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CONTRIBUTIONS IN DEVELOPING ADVANCED ENGINEERING METHODS FOR RESEARCHING THE N.V.H. ASPECTS IN PORSCHE CAYMAN OPERATION CONDITIONS AT TECHNICAL UNIVERSITY OF CLUJ-NAPOCA

Doru-Laurean BĂLDEAN *, Bogdan-Ovidiu VARGA, Dan MOLDOVANU, Levente KOCSIS, Adela-Ioana BORZAN

Technical University of Cluj-Napoca, B-dul Muncii, Nr. 103-105, 400641, Cluj-Napoca, Romania

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Abstract: *Noise is one of the most important parameter that influences the artificial and especially automotive pollution. Developing new motor vehicles, widely known as cars, respectively their auxiliary systems is now difficult to be done without considering noise, vibration and harshness (N.V.H.) aspects in testing and operation. N.V.H. criteria facilitate the appreciation made by testers, owners and users in relation with vehicle's general behaviour and also in correlation with their auxiliary systems. Main objective of the present paper is to outline the innovative and advanced method in computer aided testing of N.V.H. parameters in a sporty car Porsche Cayman model at Technical University from Cluj-Napoca. Redefining the standards in comfort and tuning is close related to the N.V.H. research. These parameters are not acting as isolated factors in some isolated environment. They should be considered in relation with the complex whole which is the operating motor vehicle in the road traffic environment. Experimental testing results were achieved and properly analysed and interpreted. There were outlined some trend lines in each case. Development possibilities and perspectives were also pointed out in order to facilitate and encourage the further research of the specific problems.*

Keywords: *noise, vibration and harshness tests, comfort, Porsche, Cayman*

1. INTRODUCTION

Considering noise, vibration and harshness (N.V.H.) aspects we aim to achieve a specific level of testing and to improve measuring capabilities as well as to develop some innovative and advanced research methods. Increased expectations and regulations concerning noise, vibration and harshness (NVH) levels are inherent in automotive engineering and specific design. Optimal refinement is in this case one of the significant technological and design attributes to be looked at in the process of perfecting a motor vehicle and its specialized systems. Noise emissions of traction drives and the coupling of structure-borne sound to surrounding technical devices is increasingly becoming a key performance indicator (KPI) in the automotive industry and is also used as an indicator for condition monitoring e.g. of production machines 0. The present day automotive industry searches for any option to attract the customers with products which will be generating lower noise and consume less fuel but also be equally powerful. This unique requirement drives the motivation for building each and every aspect of a dynamical system to be represented in the math model. The car should be durable, less noisy, powerful, as well as elegant 0. In order to develop new vehicle products with well-refined noise performance, vehicle noise measurements and analysis have to be conducted to validate designs and acoustic refinement 0. High standards of NVH (Noise, Vibration and Harshness) performance are expected by consumers of all modern cars. Refinement is one of the main engineering and design attributes to be addressed in the course of developing new vehicle models and vehicle components 0. By applying a specific methodology and materials the paper aims to achieve experimental results.

2. METHODOLOGY AND MATERIALS

The best results in research are gained by applying scientific method and using adeqauted materials as it is shown in the next paragraphs.

* *Corresponding author e-mail: doru.baldean@auto.utcluj.ro*

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Environmental requirements of the recording equipment: operating ambient temperature 0° to 35° C; nonoperating temperature -20° to 45° C; relative humidity: 5% to 95% noncondensing; operating altitude up to 3000 m.
Methodology was developed at Technical University from Cluj-Napoca and materials were made available at the Automobile Laboratory from Automotive and Transportations Department.

2.1. Methodology

By measuring the noise level, considering the vibration critical range and analyzing harshness we can assess and evaluate the comfort level. The present paper shows an engineering method (Figure 1) for computer aided testing and appreciation in the field of N.V.H. parameters concerning the operation of sporty Porsche Cayman car.

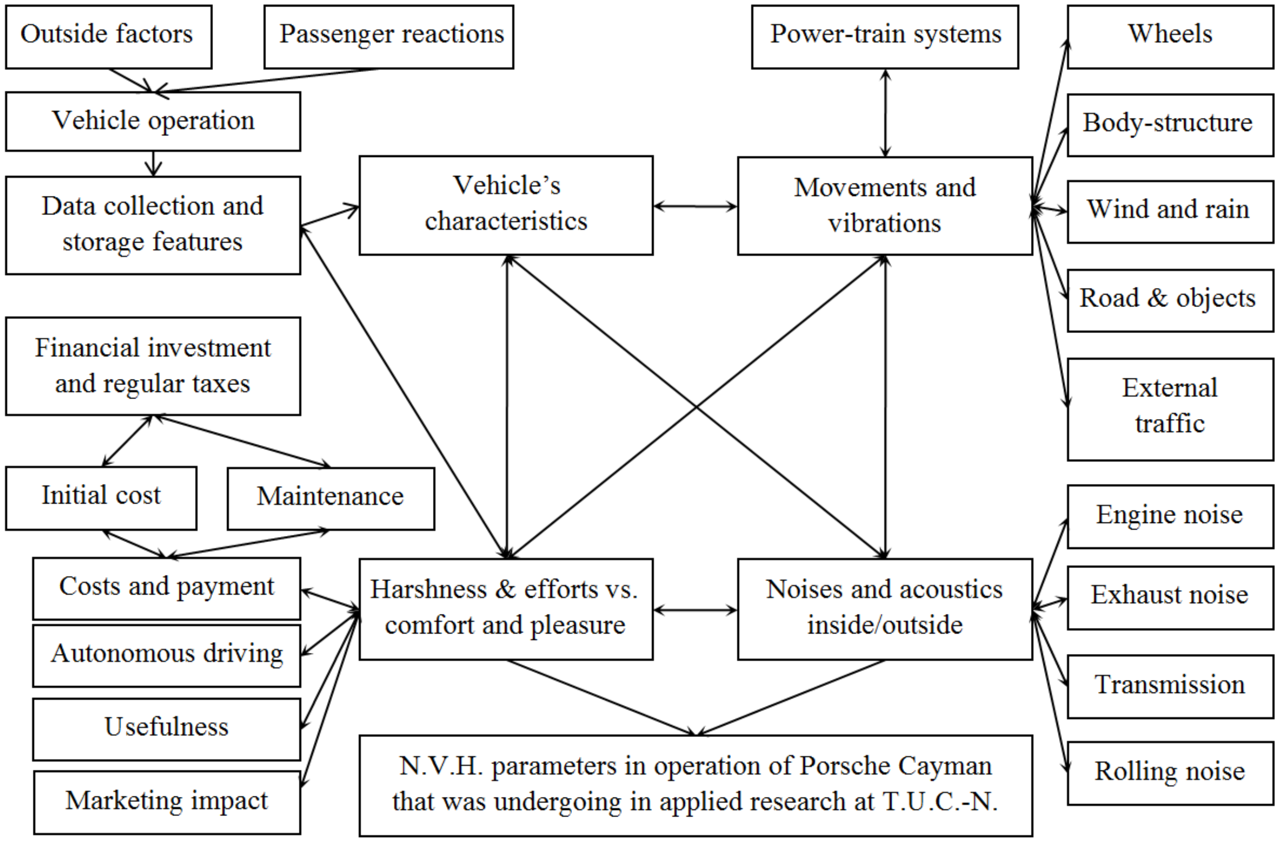

Figure 1. Short methodology in determining noise, vibrations and harshness

The tested and evaluated car is a fully operational model, which was studied in relation with noise emissions, in-use vibrations and harshness level. An innovative method of advanced engineering research was sketched out. The interaction between computer processing power and N.V.H. evaluation effort is consistently highlighted.
The new design methods are starting to consider NVH issues throughout the whole design process.
This involves integrating extensive modelling, simulation, evaluation, and optimization techniques into the design process to insure both noise and vibration comfort. New materials and techniques are also being developed so that the damping treatments are lighter, cheaper, and more effective. Noise, Vibration and Harshness, more commonly known as NVH, is an all-encompassing engineering discipline that deals with the objective and subjective structural dynamic and acoustic aspects of automobile design. The NVH engineer is interested in the structural dynamic response of the vehicle from the complete assembled system down to the normal modes of the individual components. As a vehicle is a moving dynamic system, its response to stochastic, time varying inputs is important for safety, quality, and comfort of the passengers.
One specific area of study within NVH is vehicle acoustics. Sound plays an important part in the development of a motor vehicle 0.

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2.2. Materials

Used materials consist in one full operational motored vehicle (Figure 2) Porsche Cayman model year 2016 and the auxiliary equipment and devices necessary for experimental testing.

As experimental equipment used in the developed tests were two mobile devices for noise measuring (md367fd/a) and data recording (A1367EMC2407 IC579C-E2407), also there was a test bench for vibration measurements (PF N750) as well as digital post-processing stations.

Figure 2. Available motored vehicle for experimental measurements concerning N.V.H. aspects

2.3. Mathematics and calculus

Noise is a special category of sounds and it is defined as complex sound which generates an audible harshness. The elements which define the noise analyze are a part of the physiologic acoustics.

The study of the link between the excitation and sensation intensity is made by Weber-Fechner law. According to this empirical law, for a hearing sensation variation in geometrical progression there is a correspondence in received hearing sensation variation in logarithmic progression, as in the next model 00:

$$S = \frac{l}{k} * \ln \frac{I}{I_o} \tag{1}$$

where:

S is hearing sensation, in *Np*;

k – integration constant;

I – sound intensity, in W/m^2;

I_o – minimal reference intensity perceptible in sound field, in W/m^2.

Taking into consideration the physical relation between intensity and acoustic pressure there may be determined the level of sound pressure level N_p, as mathematical model 000:

- theoretical:

$$N_p = 20 \lg \frac{p}{p_o} \tag{2}$$

and, applied for close proximity:

$$N_p = 20 * \log_{10} \frac{1.7 * 10^3}{2 * 10^5} = 85 \quad \text{dB}$$

where:

p_o is minimal acoustic pressure coresponding to the reference intensity, which has the value of $2 \cdot 10^{-5}$ N/m^2;

p – instant pressure, in N/m^2.

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3. EXPERIMENTAL SETUP AND RESULTS

The vehicle and testing equipment were interconnected via contact or contactless technical devices.
The vehicle was brought in the Automobile laboratory from the Technical University were is a specialized vibration test bed available in order to monitor the Porsche Cayman behavior in some frequencies range of motion, especially the wheel train. Beside that there was also installed specialized equipment (microphone stand alone bench and software operated device) to measure the noise level multiple locations and various distances in 15 meters range from the inspected sporty car as it is shown in figure 3.

3.1. Experimental setup for testing

In order to get accurate testing results and to make some interesting observation in relation with the sound waves generated by the operational systems and powertrain there were implemented some innovative and advanced engineering strategies in following the research protocol on Porsche Cayman (Figure 3).

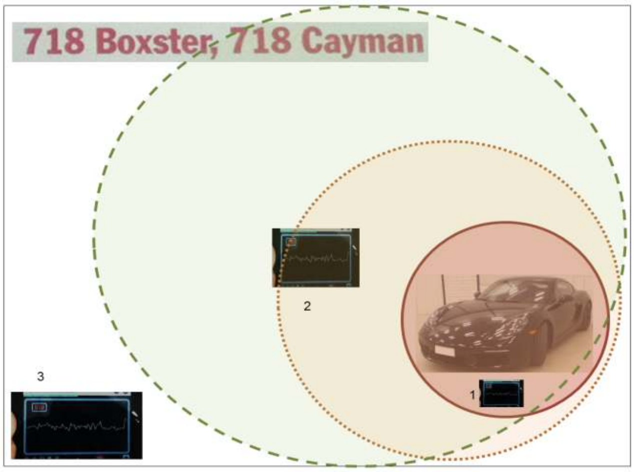

Figure 3. Points for measuring the noise level variation in motored vehicle car range (1-in close range<5m; 2-medium range, 5÷10 m; 3-distance of 15 m)

The vibrations are analyzed on a specialized test bed available at Technical University of Cluj-Napoca as shown in figure 4, considering them in a cumulative protocol.
Other main sources of noise and vibrations are vehicle speed, engine speed and transmission as they are plainly presented in figure 5 and figure 6, mentioning the locations for n.v.h. generators.

