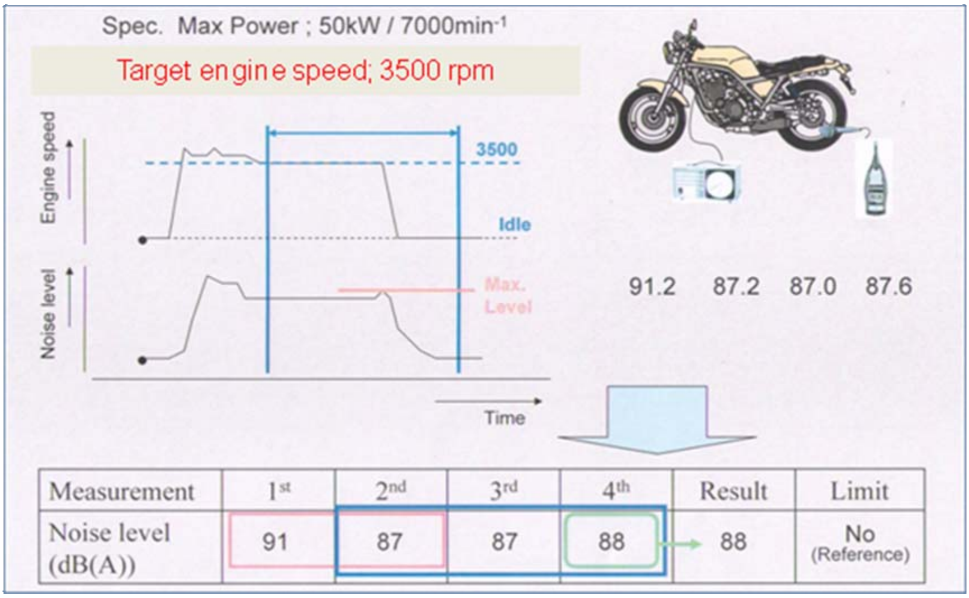

***Figure 2:** Representation of test procedure*

4. EXAMPLES OF EXHAUST NOISE MEASUREMENT OF ROAD VEHICLES

In order to show in detail the stationary noise measurement procedure for vehicles in use, noise measurement was performed for one vehicle of L category and two vehicles of M category:

1. Motorcycle model APRILIA, engine capacity 1000 cm^3,
2. Passenger car CITROEN, model C5, diesel engine 2000 cm^3, produced in 2005,
3. Passenger car CITROEN, model C1, petrol engine 1000 cm^3, produced in 2006.

Although identification label containing the reference value of the stationary noise exists only on motorcycle, noise measurement for vehicles of category M was carried out with the intention to evaluate the noise of exhaust system regarding the upper permissible limit prescribed by the applicable regulations in B&H.

Noise measurement was performed on 21st July 2012 at 9.30 pm, in the car park of the Faculty of Mechanical Engineering in Banja Luka that meets the requirements for the test site. Meteorological conditions were favorable, with air temperature of 25^{o}C and without wind.

Acoustical measurements was performed using the sound level meter type 2270, produced by Bruel&Kjaer. This instrument meets the requirements of Type 1 measurement instruments, as described on its identification label. The measurement was performed in the frequency range of 16 kHz, using the "A" frequency weighting and "F" time weighting. We checked the level of background noise, LAF_{max} = 46.1 dB (A), which was more than 10 dB (A) below the expected level of noise of tested vehicles.

4.1. Measurement of exhaust noise of motorcycle

Motorcycle posses the identification label with information about the reference stationary noise level, L = 92 dB(A), at engine speed n = 4100 min^{-1}, which is also the target engine speed to be achieved during measuring procedure, Figure 3a.

Motorcycle was placed in the proper position at the test site and approximate microphone position was marked: at the distance about 500 mm from the exhaust pipe, at an angle of 45^o with respect to the axis of the exhaust pipe, the height of the microphone matches the height of the exhaust pipe, Figure 3b. Motorcycle possesses two exhaust pipes installed at the horizontal distance of 280 mm to each other, so it was sufficient to measure the noise in one measurement position.

Once the engine reached operating temperature of T = 80^oC, noise measurement was performed with a motorcycle operating in a neutral gear position whith the engine speed of n = 1350 min^{-1}. After this, measurement of motorcycle noise has been done at the target engine speed of n = 4100 min^{-1} and the engine operating temperature T = 95^oC. The measurement was repeated three times, and the results did not differ by more than 2 dB (A). Measurement results are given in Table 1. Stationary noise level of a motorcycle is defined as the arithmetic average of the measured values rounded to the first integer value, L_{AFmax} = 96 dB (A).

Figure 3: *a) Identification label, b) motorcycle and microphone position at the test site*

Due to the uncertainty influence, differences between the sound pressure level of the vehicle in-use and that in corresponding reference tests should not be considered significant unless they are equal to or larger than 3 dB according to corresponding regulations in B&H, [3]. The measured noise level of motorcycle exceeds the reference value for 4 dB (A), which means that tested motorcycle does not satisfy in terms of noise emission. The owner of the motorcycle made certain changes to the exhaust system, which resulted in an increased noise level.

Table 1: *Motorcycle noise in the vicinity of the exhaust systems*

Motorcycle APRILIA	Sound level meter	Measured noise level L_{AFmax} dB(A)		
		1.measurement	2. measurement	3. measurement
Engine speed n=1350 min−1	B&K Type 2270	90,6	90,7	89,6
Target engine speed n=4100 min−1	B&K Type 2270	**96,2**	**96,5**	**96,3**
Noise level of Motorcycle APRILIA: L_{AFmax}= 96 dB(A)				

4.2. Measurement of exhaust noise of passenger cars

Passenger cars were successively placed into the measurement position on test site and position of microphone was determined: the distance about 500 mm from the exhaust pipe, at an angle of 45^0 with respect to the axis of exhaust pipe, the height of the microphone matches the height of the exhaust pipe at the distance of 230 mm from asphalt surface, Figure 4.

The background noise level was L_{AFmax} = 51.9 dB (A), which is appropriate for measurement of vehicle noise. For CITROEN C5, noise was measured for the engine speed of n = 900 min^{-1} in idling position of vehicle, and for the target engine speed of n = 3750 min^{-1}, while for CITROEN C1 noise was measured only for the target engine speed of n = 3750 min^{-1}. Measurement results are given in Tables 2 and 3.

Table 2: *Noise in the vicinity of the exhaust systems of the vehicle CITROEN C5*

CITROEN C5	Sound level meter	Measured noise level L_{AFmax} dB(A)		
		1.measurement	2. measurement	3. measurement
Engine speed n=900 min−1	B&K Type 2270	71,4	70,3	68,3
Target engine speed n=3750 min−1	B&K Type 2270	**84,2**	**83,2**	**84**
Noise level of CITROEN C5: L_{AFmax}= 84 dB(A)				

Figure 4: *Position of cars at the test site and a microphone position*

Table 3: *Noise in the vicinity of the exhaust systems of the vehicle CITROEN C1*

CITROEN C1	Sound level meter	Measured noise level L_{AFmax} dB(A)		
		1.measurement	2. measurement	3. measurement
Target engine speed n=3750 min−1	B&K Tip 2270	**77,4**	**77,3**	**76,8**
Noise level of CITROEN C1: L_{AFmax}= 77 dB(A)				

Tested vehicles of M category are produced in 2005 and 2006, and there are no obvious defect in the engine and exhaust system operation. Measurement results obtained for these vehicles exceed the upper limit of permissible vehicle noise prescribed in ECE R 41 and R 51 regulations (74 dB(A) is limit for passenger cars). These measurement results confirm that the noise level of stationary vehicle in the vicinity of an exhaust system is greater than the level of the external noise of a moving vehicle, where upper limits are prescribed in R 41 and R51 regulations. It is wrong to compare the stationary vehicle noise near the exhaust system with permissible values of vehicle noise prescribed in standards R41 and R51. Assessment of the vehicle in use regarding noise emission should be conducted only in relation to the reference value of the stationary noise, and only if there is an identification label containing these data marked on vehicle.

4.3. Measurement of vehicle exhaust noise in vehicle inspection stations

Same vehicles have been tested in several technical inspection stations, in order to check if technical controllers implemented the prescribed measurement methodology in a good manner.

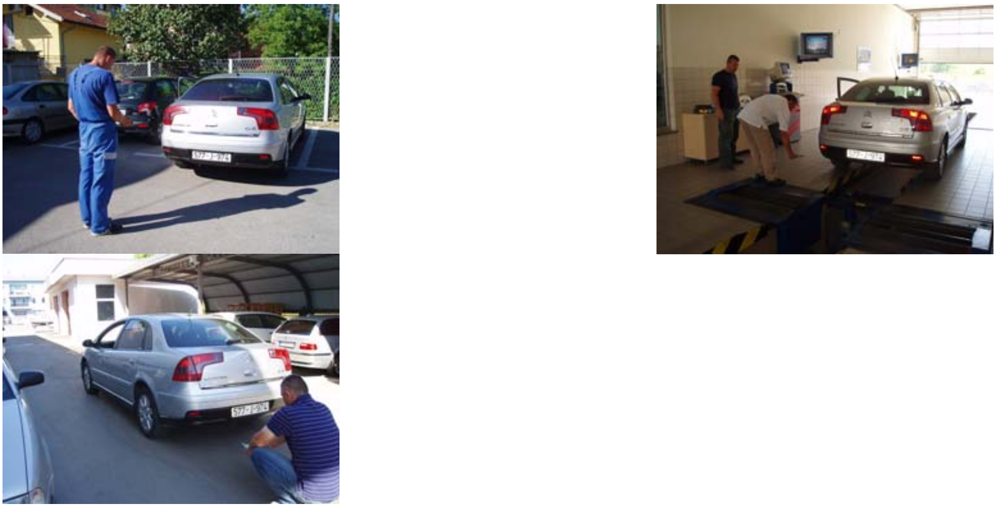

Figure 5: *Measurement of noise in vehicle inspection stations*

Some failures in the performance of measurement are noted, Figure 5:

a) minimum distance of 3 m between the vehicle and the surrounding obstacles is not provided;
b) measurement is performed indoors at some inspection stations;
c) Sound Level Meter doesn't satisfy the required class I accuracy
d) " S " time weighting is used instead " F " time weighting;
e) the background noise is not checked;
f) the vehicle noise is not measured for the target engine speed;
g) the sound pressure level is not measured during a period consisting of constant engine speed of at least one second and throughout the entire deceleration period
h) some controllers do not take the prescribed position of the microphone in relation to the exhaust pipe, etc.