Figure 4. Schematic image of vehicle on vibrational test bed (1-vibrational plates of test bench; 2-tested vehicle for vibrations and harshness; 3-vibration damping testing results on vehicle's front axle; 4-vibration damping testing results on vehicle's rear axle)

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Figure 5. Sources of N.V.H. in vehicle operation
(1-vehicle speed meter on board of tested vehicle; 2-tahometer for engine speed on board of Cayman tested model; 3-complex cutaway transmission model with 7 + 1 gears and multi disc dual clutch for both input shafts)

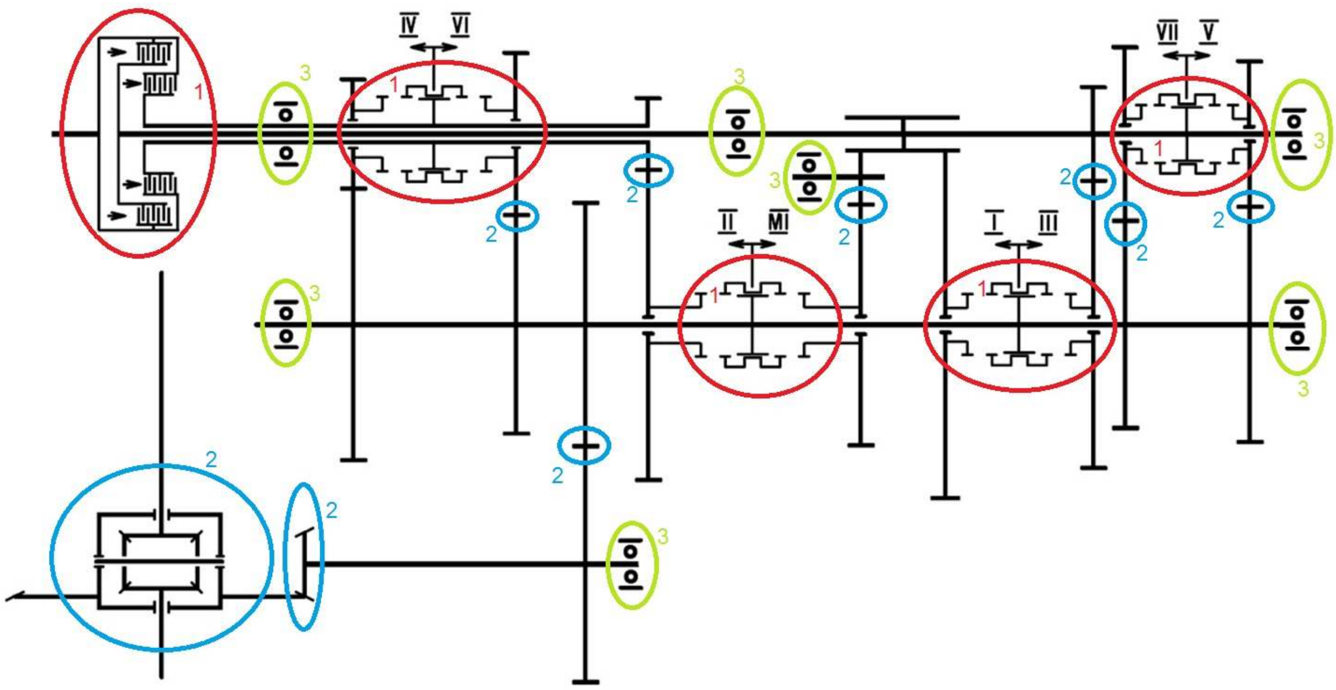

Figure 6. Simplified model of complex Direct Shift Gearbox 7 + 1 as a N.V.H. source.
1-main sources for noise and vibrations at gear change; 2-secondary sources of noise and vibrations; 3-lowest N.V.H. generation sources.

3.2. Experimental testing results

By setting up a computer assisted experimental model (Figure 7) and measuring the noise level (Figure 8), considering the vibration critical range and analyzing harshness we can assess and evaluate the comfort level. The present chapter shows a synthesis of some results collected with an engineering method for computer aided testing (C.A.T.) and appreciation in the field of N.V.H. parameters concerning the operation of sporty Porsche Cayman car (see table 1), with specific applicability on optimization process of the studied vehicle. The tested and evaluated car is a fully operational model, which was inspected concerning noise emissions (Figure 9 and Figure 10), in-use vibrations and harshness level. An innovative method of advanced engineering research was sketched out. The interaction between computer processing power and N.V.H. evaluation effort is consistently supporting the evaluation process.

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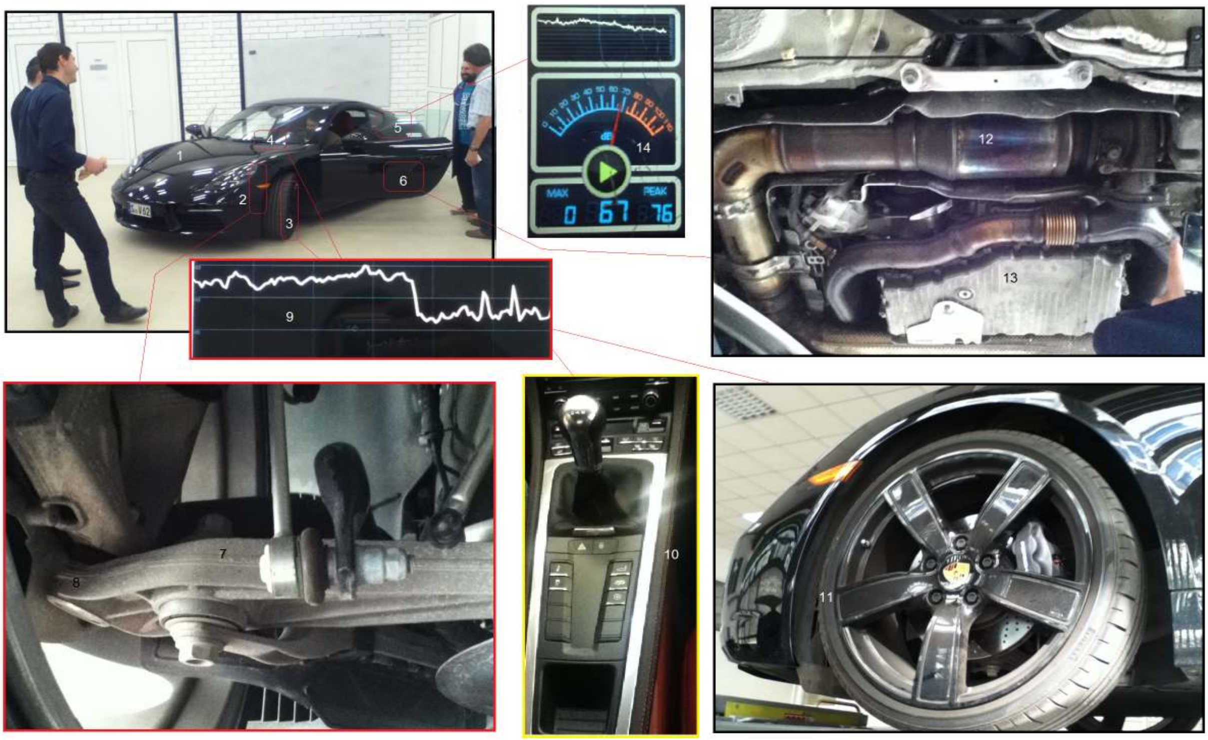

Figure 7. Research equipment from Technical University and studied Porsche Cayman (1-Tested car; 2-front suspension; 3-wheels; 4-on-board recording equipment; 5-engine sound measurement; 6-power train; 7-lower suspension arm; 8-joint; 9-sound curve; 10-gear change; 11-tire; 12-exhaust; 13-gearbox; 15-software app)

Noise level [dB]

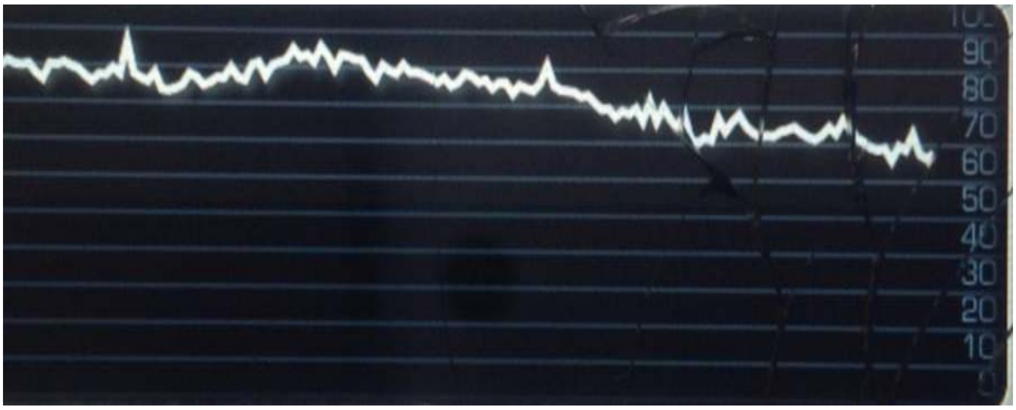

Figure 8. Noise variation from start in accelerating process of tested car.

Table 1.
Vehicle technical date

Parameter	Value
Manufacturer	Porsche
Model	Cayman
Code	982
Fuel	Gasoline
Pollution standard	Euro 6

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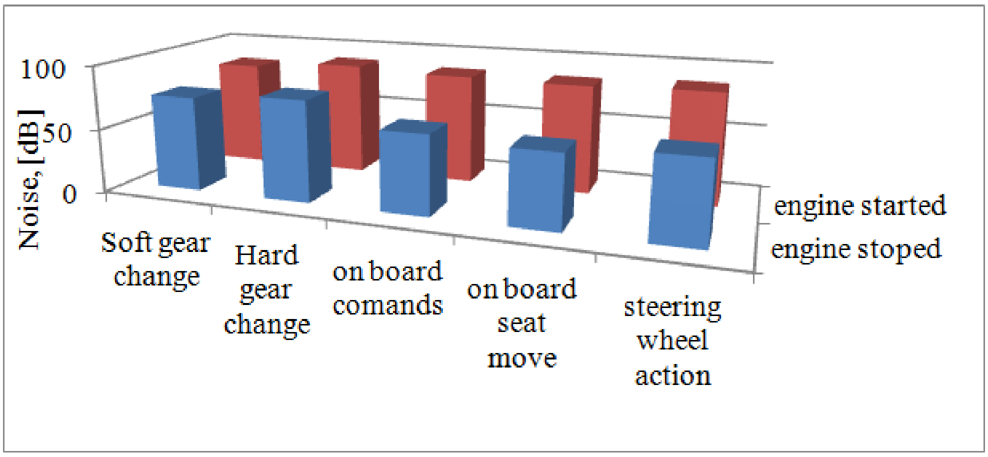

Figure 9. Noise values in various conditions of Porsche Cayman

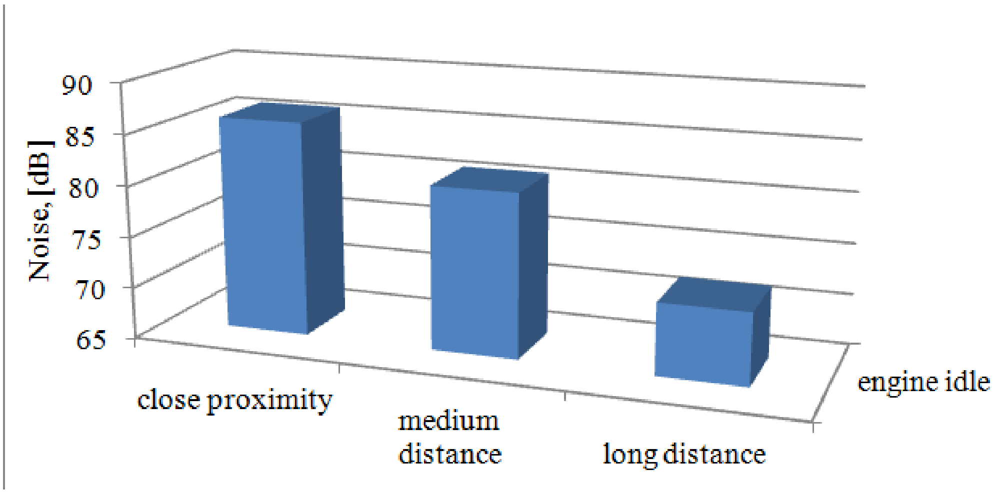

Figure 10. Noise values at various distances from Porsche Cayman

3.3. Limitations of the proposed study

The research is restricted to the development and validation of an innovative affordable engineering method in computer aided testing or evaluating the N.V.H. parameters for Porsche Cayman sporty motor vehicle and it surely may be extended and applied in other cases and situations also.

The proposed personalized analysing method worked adequately in the developed work for Porsche Cayman, but needs to be highly defined in order to apply it for large series of motor vehicles.

3.4. The novelty of the achievement

The research crew members are experienced in internal combustion engines, transmissions, pollution, alternative energy sources for green vehicles and mechanical construction technologies. The consideration of N.V.H. aspects in operating conditions represents a synthesizing endeavour of capabilities, technologies and experiences. Even if there were also considered in the previous work some particular tests of vehicle's mass oscillations, this is the first integrating process of so many important aspects that define N.V.H. evaluating criteria and advanced engineering methods based on computer processing power available today. Originality of the paper consists in research method and applied protocol, while the applicability is both in acoustics engineering and automotive development.