Results of noise measurements performed at vehicle inspection stations are given in Table 4. These results are compared to results of measurements previously done at the Faculty of Mechanical Engineering by following the prescribed measurement methodology and using the sound level meter 2270, Bruel&Kjaer. Some of measurement results are approximate to reference values, but are contaminated due to failures occurred during the measurement.

Table 4: *Vehicle noise measured at target engine speed*

Vehicle	Measured noise level, L dB(A)			Noise level measured at Faculty of Mechanical Engineering (reference value)
	1. insp. station	2. insp. station	3. insp. station	
	Fonometer Mar. - Hofman Voltcraft	Sound level meter Luton SL - 4001	Sound level meter Luton SL - 4001	
Citroen C1	$L_{ASmax} = 68$	$L_{AFmax} = 77$	$L_{AFmax} = 78{,}9$	$L_{AFmax} = 77$
Citroen C5	$L_{ASmax} = 75$	$L_{AFmax} = 84$	$L_{AFmax} = 82{,}3$	$L_{AFmax} = 84$
Motorcycle APRILIA	$L_{ASmax} = 96$	$L_{AFmax} = 100$	$L_{AFmax} = 107$	$L_{AFmax} = 96$

It should be recognized that variations in measured noise levels can occur due to variations in test sites, atmospheric conditions and test equipment. So, it is essential that persons technically trained and experienced in current sound measurement techniques select the test instrumentation and conduct the test.

5. CONCLUSIONS

Measurement methodology for stationary vehicle noise has been developed for use in the engineering evaluation of the sound pressure level performance of road vehicles in the vicinity of the exhaust systems. The purpose of this method is to check the vehicles that are in use and determine changes in noise level of exhaust system occurred due to wearing and modification of certain components of the exhaust system or removing the devices that are used for noise reduction. Condition of the tested vehicle in terms of noise emission can be established by comparing the measured noise levels with the reference noise values measured under similar conditions, for example during the vehicle type approval procedures. To correctly estimate a vehicle in use against emitted noise, it is essential that information about reference level of stationary noise is available, that is indicated on the identification label on the vehicle. Vehicle noise measurement procedure presented in this paper has shown that the level of stationary noise measured near the vehicle exhaust system is higher than the upper limit of permissible vehicle noise prescribed in ECE R 41 and R 51 regulations. It is wrong to compare the measured stationary vehicle noise against upper limits of permissible exterior vehicle noise. Also, in order to accurately assess the vehicle condition against emitted noise, it is necessary to consistently comply with the prescribed measurement methodology. It is noted that controllers at vehicle inspection stations does not fully comply the prescribed noise measurement methodology, and measurement results obtained in this way are not a reliable indicator of vehicle condition and cannot be used for vehicle estimation (satisfy / not satisfy).

6. REFERENCES

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[7] Regulation No. 51 - Uniform provisions concerning the approval of motor vehicles having at least four wheels with regard to their noise emissions

[8] Vehicle Standard (Australian Design Rule 28/01 — External Noise of Motor Vehicles), 2006.

[1]AN ANALYSIS OF VEHICLE SEMISHAFT LOADING WHEN THE WHEEL PASSES OVER SUCCESSIVE ROAD IRREGULARITIES

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Milanko Damjanović, assistant

UDC: 629.07

Summary

The paper presents an analytical and experimental analysis of the vehicle semishaft loading when the wheel passes over successive road surface irregularities with defined geometry. It has been found that the intensity of the dynamic loading considerably exceeds values of the loading generated in the case when the wheel passes over a flat surface. A mathematical model of the dynamics of the elastic suspension-wheel-base system is presented along with the mathematical model used for computer simulation. Verification of this approach is performed by comparing simulation results with the results obtained experimentally. In that way the usability of the established mathematical model is confirmed. Such models are the basis for determining the influence of road irregularities on vehicle transmission elements loading and service life. Also, the models enable an analysis of the interaction between individual systems of the vehicle.

Key words: vehicle, irregularity, mathematical model, simulation

ANALIZA OPTEREĆENJA POLUVRATILA VOZILA PRI PRELASKU TOČKA PREKO UZASTOPNIH NERAVNINA PODLOGE

Rezime

U radu je data analitička i eksperimentalna analiza opterećenja poluvratila vozila koja nastaje pri prelazu točka preko uzastopnih neravnina podloge definisane geometrije. Utvrđeno je da intenzitet opterećenja značajno premašuje vrijednosti opterećenja za slučaj kretanja točka preko ravne podloge. Prikazan je matematički model dinamike sistema elastično oslanjanje-točak-podloga i matematički model korišćen za računarsku simulaciju. Verifikacija ovog pristupa je provedena upoređenjem rezultata simulacije sa rezultatima eksperimenta. Na taj način je potvrđena upotrebna vrijednost postavljenog matematičkog modela. Ovakvi modeli predstavljaju osnovu za određivanje uticaja neravnina podloge na opterećenja i radni vijek elemenata pogonskog sistema. Takođe, modeli omogućavaju i analizu međusobnog uticaja pojedinih sistema vozila.

Ključne riječi: vozilo, neravnina, matematički model, simulacija

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AN ANALYSIS OF VEHICLE SEMISHAFT LOADING WHEN THE WHEEL PASSES OVER SUCCESSIVE ROAD IRREGULARITIES

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UDC:629.07

1. INTRODUCTION

A vehicle as a complex system is the subject of continuing research with view to improve its performance. In this sense, there are ongoing efforts to make the selection and adjustment of the vehicle system to exploitation conditions dictated by its purpose.

Continual analyses and experiments are conducted which, when the research topics are concerned, to a certain extent neglect some aspects of the vehicle use. For instance, a relatively neglected aspect is the analysis of certain influences on vehicle transmission loading. Currently are these influences taken into account through theoretically defined calculations in certain vehicle operating regimes and extreme conditions that may occur in the vehicle exploitation.

The analysis is usually performed by applying a mathematical model of a certain complexity neither the accuracy nor the reliability of which is determined [2]. Basically, it is an analysis of the vehicle loading in normal operation, i.e., when a vehicle passes over a relatively flat surface, while in special cases a vehicle passing over curbs or dents on the road surface is analyzed. In doing so, a number of models is used, some of which are commercial in nature, but it is notable that this field of analysis requires a significant effort that will lead to the establishment of a model, which will introduce, describe and provide an analysis of the drivetrain or any other vehicle system under the influence of road irregularities in the most reliable way. For this purpose, it is also necessary to make a great number of experimental studies and link them with the theory in a clear way.

2. THE INFLUENCE OF ROAD IRREGULARITIES ON VEHICLE BEHAVIOUR

By observing the values of the dynamic forces and moments generated when a vehicle passes over road surface irregularities it can be clearly stated that in all cases their value passes through an intensive change, Figure 1. It is confirmed by both observing the forces generated at the point of contact between the vehicle and the road surface, the load of each element due to the effect of these forces, the moments on some of the elements of the drivertrain or the position change generated within the suspension system.

Monitoring of this change is often the subject of analyses and development, on the basis of which are certain conclusions obtained. However, it must be noted that many of these conclusions are followed by statements which highlight some shortcomings of the approaches applied. The most common drawbacks in this respect are a large number of adopted simplifications and the lack of comparison and quantification of errors incurred by applying certain approaches. This is especially true when one takes into account the

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aforementioned simplifications, which will inevitably lead to the question of how the accuracy of a certain approach is reduced by the simplifications adopted, how certain parameters affect the accuracy, that is, whether the adopted simplifications are justified in terms of a small impact on the results achieved.

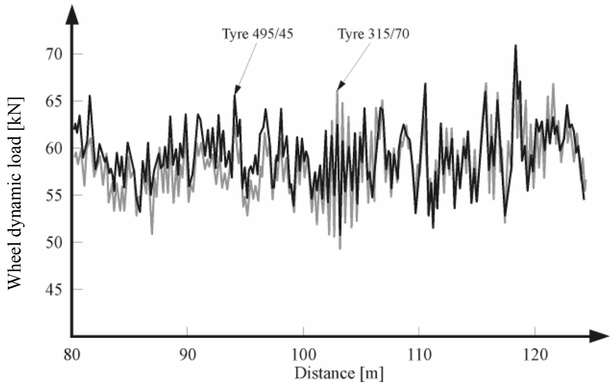

Figure 1: *Wheel dynamic load at rolling along flat road surface*

In this regard, analyses of errors generated were carried out by comparing the results of mathematical models with the experimental data, but again with a disadvantage that has to be mentioned and is reflected in construction of experimental installations which do not correspond to the real conditions of the vehicle use. This deficiency can be explained as mostly due to the impossibility of constructing experimental installations with real vehicles, mainly due to technical and financial constraints in implementing the experiment.

The aforementioned limitations are particularly pronounced when one takes into account the need of performing generalized conclusions, which in itself generates the need for conducting a large number of experiments in different conditions, for example, at different values of the vehicle speed, road surface conditions, types or characteristics of certain vehicle systems.

In this sense we can argue that a rational approach entails gradually collecting data and setting individual conclusions for the observed specific conditions of the vehicle use while using the data obtained under the relatively same conditions in which the vehicle testing was carried out. After this, the approach is expanded. The number of observed parameters and influences increases. The new findings are given and some of the parameters may be ignored when their limited impact is stated.

The authors of this paper approached the analysis in this very way, ie. in the way that the analysis is done gradually, step by step, with increasing gradually the complexity of the model to the point when it is possible to set a valid conclusion. Thus, an analysis of the dynamic behavior of the system under the influence of an individual irregularity has been performed [4], then an analysis of the dynamic behavior of the vehicle's drivetrain under the influence of an individual irregularity [3], [5]. The next step in the approach is an analysis presented in this paper, which aims to explore the validity of the models set in the case of a vehicle passing over a series of clearly defined irregularities. As in the papers mentioned in order to establish certain conclusions the established models should be tested in such a way that the results will again, be compared to those obtained experimentally, that is, obtained by recording on a real vehicle that passes over a clearly defined irregularity.