4. CONCLUSION

Redefining the standards in comfort and tuning is close related to the N.V.H. research. N.V.H. parameters are not acting as isolated factors in some isolated environment.

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They should be considered in relation with the complex whole which is the operating motor vehicle in the road traffic environment. The N.V.H. aspects have an engineering influence in the vehicles development and operation, but also they have a financial impact when it comes to car marketing.
Noise levels of analysed sporty car are higher than regular series BMWs or other usual motored vehicles, an aspect which influences both user's satisfaction and attention.
Close range measurements of noise show actual values between 75 to 130 dB, when the engine is running and speeds up at high revs level, with applicability in car design and operation optimization.
By departing from the operating vehicle the sound level is ranged from 45 to 68 dB.
The initiated study creates development possibilities and perspectives which facilitate and encourage further research of the specific problems.

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TURBOSYSTEM MODELING WITH RECIPROCATING SLIDING

Andrei POSTARU, Gheorghe POSTARU, Victor CEBAN*

Technical University of Moldova, Str. Studentilor 9/8, Chisinau, Republic of Moldova

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Abstract: *The paper addresses the issue of the influence that the law of friction has on the dynamic behavior of the mechanical system that interacts with a tribosystem. The emergence of certain nonlinearities of higher order into the law of friction leads to an intensification of the dissipative process and to tribosystem destabilization. Consequently, friction excited self-oscillations are generated into the elements of the mechanical system with a wide spectrum of frequencies, sustained from the external source of energy. The theoretical and experimental modelling of the dissipative process and of the generation of is based on the frictional harmonic oscillator that interacts with the tribosystem. The oscillator is used as a sensitive element to the fluctuations of the frictional force and as a measure of the dissipated energy. Starting from the model, the elaboration of a method and of devices for experimental research provided the opportunity to study the behavior of the turbosystem in unstable operating.*

Key-Words: *Turbosystem, friction, sliding, wearout, Lagrange equation.*

1. INTRODUCTION

As functional components of mechanical systems, tribosystems collaterally affect the former's dynamic behavior and have a predominant role in energy dissipation. The evolution character of the dissipative process is influenced and correlatively connected with the friction characteristic (law) occurring at the relative motion of contact surfaces. At present, a series of different laws have been formulated only for dry friction: simple-static; complicated-dynamic. Considering also lubrication (with Stribeck effect), the series of the friction laws diversify [2][5][8]. In fact, the laws of friction are complex and include the influence of a series of factors of a different nature related to working, geometric and micro geometric, to tribosystem structure, to the source, properties, and characteristics of the materials for triboelements, and to the working environment.

When nonlinearities of different orders occur into the friction characteristic (with fluctuations in the frictional force), in the mechanical system elements are generated noises under the form of self-oscillations with a wide spectrum of frequencies [2][5][8].

The structure of the spectrum, the amplitude and shape of the oscillations are influenced by charging parameters, the friction regime and energy dissipation factors, the properties and state of the materials for triboelements and the lubricant, the origin and intensity of processes arising in the contact area. For problems of such complexity a reliable research method remains the experimental one. However, experimental modelling should be formalized and executed within the framework of the fundamental equations of nonlinear dynamics.

2. DYNAMIC MODELING OF MECHANICAL SYSTEM-TURBOSYSTEM INTERACTION

A model commonly used to describe and study the oscillatory processes in different systems is the harmonic oscillator. The oscillator is the basis of both mathematical models and necessary technical devices for tests and the experimental research of the studied systems.

The oscillator has been accepted as a model for studying the interaction between the tribosystem characteristics and the mechanical system (figure 1). It consists of block 1 with mass *m* link connected to housing 4, fixed on both sides by means of two similar elastic elements 2, of low rigidity *c*.

* *Corresponding author e-mail: dr_ceban@yahoo.com*

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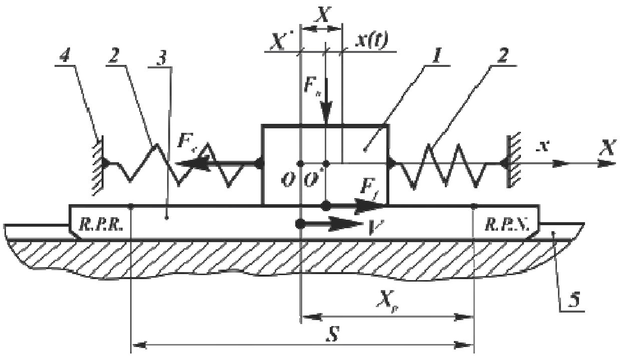

Figure 1. Mechanical oscillator scheme

The angular frequency of the oscillator is $\omega = (c/m)^{1/2}$. The tribological connection between the oscillator and the triboelements is realized through the contact between block 1 and platform 3.
Platform 3, driven by a crank-type mechanism, performs a translational reciprocating movement on guide 5 within distances S, with speed:

$$V = r\Omega\left(\sin\varphi_m + \frac{\lambda}{2}(1 - \sin 2\varphi_m)\right) \tag{1}$$

where:
Ω - the angular speed of the crank,
r - the radius of the crank;
l - the length of the rod;
λ - the rotation angle of the crank.
and
$\lambda = r\varphi_m / l$.
Initially, the *X* coordinate's origin of the gravity center of block 1 is in the stable equilibrium point *O*. When the platform begins to move with speed *V* on distance *S*, block 1, influenced by the friction force F_f will move in direction *X* with speed $\dot{X}$.
The relative speed between the contact surfaces of the friction bond becomes $v_r = \dot{X} - V$.
Connecting the oscillator to the tribosystem results in a system composed of two subsystems of different nature (mechanical and dissipative) with own dynamic behaviour, influencing each other during working. The evolution of the dissipative process (of energetic essence) can be studied only from the perspective of Lagrangean formalism, according to which the generalized dissipative force Q_d derives from a force function called Rayleigh dissipative function [6][7], defined by the relationship:

$$\Phi_d = \sum_{j=1}^{N} k_j \int_0^{v_j} f_j(u)\,du \tag{2}$$

where: k_j and $f_j(u)$ – the positive functions defined on spaces j of the contact real elementary areas that are dependent on the $q=X$ coordinate and on the generalized speed $\dot{q} = \dot{X}$ of the oscillator, on speed *V* of the platform, and on the internal and external parameters of the tribosystem; v_j– the relative local speed of the surfaces on the contact real elementary areas of the spaces; N – the number of real elementary areas within the boundaries of the contact nominal area. In the sliding tribosystem the role of dissipative generalized force Q_n is played by the total force of friction F_f, defined by the gradient of the dissipative force in the direction of the relative motion of the contact surfaces [2][8]

$$Q_d = F_f = -\frac{\partial}{\partial(\dot{X} - V)} \sum_{j=1}^{N} k_j \int_0^{v_j} f_j(u)\,du = -\frac{\partial \Phi_d}{\partial v_r} \tag{3}$$

where: X- $V = v_r$ – the generalized speed is represented by the relative speed of the surfaces when the platform and the block of the oscillator move.

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The movement of the oscillator under the action of the dissipative forces is described by the Lagrange equation

$$\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{X}}\right) - \frac{\partial L}{\partial X} = -\frac{\partial \Phi_d}{\partial v_r} \tag{4}$$

where: $L= (T - \Pi)$ – the Lagrange function (kinetic potential); ($T=mx^2/2$) - the kinetic energy of the oscillator; ($\Pi = cx^2/2$) – the potential energy accumulated into the elastic elements of the oscillator. If the working parameters are maintained at constant level, the oscillator asymptotically stabilizes his position temporarily "*freezing"* in the vicinity of the unstable equilibrium point O* (figure 1) with coordinate X*. When the position of the block is stabilized, speed $\dot{X}\to 0$, and the contact relative speed becomes $v_r = -V$. In this state the oscillator passes in a steady and stable working mode relative to the point X^*, where block 1 will be in the balance of forces in the movement direction of the platform $(F_f + F_s = 0)$, where:
$F_f = F_c$ – the constant component of the Coulomb type friction force;
$F_s = -cX^*$ - the force of elasticity.
On the oscillator's passing in steady state, the dissipative friction force between the block and the platform remains linearly dependent on the generalized coordinate X^* and the platform speed V. Lagrange equation for the steady state conditions of the oscillator takes the form.

$$-\frac{\partial L}{\partial X^*} = \frac{1}{2}c\frac{\partial}{\partial X^*}(X^*)^2 = cX^* = -\frac{\partial \Phi_d}{\partial(-V)} = F_c = const \tag{5}$$

According to (5), the friction force F_c and the energy dissipation power $P_d(V)$ for motion with relative speed $(-V)$.

$$F_c = cX^*$$
$$P_d(V) = cX^*V \tag{6}$$

The loss of stability violates the balance of forces, the movement of the oscillator being determined by the variation of the dissipative forces. In the event of some instability in the operation of the tribosystem, with disruptive fluctuations of the friction force, the oscillator passes into a self-oscillation regime maintained from the external source of energy.
The nature and evolution of the dissipative process can be efficiently set in the analysis result of the oscillator's motion in the phase space (figure 2) built in the phase coordinates $Y=X$ and $Z=\dot{X}/\omega$.
To this aim, the ratio between the motions of the representative point M on the phase trajectories for each two cycles in a row (i) and (i+1) are examined step by step.
In examining the movement on the phase trajectory of cycle i in self-oscillation state (figures1, 2) the coordinate and speed of the oscillator are:

$$X_i = X_i^* + x_i,\ \dot{X}_i = \dot{X}_i^* + \dot{x}_i,$$

where:
$X_i^* = (X_i^{max} + X_i^{min})/2$ - average component;
x_i - variable component;
$\dot{X}_i^* = \Delta X_i^* \omega$ - speed of passage from the previous cycle (i-1) to cycle i.
On the i cycle path, two types of movement can be identified: 1 - with low speed $\dot{X}_i^*$, determined by the variation of the mean component coordinate X_i^* of the cycle; 2 -with high speed $\dot{x}_i$, determined by the variation of the x_i coordinate and angular own frequency ω of the oscillator.
Admitting the prime integral $\left(\dot{X}_i \frac{\partial L_i}{\partial \dot{X}_i} - L_i = E_i\right)$ of equation (4) as total energy of the oscillator, represented in the phasic space (figure 2) through the orbit of level h_i drawn with the representative radius R_i is the following is obtained:

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$$E_i = \frac{1}{2}c\left[X_i^2 + \left(\frac{\dot{X}_i}{\omega}\right)^2\right] = \frac{1}{2}c(X_i^*)^2 + cX_i^* x_i + \frac{1}{2}c\left[x_i{}^2 + \left(\frac{\dot{x}_i}{\omega}\right)^2\right] = \frac{1}{2}cR_i^2 = h_i \qquad (7)$$

where:

$$R_i = \sqrt{Y^2 + Z^2} = \sqrt{X_i^2 + \left(\frac{\dot{X}_i}{\omega}\right)^2}$$

- the representative radius of the h_i orbit at intersection with the trajectory of the representative point M_i trajectory at movement on cycle in the phasic space.

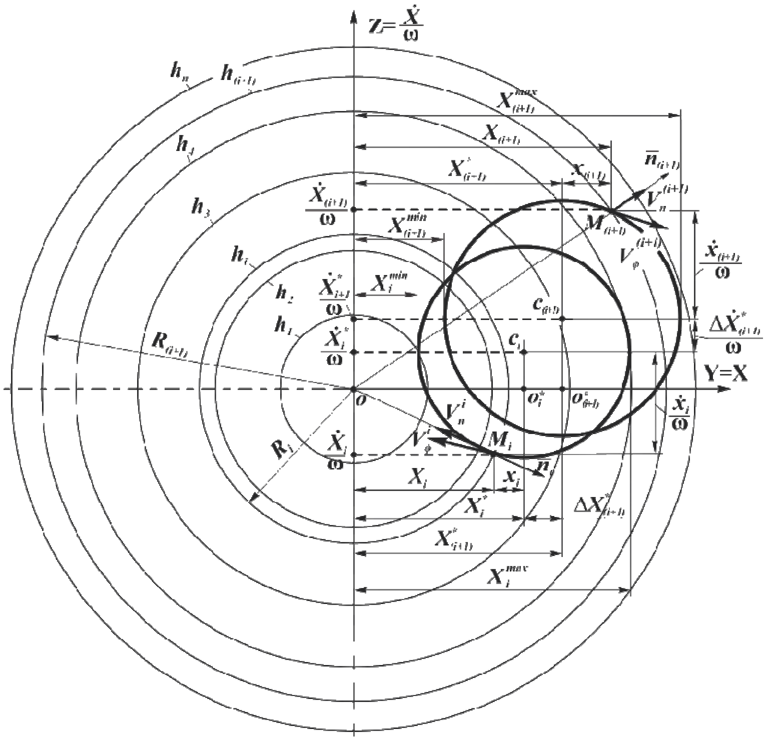

Figure 2. Representation of self-oscillatory process in phase space

Differentiating equation (7) by time, the energy variation of the oscillator and the evolution of the dissipative process under the influence of the friction force during the *i* cycle is determined:

$$\frac{dE_i}{dt} = c\left(X_i\dot{X}_i + \frac{\dot{X}_i\ddot{X}_i}{\omega^2}\right) = c\left[X_i^*\dot{X}_i^* + \left(X_i^*\dot{x}_i + \frac{\dot{X}_i^*\ddot{x}_i}{\omega^2}\right) + \left(x_i\dot{x}_i + \frac{\dot{x}_i\ddot{x}_i}{\omega^2}\right)\right] = cR_iV_n^i \qquad (8)$$

where V^i_n – the projection of the phasic speed V^i_Φ on the direction of the normal $\tilde{n}_i$, drawn through the M_i representative point moving on the phasic trajectory of the cycle when intersecting the representative orbit h_i .