As the starting point of this analysis is identical to those of the analysis in the abovementioned papers, the conclusions based on them can be mentioned as a starting point of the analysis. In that sense it is not necessary to make any comparison or experimental monitoring of the values of the vertical and horizontal forces generated, but the behavior of the suspension systems will be tested, that is, in addition to the value of the load torque change on the vehicle semi-shaft a comparison will also be performed with recorded values of position change in the suspension system, that is in the shock absorber-spring system.

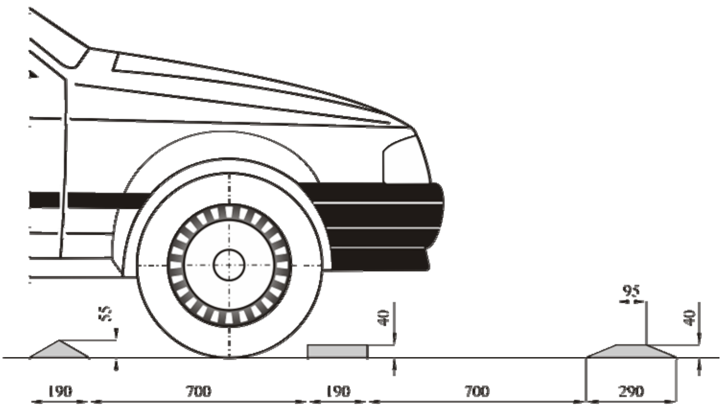

Figure 2: *Experimental scheme*

3. A MATHEMATICAL MODEL OF INFLUENCE OF ROAD IRREGULARITIES ON VEHICLE

For this analysis mathematical models set in the already defined literature sources will be used, and for the vehicle's drivetrain they are built on the basis of the scheme of the drive wheel load, figures 3 and 4.

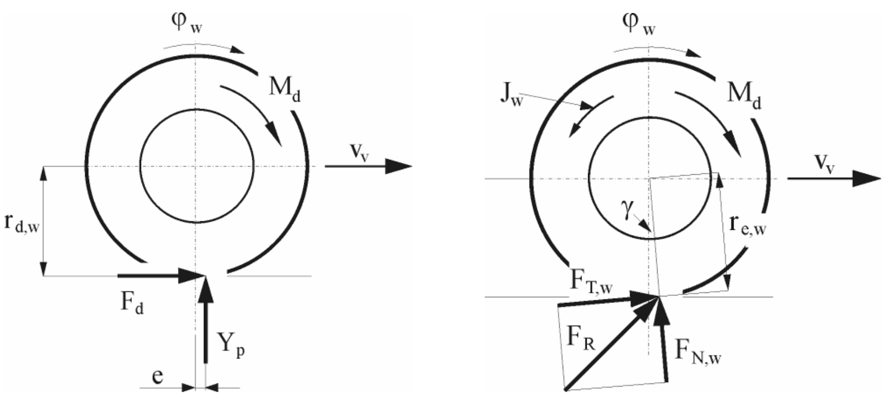

Figure 3: *Loading of vehicle wheel*

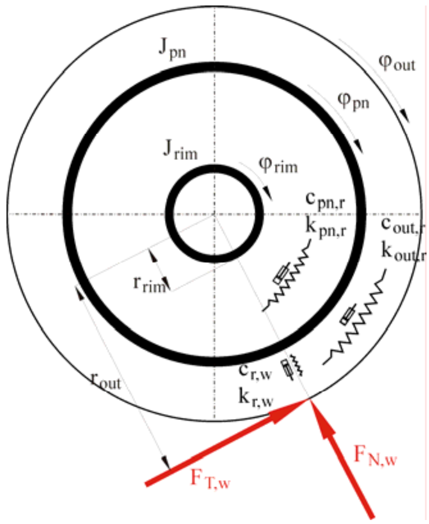

Figure 4: *Wheel represented as two mass dynamical system*

The simulation of the dynamic behavior of the vehicle system is performed according to the linearized mathematical models, as follows:

- drive train:

a. drive engine:

$$\Delta M_e = \frac{\partial M_e}{\partial \omega_e} \cdot \Delta\omega_e + \frac{\partial M_e}{\partial h} \cdot \Delta h$$

$$J_e \cdot \Delta\ddot{\varphi}_e = \frac{\partial M_e}{\partial \omega_e} \cdot \Delta\omega_e - [c_e \cdot (\Delta\varphi_e - \Delta\varphi_{out,e}) + k_e \cdot (\Delta\dot{\varphi}_e - \Delta\dot{\varphi}_{out,e})]$$

where c_e and k_e - are equivalent stiffness and damping of the engine, J_e - moment of inertia of the engine, φ_e - angle of rotation of the motor shaft, $\varphi_{out,e}$ - angle of rotation of the motor output shaft, M_e - torque of the engine,

b. gearbox:

$$\left(J_1 + \frac{J_2}{i_g^2}\right) \cdot \ddot{\varphi}_g = M_{p,g} - \frac{1}{i_g} \cdot \left[c_g \cdot \left(\frac{\varphi_g}{i_g} - \varphi_{out,g}\right) + k_g \cdot \left(\frac{\dot{\varphi}_g}{i_g} - \dot{\varphi}_{out,g}\right)\right]$$

where c_g and k_g - are equivalent stiffness and damping of the gearbox, $J_{1,2}$ - moments of inertia of the gearbox, φ_g - the angle of rotation of the input gearbox shaft, $\varphi_{out,g}$ - the rotation angle of the output transmission shaft,

c. semishaft:

$$J_s \cdot \frac{d^2\varphi_s}{dt^2} = M_{p,s} - [c_s \cdot (\varphi_s - \varphi_{out,s}) + k_s \cdot (\dot{\varphi}_s - \dot{\varphi}_{out,s})]$$

where c_s and k_s - are equivalent stiffness and damping of the semishaft, J_s - moment of inertia of the semishaft, φ_s - the rotation angle of the semishaft,

d. differential:

$$\left[J_1 + \frac{J_2}{i_0^2} + \frac{J_3}{\left(i_{p(l)} \cdot i_0\right)^2} + \frac{J_4}{\left(i_{z(d)} \cdot i_0\right)^2}\right] \cdot \ddot{\varphi}_d = M_{p,d} - \left[M_{res,f(l)} + M_{res,r(r)}\right]$$

$$M_{re,f(l)} = \frac{1}{i_0 \cdot i_{f(l)}} \cdot \left[c_{out,f(l)} \cdot \left(\frac{\varphi_d}{i_0 \cdot i_{f(l)}} - \varphi_{out,f(l)}\right) + k_{out,f(l)} \cdot \left(\frac{\dot{\varphi}_d}{i_0 \cdot i_{f(l)}} - \dot{\varphi}_{out,f(l)}\right)\right]$$

$$M_{res,r(r)} = \frac{1}{i_0 \cdot i_{r(r)}} \cdot \left[c_{out,r(r)} \cdot \left(\frac{\varphi_d}{i_0 \cdot i_{r(r)}} - \varphi_{out,r(r)}\right) + k_{out,r(r)} \cdot \left(\frac{\dot{\varphi}_d}{i_0 \cdot i_{r(r)}} - \dot{\varphi}_{out,r(r)}\right)\right]$$

where $i_{0,f(l),r(r)}$ - gear ratios in the differential, $c_{out,f(l),r(r)}$ and $k_{out,f(l),r(r)}$ - the equivalent stiffness and damping of the differential shaft, $J_{1,2,3,4}$ - moments of inertia of the differential, φ_d - the rotation angle of the differential input shaft, $\varphi_{out,f(l),r(r)}$ - the rotation angle of the differential output shafts, $M_{res,f(l),r(r)}$ - moment of resistance of the differential output shafts,

e. drive wheel:

$$J_{rim} \cdot \ddot{\varphi}_{rim} = M_{p,w} - \left[c_{pn,r} \cdot \left(\varphi_{rim} - \varphi_{pn}\right) + k_{pn,r} \cdot \left(\dot{\varphi}_{rim} - \dot{\varphi}_{pn}\right)\right]$$

$$J_{pn} \cdot \ddot{\varphi}_{pn} = \left[c_{pn,r} \cdot \left(\varphi_{rim} - \varphi_{pn}\right) + k_{pn,r} \cdot \left(\dot{\varphi}_{rim} - \dot{\varphi}_{pn}\right)\right] - M_{res}$$

$$M_{res} = F_{pn,z,ekv} \cdot e_{pn,ekv} \quad , \quad e_{pn,ekv,ul} = r_{out} \cdot \frac{z_{p,t,peg}}{s}$$

$$\ddot{e}_{pn,ekv} + A \cdot B \cdot \dot{e}_{pn,ekv} + A^2 \cdot e_{pn,ekv} = A^2 \cdot C \cdot e_{pn,ekv,ul} \,,$$

$$e_{pn,ekv} = e_{pn,ekv,0} \quad za \quad z_{p,t,peg} = 0$$

where $c_{pn,r}$ and $k_{pn,r}$ are tangential stiffness and damping of the tyre, $J_{rim,pn}$ - moments of inertia of the wheel rim and tyre, $\varphi_{rim,pn}$ - the rotation angle of the wheel rim and tyre, M_{res} - moment of resistance, $F_{pn,z,ekv}$ - equivalent vertical force on tyre, $e_{pn,ekv}$ - arm of the vertical force, r_{out} - outer radius of the tyre, $z_{pt,peg}$ - effective value of irregularity height, A, B, C - constants,

- the suspension system (suspended and unsuspended mass):

$$m_{sus} \cdot \ddot{z}_{sus} = c_s \cdot (z_{nsus} - z_{sus}) + k_s \cdot (\dot{z}_{nsus} - \dot{z}_{sus})$$

$$m_{nsus} \cdot \ddot{z}_{nsus} = \left[c_{r,w} \cdot \left(z_{p,t} - z_{nsus}\right) + k_{r,w} \cdot \left(\dot{z}_{p,t} - \dot{z}_{nsus}\right)\right] -$$

$$- \left[c_s \cdot (z_{nsus} - z_{sus}) + k_s \cdot (\dot{z}_{nsus} - \dot{z}_{sus})\right] \cdot \cos\delta \cdot \frac{a}{b}$$