From a dynamic point of view, V^i_n is the rate of change (dissipation) of the oscillator's energy at movement of the M_i point on the phasic path.

According to relation (8), the variable component of the oscillator's energy includes three groups of factors:

$X_i^*\dot{X}_i^*$ – for variation of the low speed and low frequency component;

$x_i\dot{x}_i + \frac{\dot{x}_i\ddot{x}_i}{\omega^2}$ – for variation of the high speed and high frequency component;

$X_i^*\dot{x}_i + \frac{\dot{X}_i^*\ddot{x}_i}{\omega^2}$ – for mutual influence of the low and high speed factors.

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Setting the mechanical status of the oscillator by experimental methods, based on expressions (7) and (8), the normal component V_n^i of the phasic speed is determined, which comprises the three groups of influence factors.

$$V_n^i = \frac{X_i \dot{X}_i + \frac{\dot{X}_i \ddot{X}_i}{\omega^2}}{R_i} = \frac{X_i^* \dot{X}_i^* + \left(X_i^* \dot{x}_i + \frac{\dot{X}_i^* \ddot{x}_i}{\omega^2}\right) + \left(x_i \dot{x}_i + \frac{\dot{x}_i \ddot{x}_i}{\omega^2}\right)}{\sqrt{X_i^2 + \left(\frac{\dot{X}_i}{\omega}\right)^2}} = (V_n^i)^{**} + (V_n^i)^* + (V_n^i)^v \tag{9}$$

At temporary *"freeze"* of the oscillator's mechanical status in the point of coordinate X_i^* the instant steady state functioning conditions are obtained fixed on the average level of the oscillation cycle generated by the stationary friction force F_c^i at the platform movement with speed V^i . Based on relations (5) and (6), the solution for friction force and energy dissipation power on the average cycle component is obtained.

$$F_c^i = cX_i^* \tag{10}$$

$$P_d^i(V^i) = -cX_i^* V^i \tag{11}$$

The total energy dissipation power in the contact area for each movement cycle consists of two basic components: $P_d^i(V^i)$ – instantly constant, defined by speed V^i within cycle limits; variable $P_d^i(V_n^i)$, defined by speed V_n^i.

$$P_d^i = P_d^i(V^i) + P_d^i(V_n^i) = -c(X_i^* V^i + R_i V_n^i) \tag{12}$$

Taking into account relation (9) the following is obtained:

$$P_d^i = F_f^i v_r^i = -\frac{\partial \Phi_d^i}{\partial v_r^i} v_r^i = -[cX_i^* V^i + cR_i (V_n^i)^{**} + cR_i (V_n^i)^* + cR_i (V_n^i)^v] \tag{13}$$

The total instantaneous value of the friction force during cycle (*i*) conditioned by the achievement of the working unstable self-osscilation state:

$$F_f^i = \frac{P_d^i}{v_r^i} = -\left[cX_i^* \frac{V^i}{v_r^i} + cR_i \frac{(V_n^i)^{**}}{v_r^i} + cR_i \frac{(V_n^i)^*}{v_r^i} + cR_i \frac{(V_n^i)^v}{v_r^i}\right] \tag{14}$$

or

$$F_f^i = [F_c^i + (F_v^i)^{**} + (F_v^i)^* + (F_v^i)^v] \tag{15}$$

where:

$$F_c^i = -cX_i^* \frac{V^i}{v_r^i}$$

$$(F_v^i)^{**} = -cR_i \frac{(V_n^i)^{**}}{v_r^i}$$

$$(F_v^i)^* = -R_i \frac{(V_n^i)^*}{v_r^i}$$

$$(F_v^i)^v = -cR_i \frac{(V_n^i)^v}{v_r^i}.$$

The component F_c^i (of Coulomb type) of the friction force is defined by the linear factors of the dissipative function Φ_d in the vicinity of the X_i^* coordinated point.

The variable (fluctuating) components of the friction force $(F_v^i)^{**}$, $(F_v^i)^*$, $(F_v^i)^v$ occur as a result of various dynamic effects from the contact zone with higher order nonlinearities and may vary in a wide range of frequencies and amplitudes.

When condition $v_r = \dot{x} - V < 0$ is accomplished, the F_c component of the friction force has always the same sense as platform speed vector V and determines the energy dissipation level in the contact zone at relative motion of the surfaces.

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The variable components (depending on the speed signs $(V_n^i)^{**}$, $(V_n^i)^{*}$, $(V_n^i)^{v}$sign) can change their sign according to direction within the limits of the same oscillation cycle.
On the negative direction, the energy previously accumulated in the oscillator's elements dissipates in the contact area, and, on the positive direction, through the tribosystem, a new portion of energy from the external source is introduced into the oscillator.
This behaviour relates to the achievement of the dynamic effect of variable dissipation on direction, also called "negative friction" effect, which is the main cause of engendering friction excited self-oscillations in the mechanical system [1] [2][5][7].
Accepting cycle (*i*) (*i*) as a benchmark, at crossing to the next cycle (*i*+1), the X_i^* (coordinated point (figure 2) acquires an additional movement $\Delta X^*_{(i+1)}$ with the speed $X^*_{(i+1)}$, where:

$$\Delta X^*_{(i+1)} = X^*_{(i+1)} - X^*_i$$
$$\dot{X}^*_{(i+1)} - \Delta X^*_{(i+1)}\omega$$
$$X^*_{(i+1)} = \left(X^{max}_{(i+1)} + X^{min}_{(i+1)}\right)/2$$

The coordinate, the absolute speed and the relative speed within the cycle limits (i + 1) with respect to cycle (i) will be:

$$X_{(i+1)} = \left(X^*_{(i+1)} + x_{(i+1)}\right) = \left(X^*_i + \Delta X^*_{(i+1)} + x_{(i+1)}\right)$$
$$\dot{X}_{(i+1)} = \left(\dot{X}^*_{(i+1)} + \dot{x}_{(i+1)}\right)$$
$$v_r^{(i+1)} = \left(\dot{X}_{(i+1)} - V^{(i+1)}\right).$$

The prime integral of equation (4) for the cycle (*i*+1).

$$E_{(i+1)} = \frac{1}{2}c\left[\left(X^*_{(i+1)}\right)^2 + \left(\frac{\dot{X}^*_{(i+1)}}{\omega}\right)^2\right] + c\left[X^*_{(i+1)}x_{(i+1)} + \frac{\dot{X}^*_{(i+1)}\dot{x}_{(i+1)}}{\omega^2}\right] + \frac{1}{2}c\left[x^2_{(i+1)} + \left(\frac{\dot{x}_{(i+1)}}{\omega}\right)^2\right] = h_{(i+1)} \quad (16)$$

Similarly, the instantaneous frictional force is determined at the motion of the oscillator during the (i+1) cycle:

$$F_f^{(i+1)} = -\frac{P_d^{(i+1)}}{v_r^{(i+1)}} = -\left[cX^*_{(i+1)}\frac{V^{(i+1)}}{v_r^{(i+1)}} + cR_{(i+1)}\frac{\left(V_n^{(i+1)}\right)^{**}}{v_r^{(i+1)}} + cR_{(i+1)}\frac{\left(V_n^{(i+1)}\right)^{*}}{v_r^{(i+1)}} + cR_{(i+1)}\frac{V_n^{(i+1)}}{v_r^{(i+1)}}\right] \quad (17)$$

$$F_f^{(i+1)} = \left[F_c^{(i+1)} + \left(F_v^{(i+1)}\right)^{**} + \left(F_v^{(i+1)}\right)^{*} + \left(F_v^{(i+1)}\right)^{v}\right] \quad (18)$$

The loss of system stability can occur for both types of movements: of low frequency and low speed, and with high frequency and high speed.
As an experimental criterion for assessing the movement regime, the deviations of the displacements $\Delta X^*_{(i+1)}$ and $\Delta x_{(i+1)}$between each pair of consecutive cycles are used:
- under steady state oscillatory motion

$$\begin{cases} \Delta X^*_{(i+1)} = X^*_{(i+1)} - X^*_i = 0 \\ \Delta x_{(i+1)} = \left(x^{max}_{(i+1)} - x^{min}_{(i+1)}\right) - \left(x_i^{max} - x_i^{min}\right) = 0 \end{cases} \quad (19)$$

- under unstable state oscillatory motion

$$\begin{cases} \Delta X^*_{(i+1)} = X^*_{(i+1)} - X^*_i \neq 0 \\ \Delta x_{(i+1)} = \left(x^{max}_{(i+1)} - x^{min}_{(i+1)}\right) - \left(x_i^{max} - x_i^{min}\right) \neq 0 \end{cases} \quad (20)$$

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3. EXPERIMENTAL RESULTS

Based on the above theoretical model, an original model of tribometer was made, with a cyclic translational motion and equipped with proper systems for measuring the dynamic characteristics (variables) of the system.
As dynamic variables the following are used:

1. displacement $X_i\ (mm)$i in the movement direction of the platform of the mass center of the oscillating part against point "O" of the oscillator's stable equilibrium;
2. linear speed $v_i = \dot{X}_i (mm/s)$ of the mass center of the oscillating part;
3. linear acceleration $a_i = \ddot{X}_i \left(mm/s^2\right)$ of the mass center of the oscillating part;
4. linear speed $V^i (mm/s)$ of the platform in the movement direction within the limits of cycle *i* of the oscillator's oscillation;
5. cyclic frequency of the platform $n_c\ (min^{-1})$;
6. experimental average temperature $\theta_k\ (^oC)$ in the contact area during cycle *k* of the platform movement.

The dynamic variables are recorded, step by step, in the form of time series for each cycle (*i*) of the oscillator's movement and over each cycle (*k*) of the platform movement.
Computerized technologies for recording the dynamic variables and for processing of the time series were used to study the behavior of the tribosystem in unstable operating conditions.
The evolution of the friction process was estimated through the variation characteristic (law) of the friction force $F_f^k = F_f^k(v_r)$ for relative speed variation; the evolution of the dissipative process (energy dissipation) was estimated through the work of the friction force W_f^k cumulated for each cycle (*k*) of the platform movement.
In the case of the translational cyclical movement, the accuracy of local friction force determination depends on the resolution $s = \frac{T_k}{T}$s of the system, established within the limits of the platform movement cycle, where $T = \frac{2\pi}{\omega}$ - the period of the oscillator's self-oscillation, $T_k = \frac{2\pi}{\Omega}$ - the period of cycle *k* of the platform movement.
Resolution *s* represents the oscillator's number of cycles included in a platform movement cycle.
As a benchmark characteristic for establishing the experimental friction law, the integrated mean within the limits of cycle (*i*) period of the local friction force is used

$$F_f = \frac{\omega}{2\pi} \int_{\frac{\pi}{\omega}}^{+\frac{\pi}{\omega}} F_f^i\, dt \tag{21}$$

where F_f^i - the instantaneous value of the friction force within the cycle (*i*) limits.
The level of energy dissipation (represented by the work of friction forces integrated on period *T* of cycle *i*.

$$W_f^i - \int_{-\frac{\pi}{\omega}}^{+\frac{\pi}{\omega}} F_f^i\, v_r^i\, dt \tag{22}$$

The work of friction forces (dissipated energy) W_f^k during the cycle (k) period of the platform movement totalled

$$W_f^k = \sum_{i=1}^{s} W_f^i \tag{23}$$

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The results of the experimental data analysis for a number of couples of materials and lubricants revealed a different and complex behavior of the frictional force (figure 3) on different portions and areas of the characteristics points, at relative movement of the contact on the platform cycle strokes (*S*).
Within a cycle the following are identified: the DSM stroke of the platform motion on the direct sense of the movement and the OSM stroke for opposite movements; portions with acceleration movement (AM) per stroke and deceleration movement zones (DM); (ZRP) zones of return points of the contact per stroke and (PMS) zones of points of maximal speed of the cycle.
If the platform is actioned by the crank mechanism, return points with different kinematic characteristics are obtained at the end of the strokes: (RPN) - return point near; (RPR) - return point removed.
A pronounced dynamic behavior of the friction force occurs when changing speed direction at entry into and exit from the areas of return points.

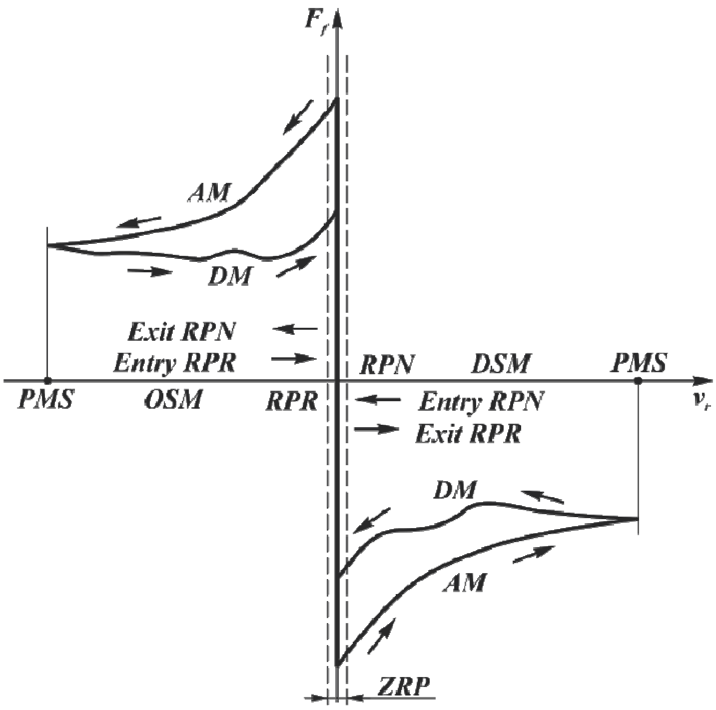

Figure 3. Law of friction determined experimentally for cyclical translatory movement

During the experimental research, the following conditions were set:
- oscillator mass *m*=0,2 kg;
- rigidity of the elastic element *c*= 100N/mm;
- angular frequency of the oscillator ω = 6280 s^{-1};
- angular frequency of the crank Ω = 31,141 s^{-1};
- cyclic frequency of the platform n_c = 300 min^{-1};
- system resolution *s* = 200;
- contact load in two ways (1 – with constant normal loading during the testing period, 2 - loading in consecutive steps); drop lubrication.

The experimental temperature θ in the contact area was measured with a *K* type mini *K* thermocouple. Temperature evolution in the contact area occurs due to heat produced at the exhaust of mechanical energy.
Contact form - flat with dimensions: width *b* = 20 mm; length *l*=40 mm.
Stroke length S = 100 mm.
Under study was the tribological behavior of the materials of the pair: block triboelement - electrolytic chromium on a steel surface; platform triboelement - 38KH2MYUA steel (similar to 41CrAlMo7).
The contact was lubricated with MT- 16P GOST 6360-83 oil (similar to SAE 40 API CB).
Experimental results are shown in figures 4 and 5.
For temperature increase in the contact area (figure 4) for a load F_n = *2,0 kN*, the energy dissipation level per cycle (*k*) of platform movement will vary non-linearly with a trend of asymptotic stabilization, between the work values of the friction forces of W^k_f = 40 … 50 J.

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If the operating parameters are maintained constant (F_n - load and cyclic speed of the platform n_c), process stabilization is complete at temperature values θ = 240 ... 250 °C.

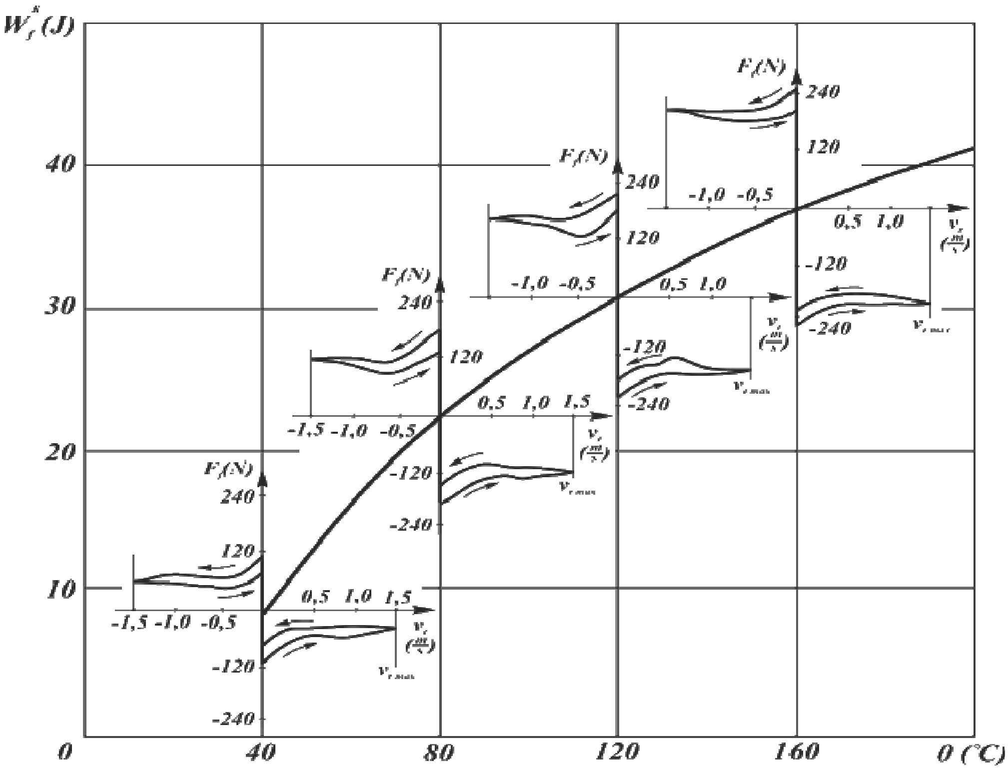

Figure 4. Character of the dissipative process and evolution of the friction law with experimental temperature variation θ in the contact area for normal force loading F_n = 2,0 kN

Of special importance for determining the dynamic behaviour of mechanical systems is the evolution of the law of friction in concrete working conditions.
Figure 4 shows a significant influence of temperature on friction force for relative speed v_r variation per cycle strokes of platform movement.
When temperature increases, the friction force changes both values and the variation manner per stroke.
Different behaviour is identified if the variation of the contact load force F_n takes place at constant temperatures θ.
To achieve a boundary lubrication regime, during the experiment the temperature was maintained at high-level in the contact zone, with a permissible variation within θ = 200 ... 210 °C.
When the contact was loaded (figure 5), a practically linear dependence of the level of energy dissipation W^k_f on force F_n was obtained.
In this case. the evolution manner of the force in the experimental friction law changes.

4. CONCLUSIONS

In cases when the friction law (for the couple of materials used in the construction of the triboelements) higher order nonlinearities occur, the friction forces generate in the mechanical system elements noises as self-oscillations in a wide range of frequencies. Based on Lagrange equation, the dynamic model of the interaction between the mechanical system and the tribosystem was developed.
The harmonic oscillator with elastic elements was accepted as mechanical system for modeling.
The examination of the dynamic model identified the structure of the friction force in unstable operating conditions of the mechanical system. In the structure of the total friction force F_f four possible components appear: a component F^i_c (of Coulomb type), defined by the linear factors of the dissipative function Φ_d; three variable (fluctuating) components within the limits of each oscillation cycle $(F_v^i)^{**}$, $(F_v^i)^{*}$, $(F_v^i)^{v}$ (occurr as a result of various dynamic effects in the contact zone with higher order nonlinearities and can vary over a wide range of frequencies and amplitudes).

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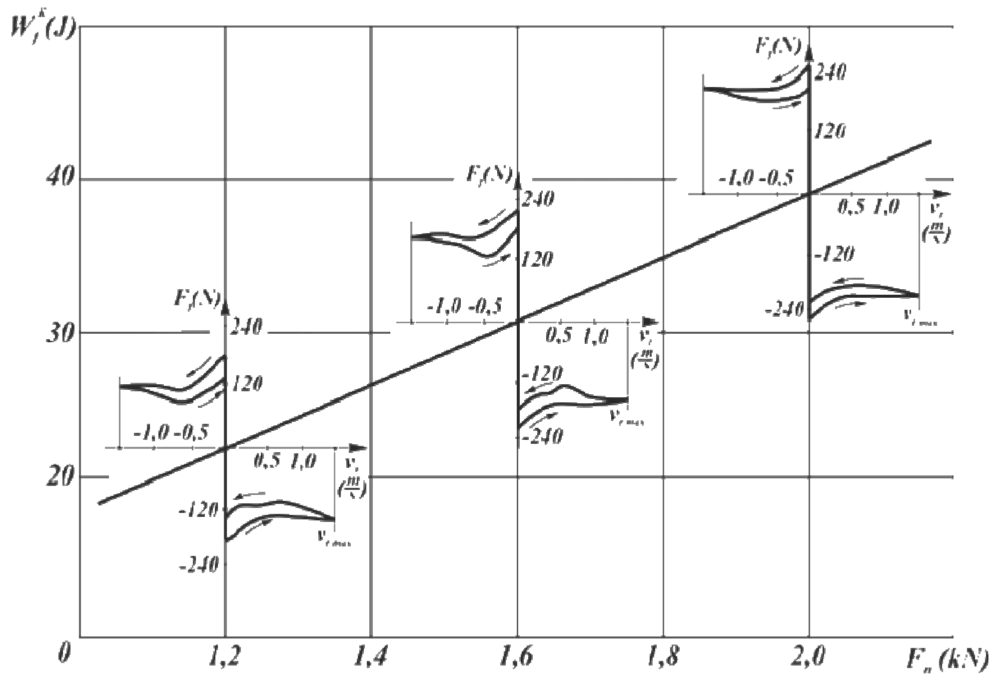

Figure 5. Nature of the dissipative process and evolution of friction law for variation of the load force F_n at constant temperature θ into contact area

Based on the model with harmonic oscillator, an original model of tribometer has been made, with cyclical translational movement, equipped with proper measuring systems for the dynamic characteristics of the oscillator, and the method for experimental determination [3] of the dynamic characteristics of the sliding tribosystem in unstable operating regime was developed.

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RESEARCH ON GEOMETRIC AND KINEMATIC ANALYSES OF SOME AUXILIARY MECHANISMS USED ON URBAN BUSES

Daniela ANTONESCU*, Daniel IOZSA, Ovidiu ANTONESCU, Gheorghe FRATILA

Politehnica University of Bucharest, Spl. Independentei, Nr. 313, Bucharest, Romania

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Abstract: *The paper presents the geometric-kinematic modeling of an articulated planar mechanism used to actuate the urban bus doors. The stages of solving the kinematic parameters as displacement, velocity and acceleration of each component of the linkage are performed. One of the goals of the paper is to use this modeling of kinematics to improve the operation of the mechanism as synchronization of the two folding doors. The velocity and acceleration of them are also important due to the fact that it can create a safe environment for the passengers. Before this study, a similar type of linkage used on trolleybuses is presented in brief. Also, a review of the main types of urban bus doors is emphasized in section 2.*

Key-Words: *urban bus, auxiliary mechanism, geometrical and kinematic analyse, bus doors*

1. INTRODUCTION

For a high frequency of stops and a high number of passengers getting on and off, as well as for ensuring the safety of the passengers on boarding and during the journey, the urban vehicles are equipped with doors consisting of two or more folding parts, pneumatically or electrically controlled.
A mechanism for actuating the city bus doors is generally composed [1][17] of two main parts: a) the control mechanism, mounted either under the stairs or above the door; b) the crank-slide final mechanisms to which either the crank and the coupler or only the couplers or the cranks are rigidly linked to one part of the folding door.
The structural and kinematic analysis of the pneumatic mechanisms used to actuate the urban bus doors [4] highlights the unitary character of the control mechanisms.
Also, a general method of structural and geometric analysis of the control mechanisms, in open-closed positions, as well middle positions, is of interest.
In section 2 of the paper the main types of urban bus doors are presented. The most used mechanisms for actuating the doors are with circular sliding or rotational-sliding motions.
In section 3 a linkage having only revolute joints in the final mechanism structure, used on trolleybuses, is presented.
Finally, in section 4, a geometric-kinematic modeling of a two folding door-part mechanism is performed. The displacement, velocity and acceleration of each element of the linkage have been calculated and displayed. Based on these results, some improvements regarding the passenger comfort are outlined.