$$F_{pn,z,ekv} = c_{r,w} \cdot \left[z_{p,t} - z_{nsus}\right] + k_{r,w} \cdot \left[\dot{z}_{p,t} - \dot{z}_{nsus}\right]$$

$$F_{pn,z} = c_{r,w} \cdot \left[z_{p,t,peg} - z_{nsus}\right] + k_{r,w} \cdot \left[\dot{z}_{p,t,peg} - \dot{z}_{nsus}\right]$$

where a and b are constants, δ - angle in the suspension system, c_s and k_s - equivalent stiffness and damping of the suspension system, m_{sus} and m_{nsus} - suspended and unsuspended mass, $c_{r,w}$ and $k_{r,w}$ - radial tyre stiffness and damping, z_{sus} and z_{nsus} - the position of the suspended and unsuspended mass, $F_{pn,z}$ and $F_{pn,z,ekv}$ - the vertical and equivalent vertical force on the wheel,

- road irregularities:

As the wheel is a deformable element of the system simultaneously in contact with a number of irregularities, it is necessary to take into account the impact of the deformable tyre that is reflected in smoothing irregularities. This characteristic of the tyre is given in the form of transfer function of the dynamic chain in which the input value is the ordinate of the road surface micro-profile and the output obtained is its mean height at the length of the contact zone between the tyre and the road surface, [1]:

$$W_{peg}(j\omega)=\frac{k_{pl}^2}{(j\omega)^2+j\omega\cdot k_{pl}\cdot\sqrt{2}+k_{pl}^2}$$

where k_{pl} is a coefficient which is calculated by the expression:

$$k_{pl}=(0.9\div 1.3)\cdot\frac{v}{l_t}$$

where l_t is the length of the contact between the raod surface and tyre, and v is the speed of the vehicle.

The parameter l_t is obtained by using the expression:

$$l_t=2\cdot\sqrt{0.1\cdot H_t\cdot(D_t-0.1\cdot H_t)}$$

where D_t and H_t are the outer radius and the height of the tyre profile.

The spectral density of the process smoothed by the tyre S_{peg} is obtained from the transfer function:

$$S_{peg}(\omega)=W_{peg}(j\omega)\cdot W_{peg}(-j\omega)\cdot S_h(\omega)=\frac{k_{pl}^2}{\left(k_{pl}^2-\omega^2\right)^2+2\cdot k_{pl}^2\cdot\omega^2}\cdot S_x(\omega)$$

where S_h is the spectral density of the microprofile.

Based on the transfer function shown above the following differential equation is obtained on the basis of which is performed the calculation of the equivalent height of an irregularity relevant for the calculation of the behavior of the system under the influence of the irregularities:

$$\ddot{z}_{p,t,peg}+\dot{z}_{p,t,peg}\cdot k_{pl}\cdot\sqrt{2}+z_{p,t,peg}\cdot k_{pl}^2=k_{pl}^2\cdot h_{mp}$$

where h_{mp} is an accidental height of the irregularity.

Applying the previously described procedure for determining the height of the relevant irregularity the characteristic of the effective irregularity shown in the diagram given in Figure 5 is obtained.

4. SIMULATION OF TORSION MOMENT ON SEMISHAFT AND COMPARISON WITH EXPERIMENTAL RESULTS

A computer simulation of the dynamic behavior of the system, using the above expressions, is performed for the connected vehicle systems which are schematically shown in Figure 6.

The simulation results are compared with the results of experimental tests performed on the vehicle installation and the test track harmonized with the scheme shown in Figure 2; these results are obtained for the vehicle speed of 5.5 m/s. Some of the elements of the test installation are shown in the Figures 7 and 8.

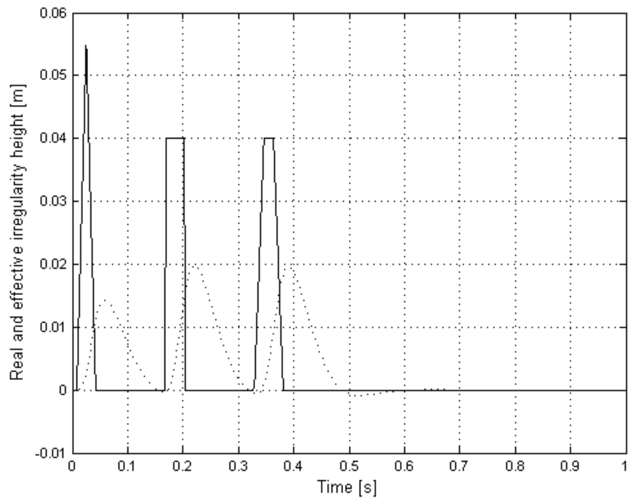

Figure 5: *The real and the effective irregularity height (real irregularity-solid line, effective irregularity-dotted line)*