2. MECHANISMS FOR BUS DOORS

2.1. Swinging door (conventional) with hinge joint

In the case of classical design buses (Figure 1a) the doors have a swinging movement around a vertical axis by means of some hinges.

* Corresponding author e-mail: daniela.mihalache@yahoo.com

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2.2. Door with circular sliding motion

In modern buses one of the door movement solutions is circular sliding, being done by means of an articulated parallelogram mechanism (Figure 1b). The kinematic scheme of such a linkage shows that the vehicle door (represented by the segment MN) is rigid connected with the coupler AB (Figure 2).

In the practical case of bus doors, the positioning of the fixed joints A_0 and B_0 is made inside the body 0, in the area of the stairway.

Figure 1. Classical swinging doors (a) and circular sliding doors (b) [18]

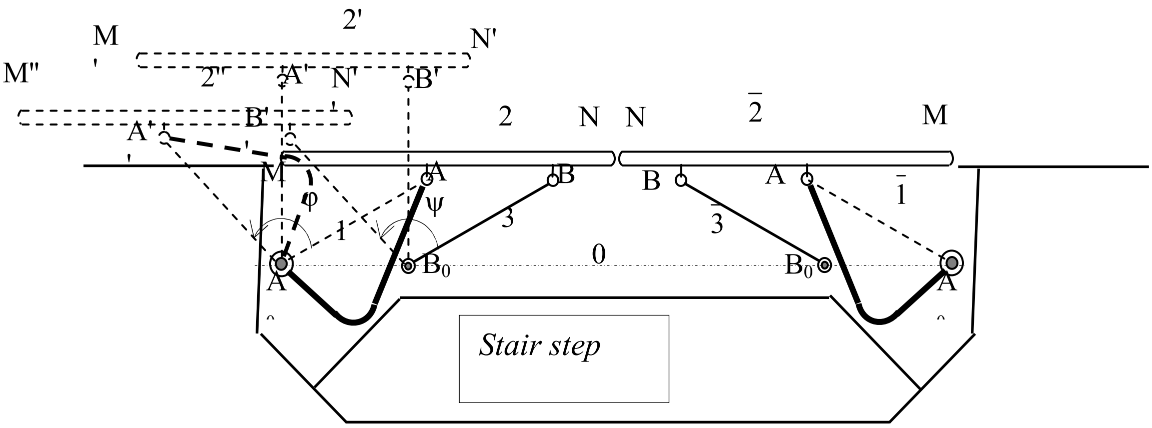

Figure 2. The kinematic scheme of the double symmetrical parallelogram mechanism

Each of the two door-parts is rigid with the coupler 2 of the parallelogram mechanism A_0ABB_0 in the closed position of the door with width *MN*. In the open position of the door *M"N"* the parallelogram mechanism is $A_0A'B'B_0$.
Note that the bar 1 cannot be in a straight line as it would interfere with the bus body, so that the shape of it is curved and it does not collide with the vehicle body (Figure 2).
The kinematic scheme of the parallelogram mechanism was drawn in three positions, two extreme positions (closed and open) and an intermediate position *M'N'* at the maximum distance of the body.
For each door-part a parallelogram mechanism is corresponding, whose kinematic schemes are symmetrically represented.
Due to the fact that the bar 1 A_0A is the driving kinematic element, it has a much larger cross-section than the bar 3, having primarily a geometric role.
Note that for the right part, the parallelogram mechanism was only in the closed position represented.

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2.3. Door with planar rotational-sliding motion

The door of the bus can be made of two parts articulated each other, of which one of the parts performs a horizontal rotation and the other one performs a planar rotational-sliding motion.

The most commonly used solution is the one in which the door is made of a single part having a horizontal planar movement (rotational-sliding). When the door is opened, the door-part is fully folded or partially folded inside the bus. In an intermediate position of the door opening movement (Figure 3) it is observed that the door-part is rigid connected to the coupler of a crank-coupler planar mechanism (Figure 4). The control and actuation mechanism of the door is usually located at the top of the bus body due to the saved space inside the bus and the lower road clearance.

Figure 3. Doors with one-part inside

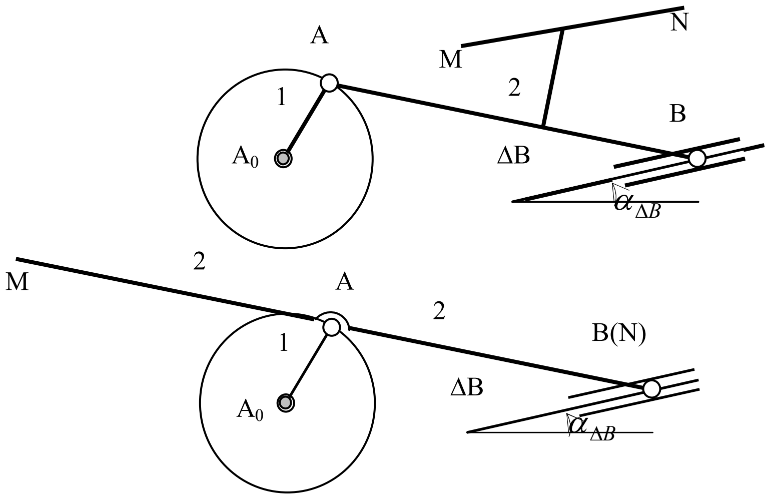

Figure 4. The kinematic scheme of the folding crank-coupler planar mechanism

3. THE DOOR MECHANISM OF A TROLLEYBUS TYPE TV

This type of mechanism (Figure 5) consists of three series-connected mechanisms: MC(1,2,3), ME1(3,4,5,6,7) and ME2 (6,8,9). The main structure of the linkage is a planar triadic chain.

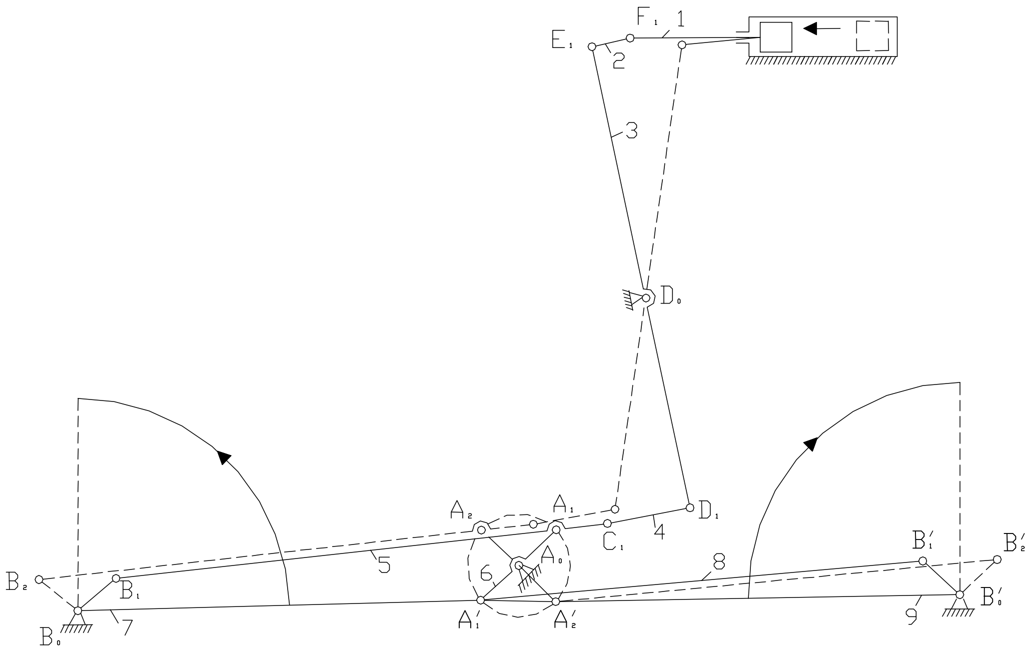

Figure 5. The kinematic scheme of planar triadic chain

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The final mechanism consists of a parallelogram (5, 6, 7) and an anti-parallelogram (6, 8, 9) so that the transmission ratio [9] achieved between the driven elements 7, 9 is -1. The piston 1 operates with double effect, the compressed air acting on both sides, and by means of the connecting rod 2 the linear sliding motion is turned into a rotational motion, swinging the rocker 3 (Figure 5).
Further, the movement is transmitted, by the coupler 4, to the parallelogram (5, 6, 7) and then, via the element 6, to the anti-parallelogram (6, 8, 9).
The doors or the main parts of the folding doors are fixed to the driven elements 7, 9 (having limited rotation up to 90°), according to the kinematic scheme.

4. THE GEOMETRIC-KINEMATIC MODELING OF A BUS DOOR MECHANISM

The next kinematic scheme (Figure 6) shows the piston 1 as driving element which is positioned, at one moment in time, by means of the linear displacement $s_{10}=s_1$ in the fixed horizontal cylinder 0.
Through the displacement of the point A, the two elements of the dyad chain LC(2,3), the coupler 2 and the rocker 3, will incline by the angles φ_2 and φ_3 measured in the points B and B_0.

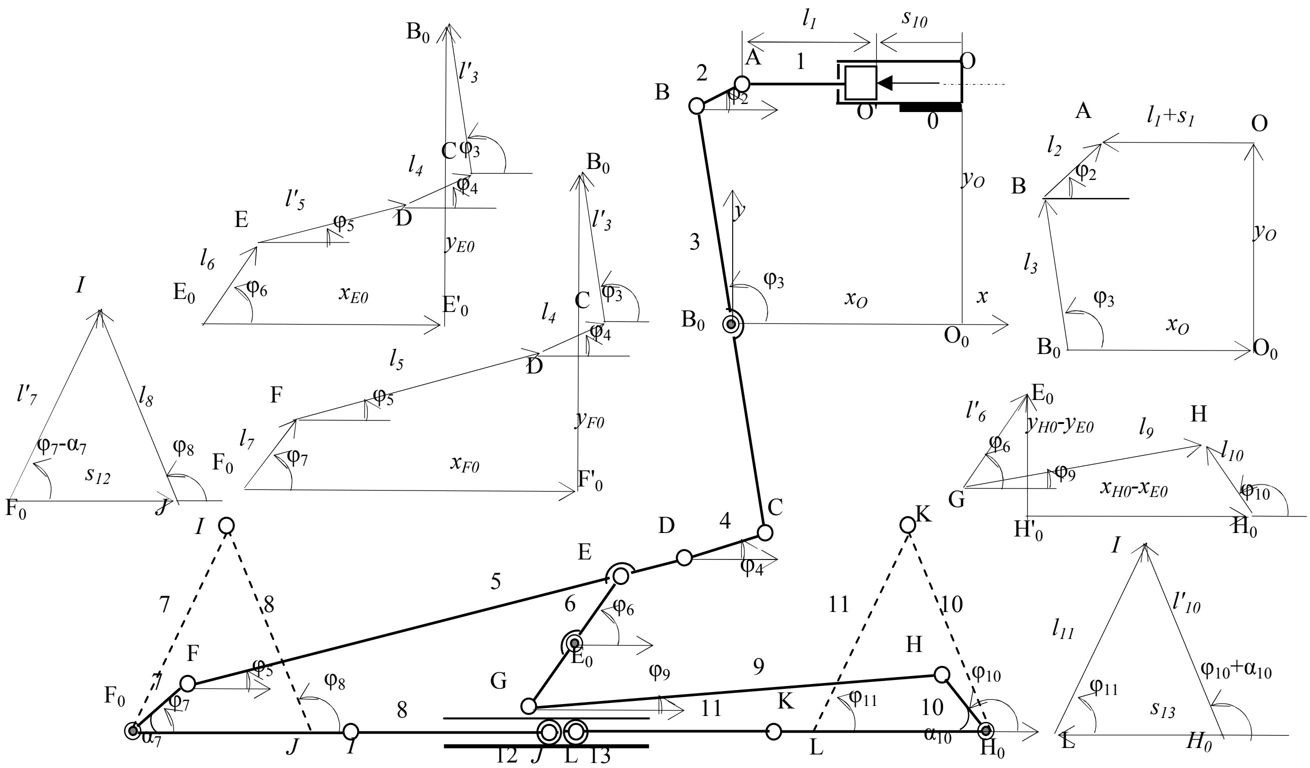

Figure 6. The kinematic scheme of the bus door mechanism (in closed position) and the independent kinematic contours