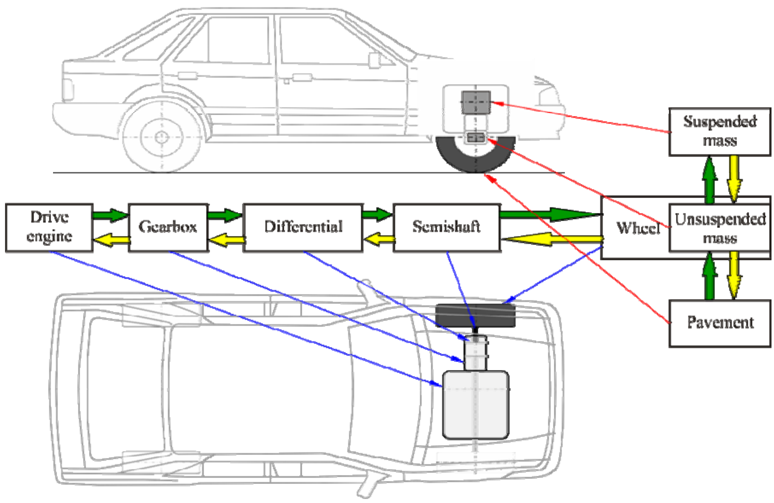

Figure 6: *Scheme of vehicle systems*

Figure 7: *Experimental installation (sensors)*

Figure 8. *Experimental installation (test track and vehicle)*

When performing the experiment, the diagrams of the changes of the relative position in the spring-damper system as well as the diagram of the torsion moment changes at the front left drive wheel semishaft are obtained.

After performing the computer simulation of the dynamic behavior of the suspension systems and drivetrain when a vehicle passes over a series of irregularities shown in Figures 2 and 8, its results are shown in the same diagram along with the results of experimental tests. The relative position change in the spring-damper system diagram is shown in Figure 9 and a torsion moment change on semishaft diagram is shown in Figure 10.

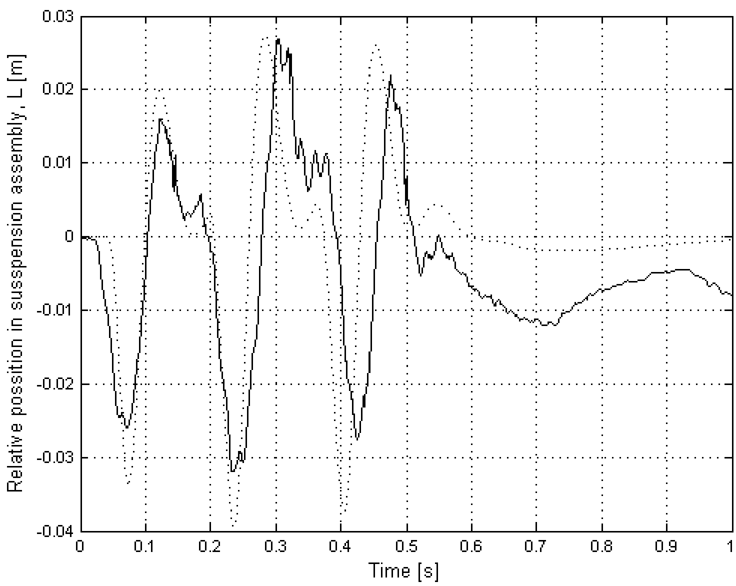

Figure 9: *Change of the relative position of the suspension assembly (experiment-solid line, simulation-dotted line)*

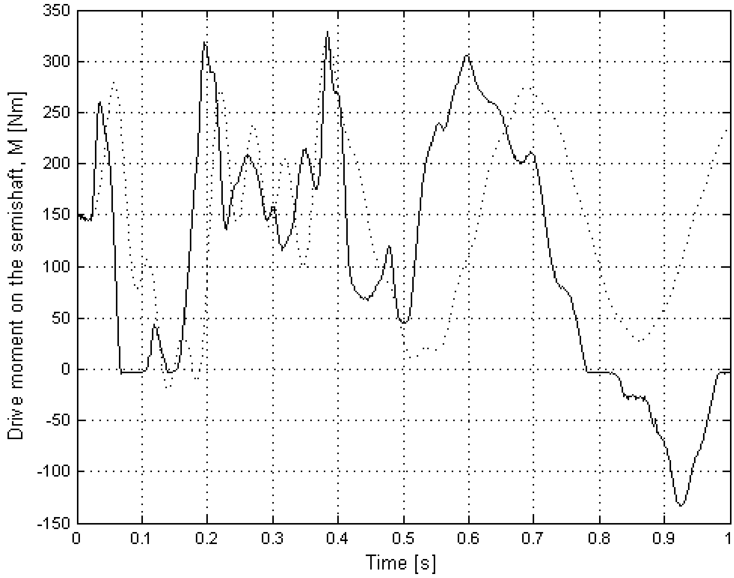

Figure 10: *Change of the drive moment on the semishaft (experiment-solid line, simulation-dotted line)*

5. CONCLUSION

The results show the complexity of the dynamic behavior of the suspension system and drivetrain. By comparing the simulation results with the experimental results it can be concluded that there is good agreement between them, which is important especially if one takes into account the relative simplicity of the model.

The approach enables further analysis that will give valid results, based on the given conclusions, for example, in the service life analysis of individual elements of the vehicle system, which is the analysis given in the dissertation of the first author.

Nevertheless, this analysis is only a step up compared to the previous analyses, and it raises a number of questions which point to the need of analysis of other influences which surface irregularities exert on certain vehicle systems. Therefore, it is necessary to continue research on this subject, for instance, by conducting additional experiments, extending the model with new parameters, studies of the impact of the vehicle speed and the like.

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