Further, the swinging movement of the point C is transmitted through the coupler 4 to the triadic chain LT(4,5,6,7) with two fixed articulations E_0 and F_0 (Figure 6).
The element 6 has two articulations opposed along the diameter E and G, so that through the rotate joint G the motion is transmitted to the dyad chain LC(9,10).
On each of the bars 7 and 10 the first door–part is rigid connected, by which the second door-part 8 and 11 is articulated (in the points I and K) whose movement is type planar rotational-sliding.
The point J and L have sliding motion through the rollers 12 and 13 that move in a rectilinear guide.
Note that for the extreme displacement of the piston 1 *(S_{1max})* the two doors are closed, the two door-parts of one door being in prolongation: (7, 8) and (10, 11).
Starting with the actuator mechanism MA(0,1), the structural formula of the planar “motor mechanism” (MM) has the following expression:

$$MM = MA(0+1) + LD(2+3) + LT(4+5+6+7) + LD(8+9) + LD(8+12) + LD(11+13) \tag{1}$$

According to this formula of structural compounding of the motor mechanism, six independent closed kinematic contours have been identified, of which four contours specific to the four dyad chains and two independent contours specific to one tetrad chain.
The characteristic lengths of the linkage are relative constant parameters with respect to the coordinate system chosen in point B_0, and the absolute constant parameters are fixed or mobile (the characteristic lengths of the kinematic elements represented by the distances between their articulation centers).
The characteristic geometric parameters with constant values are the following lengths and angles:

$$x_O = 380mm; y_O = 400mm;\ x_{E_0} = -200mm; y_{E_0} = -390mm;$$
$$x_{F_0} = -860mm; y_{F_0} = -450mm;\ x_{H_0} = 460mm; y_{H_0} = -450mm;$$
$$l_1 = O'A = 240mm; l_2 = AB = 60mm; l_3 = BB_0 = 400mm; l_3' = B_0C = 320mm;$$
$$l_4 = CD = 130mm; l_5 = DF = 740mm; l_7 = FF_0 = 90mm;\ \alpha_7 = \angle(FF_0I) = 45^0;$$
$$l_7' = F_0I = 320mm; l_8 = IJ = 320mm;$$
$$l_5' = DE = 80mm; l_6 = EE_0 = 90mm;\ l_6' = E_0G = 90mm;$$
$$l_9 = GH = 660mm; l_{10} = HH_0 = 90mm; \alpha_{10} = \angle(HH_0K) = 45^0;$$
$$l_{10}' = H_0K = 320mm; l_{11} = KL = 320mm;$$

The variable lengths are the linear displacements achieved by the elements 1 (actuator piston) and the rollers 12 and 13 in the horizontal guide (Figure 6):

$$OO' = s_{10} = s_1; FJ = s_{12}; H_0L = s_{13}.$$

For each planar closed contour two scalar equations can be written, so that for the entire mechanism twelve scalar equations with twelve unknowns are deduced, that represent ten angular displacements $\varphi_i(i = 2,...,11)$ and two linear displacements $s_i(i = 12,13)$.

$$l_2\cos\varphi_2 + l_3\cos\varphi_3 = x_O - (l_1 + s_1) \tag{2}$$
$$l_2\sin\varphi_2 + l_3\sin\varphi_3 = y_O \tag{3}$$
$$l_3'\cos\varphi_3 + l_4\cos\varphi_4 + l_5\cos\varphi_5 + l_7\cos\varphi_7 = -x_{F_0} \tag{4}$$
$$l_3'\sin\varphi_3 + l_4\sin\varphi_4 + l_5\sin\varphi_5 + l_7\sin\varphi_7 = -y_{F_0} \tag{5}$$
$$l_3'\cos\varphi_3 + l_4\cos\varphi_4 + l_5'\cos\varphi_5 + l_6\cos\varphi_6 = -x_{E_0} \tag{6}$$
$$l_3'\sin\varphi_3 + l_4\sin\varphi_4 + l_5'\sin\varphi_5 + l_6\sin\varphi_6 = -y_{E_0} \tag{7}$$
$$l_7'\cos(\varphi_7 - \alpha_7) - l_8\cos\varphi_8 - s_{12} = 0 \tag{8}$$
$$l_7'\sin(\varphi_7 - \alpha_7) - l_8\sin\varphi_8 = 0 \tag{9}$$
$$l_6'\cos\varphi_6 - l_9\cos\varphi_9 + l_{10}\cos\varphi_{10} = -(x_{H_0} - x_{E_0}) \tag{10}$$
$$l_6'\sin\varphi_6 - l_9\sin\varphi_9 + l_{10}\sin\varphi_{10} = -(y_{H_0} - y_{E_0}) \tag{11}$$
$$l_{10}'\cos(\varphi_{10} + \alpha_{10}) - l_{11}\cos\varphi_{11} + s_{13} = 0 \tag{12}$$
$$l_{10}'\sin(\varphi_{10} + \alpha_{10}) - l_{11}\sin\varphi_{11} = 0 \tag{13}$$

This system with twelve scalar equations (2 ...13) can be divided in five systems of nonlinear equations, of which four systems of two equations (specific to the dyad chain) and one system of four nonlinear equations (specific to the triad chain). The resolving order of these systems of two or four nonlinear equations is given by the structural formula (1).
Using MathCAD software the kinematics of the bus door linkage has been accomplished.

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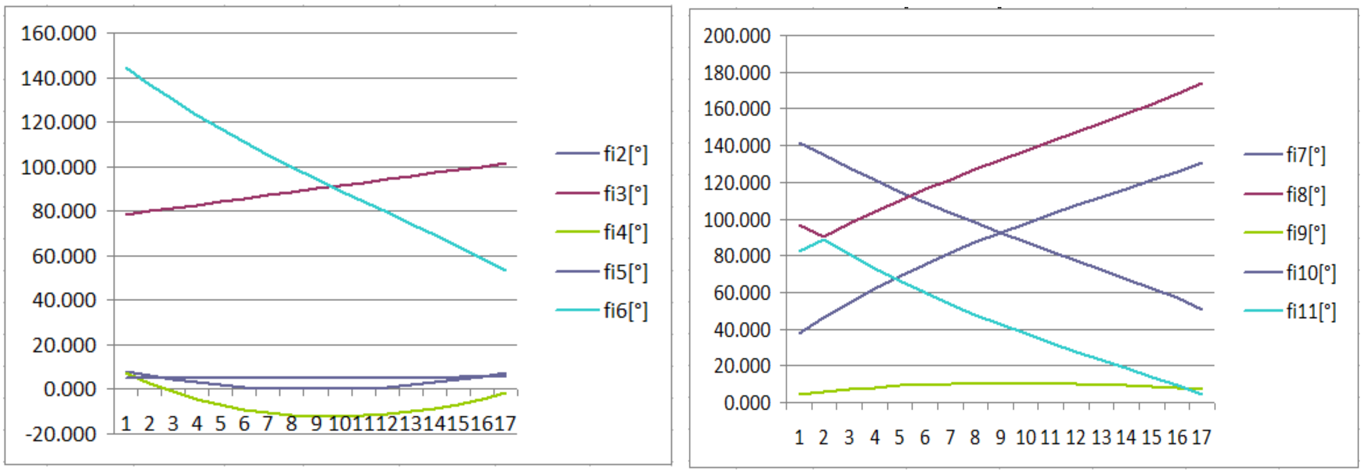
The software managed to resolve the twelve equation system only in stages as it has been mentioned above. The twelve variables (ten angular and two linear displacements) have been displayed in figure 7 and respectively in figure 8.
Along with these graphs the numerical tables 1 and 2 were also shown. As it can be observed in figure 8, the linear displacements s_{12} and s_{13} have pretty matched trajectories which means that the mechanism is operating almost in a symmetry way.
This is a positive sign regarding the working balance between the two left/right folding doors.
In figure 9 the angular and linear velocities have been displayed.
Also, in figure 10 the angular and linear accelerations have been displayed.
The issue of "*how fast the bus door should operate*?" is an optimization problem. If the speed of it is too high, the passengers could be struck, and if it is too low, the bus will be delayed.
The main advantage of folding bus doors is the space saving inside the bus and the safety that it can provide to the passengers.

Tablle 1.
Angular displacements of the mechanism elements

1	fi2[°]	fi3[°]	fi4[°]	fi5[°]	fi6[°]	fi7[°]	fi8[°]	fi9[°]	fi10[°]	fi11[°]	S1k
2	7.852	78.381	7.547	4.934	144.234	141.651	96.651	4.127	37.590	82.590	0
3	5.960	79.874	2.914	4.991	136.968	134.682	90.318	5.758	46.054	88.946	0.01
4	4.351	81.348	-1.041	5.037	129.964	127.879	97.121	7.131	54.156	80.844	0.02
5	3.007	82.807	-4.323	5.073	123.278	121.330	103.670	8.235	61.765	73.235	0.03
6	1.918	84.256	-6.962	5.105	116.923	115.069	109.931	9.082	68.847	66.153	0.04
7	1.077	85.697	-9.002	5.132	110.887	109.094	115.906	9.699	75.422	59.578	0.05
8	0.478	87.134	-10.489	5.157	105.140	103.385	121.615	10.114	81.534	53.466	0.06
9	0.119	88.567	-11.470	5.180	99.644	97.908	127.092	10.354	87.243	47.757	0.07
10	0.000	90.000	-11.984	5.201	94.357	92.624	132.376	10.444	92.610	42.390	0.08
11	0.119	91.433	-12.066	5.223	89.234	87.488	137.512	10.403	97.697	37.303	0.09
12	0.478	92.866	-11.742	5.244	84.227	82.456	142.544	10.245	102.563	32.437	0.1
13	1.075	94.300	-11.030	5.266	79.287	77.476	147.524	9.979	107.269	27.731	0.11
14	1.912	95.734	-9.940	5.289	74.359	72.492	152.508	9.611	111.872	23.128	0.12
15	2.987	97.169	-8.467	5.314	69.380	67.438	157.562	9.138	116.439	18.561	0.13
16	4.301	98.602	-6.592	5.342	64.270	62.226	162.774	8.555	121.046	13.954	0.14
17	5.852	100.033	-4.270	5.375	58.919	56.736	168.264	7.845	125.794	9.206	0.15
18	7.637	101.459	-1.408	5.417	53.153	50.771	174.229	6.972	130.836	4.164	0.16



a) *b)*
Figure 7. Graphs of the angular displacements: φ_2 to φ_6 (*a*) and φ_7 to φ_{11} (*b*)

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Table 2.
Linear displacements of the mechanism elements

1	S12	S13	S1k
2	0.00000	0.00000	0
3	0.003554	0.011774	0.01
4	0.079337	0.101838	0.02
5	0.151249	0.184608	0.03
6	0.218171	0.258754	0.04
7	0.279612	0.324069	0.05
8	0.335495	0.380991	0.06
9	0.385983	0.430254	0.07
10	0.431359	0.472686	0.08
11	0.471945	0.509081	0.09
12	0.508045	0.540151	0.1
13	0.539914	0.566489	0.11
14	0.567727	0.588564	0.12
15	0.591549	0.606712	0.13
16	0.611291	0.621114	0.14
17	0.626621	0.631757	0.15
18	0.636757	0.638311	0.16

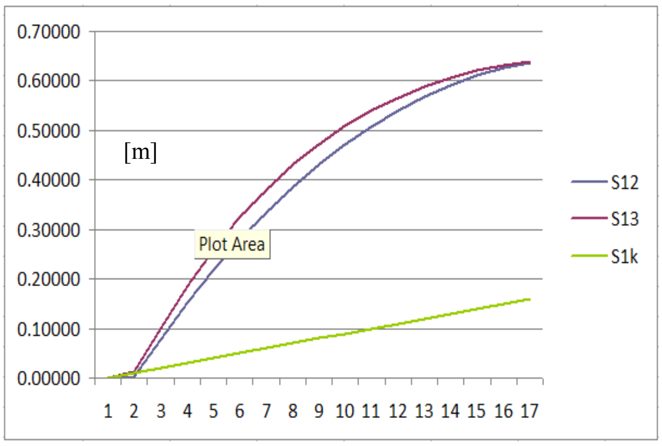
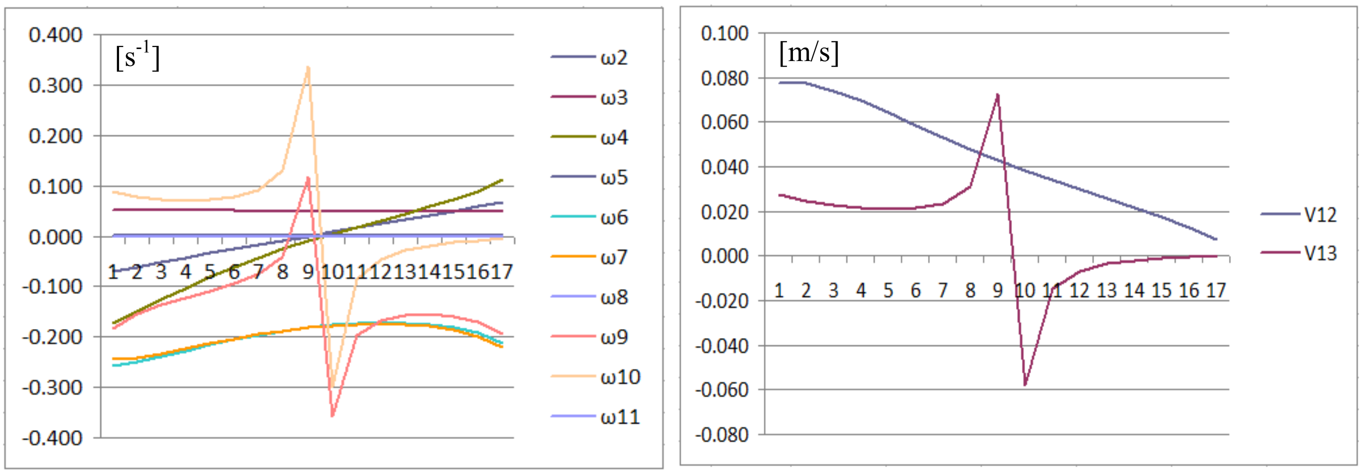
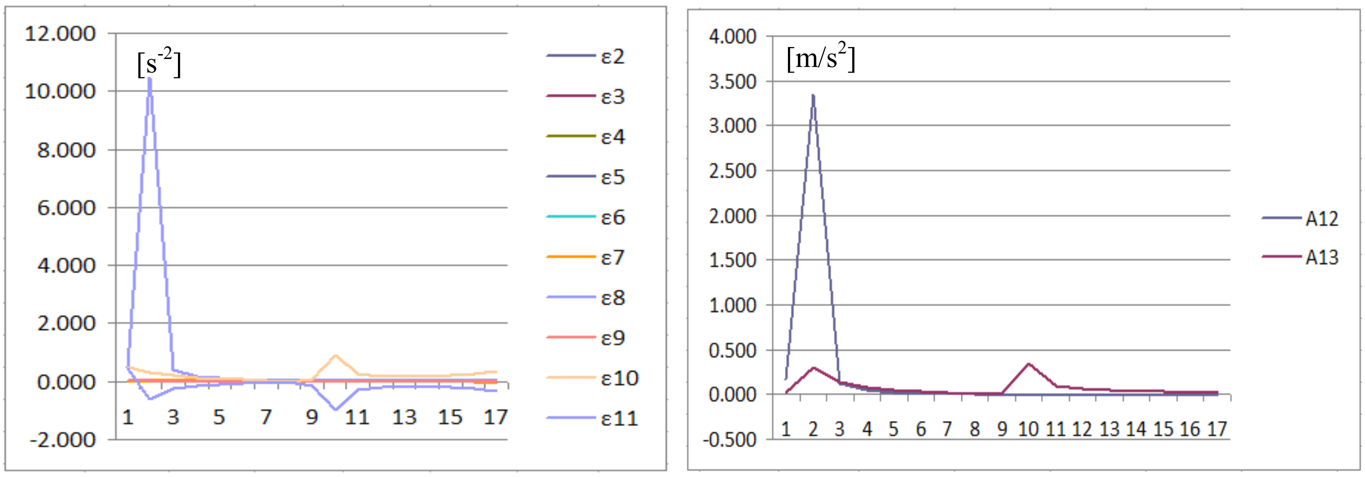


Figure 8. Graphs of the linear displacements: s_{12} and s_{13}



a) *b)*

Figure 9. Graphs of the angular/linear velocities: ω_2 to ω_{11} (a), v_{12} and v_{13} (b)



a) *b)*

Figure 10. Graphs of the angular/linear accelerations: ε_2 to ε_{11} (a), a_{12} and a_{13} (b)

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5. CONCLUSIONS

The paper presents a geometric-kinematic modeling of an articulated bar mechanism for actuate the doors of the urban buses.
It is emphasized that in the linear displacement graph of the kinematics of the linkage final elements, the two curves are nearly matched. This is pretty acceptable due to the fact that the mechanism should operate in a synchronized way.
Also, it has been outlined that the velocities and accelerations of the folding doors are very important for the passenger comfort and for the tight schedule of the bus driver. An optimized operation of the mechanism is the solution to this matter.

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DEVELOPMENT OF PUBLIC PASSENGER TRANSPORT - THE ROLE AND PLACE OF PUBLIC PASSENGER TRANSPORT IN METROPOLITAN AREAS

Victor BRATU*

University of Pitesti, Str. Targu din Vale, Nr. 1, Pitesti, Romania

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Abstract: *The study concludes governance models of metropolitan areas is that whatever solution is adopted, depending on the specific national or regional problems facing will be necessary authority or agency with the capacity planning (spatial) control, revision , preservation and application of metropolitan strategy. The metropolitan area must have the professional resources strategic planning in the medium and long term policy analysis at the metropolitan level, correlate and establish balance between sectorial interests and those of the metropolitan area. With the advent of the automobile and the transition to mass production of various auto brands were developed in parallel and road transport networks, linking the large urban areas. Populated urban centers, rural areas and are directly affected by the exponential growth of population mobility and more striking movement of goods. Road traffic is orderly movement of vehicles and people, focusing on areas of land specially arranged for this purpose, ie roads. World car park reached impressive numbers currently circulating in over 800 million vehicles of all types and categories, and every year we witness the manufacture increasingly more means of transport. Concerns worldwide enhancers public transport system people refer to various measures to reduce congestion and increase the attractiveness of high-capacity transport, especially underground and use its premises for cultural or commercial. Guidelines on the evolution of human mobility, depending on certain input variables such as socio-economic, spatial planning, transport policies and behavior of the population, bring into relief some scenarios, namely "Homo Technicus", "Homo Economicus", "Homo Contractor", "Homo Politicus" and "Homo Civis”.*

Key-Words: *public transport, metropolitan areas, networks, road traffic*

1. INTRODUCTION

Transport is one of the main components of social and economic life, of human society. They continue and complete the process of production of material goods, moving them to the place of consumption. Transport creates new products, unlike industry and agriculture, which transforms the production process work items into new products.

Public passenger transport is a subfield of social-economic activity through which the movement of people in space, using vehicles on certain traffic routes in order to meet certain needs and spiritual society. Development of transport was required for the development of production factors.

Continuous increase in production volume materials, the need to exchange goods need to travel long distances for goods and people development were the main drivers of upward transport of deepening labor dimension in this activity. Need to go to people is the consequence of various places of residence location of the work and other social needs.

Decisive influence on the development of transport in general (freight and passenger) had an invention of the steam engine, which facilitated the emergence of railways, navigation by vessels set in motion by the force of steam discoveries in the art such as : construction of the first automobile powered by an internal combustion engine (1855), the invention of the internal combustion engine with spark ignition (1889), the invention of the first plane (1905) - are events that moved the evolution and diversification of transport - the emergence and road transportation of the air.

* *Corresponding author e-mail: victor.bratu@upit.ro*

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In the US, "general concept of a metropolitan area is that of a territory containing a nucleus densely populated central and adjacent communities with a high coefficient of socio-economic integration with central territory".
Current standards require that every new metropolitan area include at least: a city with less than 50 000 inhabitants and a population metropolitan 1000 000 inhabitants (1999, US Bureau of the Census), and region / Metropolitan Area (MA) It refers to an area that contains a number of autonomous administrative units, focusing on independence both metropolitan and metropolitan affairs coordination (1993, the World Conference, Tokyo).
Such an approach is essential in metropolitan areas in Romania to consider a number of programs, such as those of:

- Connecting metropolitan areas to major European transport involves certain corridors connecting metropolitan areas and also improve road transport infrastructure in the localities that make up the metropolitan area so that they correspond to European standards;
- Optimization of movement within metropolitan areas by rethinking the entire traffic system and traffic light so as to ensure smooth traffic junctions in linking localities that are in the metropolitan area;
- Improve public transport joint statement aimed at decongesting transport means and especially the realization link between localities metropolitan area;
- The development of appropriate road infrastructure for movement in metropolitan areas.

The organization of the process of passenger or passengers decisive influence both efficiency and its quality, so it is necessary to pay special attention to preparing the transport process itself and the preparation (training / education) staff engaged in -a such complex and diversified to meet the rules of the road (road discipline) and compliance of employment (work discipline) to ensure the regularity of public service transport, traffic safety, etc.
Benefits are achieved by means of passenger constitute "production process" in this field and transport cycle is defined as the number of strokes, which can be performed over the cycle. Race is the work of a middle passenger transport between the two ends of the line.

2. METHOD OF DETERMINING METROPOLITAN AREAS

Accepting the idea and the reality that metropolitan areas are made up of a city (or more if they are united space) polarizing and settlements in territory surrounding strongly linked to it, the essential question that arises is why the demarcation of borders, settlements falling within the same area.
We emphasize the interrelationship between the city and its surrounding towns because many people believe, mistakenly, metropolitan area as an area suburban, outer city which excludes overall socio-economic concentration just its core central polarizing. There are countless methodologies for determining the settlements included in the metropolitan area, ranging from the simplest, establishing the maximum distance from the city center and to the deepest, whereby borders are drawn after applying a methodology for assessing the interrelation between the central city and its outer area (lit. this area is called differently: peri-urban, sub-urban, urban driving, commuting, suburban, premetropolitana, etc.).
There are numerous concerns of geographers, economists, statisticians, sociologists, planners, etc., to determine the boundaries of metropolitan areas in some metropolitan areas are included only settlements declared suburbs or suburban localities, others are included in all settlements in the surrounding territory at a distance the central city (usually up to 50 km), but there are studies in this area is extended to 60 km. For example in the case of Bucharest, there are studies that include Olteniţa, for instance in Bucharest Metropolitan Area.
Among the most commonly used criteria for determining the limits of the metropolitan area, in addition to the distance from the city (more commonly estimated traveling time calculated in time with the means of transport most used by the population of cities in the surrounding territory) are:

- the share of people in the community who come daily to work in the metropolis (in the US, 15% of the workforce of the village is made up of commuters working in the city);
- the share of the population in these villages occupied in agricultural activities related to city (at least 75% of the population working in non-agricultural activities)
- the share of employed in agricultural production activities designed city
- the share of those trapped residents in the city, tourism potential of the town, the townspeople capitalized etc.

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Three experiences are known and Romania closer to reality.
European Centre for coordination and research in social sciences in Vienna, which has drafted the 1972-1973 model for determining the Functional Urban Region (The Functional Urban Region) as SMLA (Metropolitan Area labor standard) and MELA (Metropolitan Area economic benefits) where SMLA includes the territory where more than 15% of the economically active residents moving metropolis daily and includes MELA territory where there is population moving every day to work in the central city.

$$I_i = \frac{C_{ij}}{REA_i} \tag{1}$$

where:
I_i = extent of commuting in the area,
C_{ij} = the area commuting between i and j
REA_i = economic active residents (employed) in the area i.

In Romania, the most popular experiences for determining the boundaries of the metropolitan area is the geographer John Jordan, in the book "suburban area of Bucharest", published in 1973, the municipalities in this area following formula:

$$X_i = \frac{\frac{1}{n}\sum_{i=1}^{n} R_i S_i}{\sqrt{D}} \tag{2}$$

where
R = each of the indicators listed location "i" respectively index on non-agricultural activities, commuting, urban renewal, forest surfaces, perishable goods for city ware plant and animal intended for city and tourism potential;
S = the significance of these factors;
D = distance to the city.

Also in Romania are known to Dorel Abraham published studies in 1979 and 1991 (*Introduction to urban sociology*) based on research carried out in collaboration with sociologists, geographers and economists to determine the peri-urban area of Bucharest.
They were taken into account indicators on the following dimensions: labor, supply the city with perishable products, tourism potential, distance and spatial continuity with the polarizing city.

$$ZP = \frac{P_1A_1 + P_2A_2 + P_3\sqrt{A_3}}{\sqrt{D}} \tag{3}$$

where:
P_1, P_2, P_3 = weight for the three indicators,
A_1, A_2, A_3 = standardized values for the three indicators (labor, agriculture and tourism potential for the city)
D = distance from the city polarizing.

Applying this formula settlements at a given distance from the city polarizing, obtain a peri-urban area comprising a number of settlements, some of which is immediately suburban, suburban other places near and difference, suburban area removed; with these places in town polarizing defining regional function suburban area of the city impact: non-renewable resource consumption, air pollution, discomfort to residents.
Environmental issues are: emissions into the atmosphere, discharges into water, waste generation, contamination of soil and groundwater, use of raw materials and natural resources, energy use, emitted energy (heat, radiation, vibration) noise generation.

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3. COMPONENTS AND ORGANIZING SYSTEM PUBLIC TRANSPORT

During the race runs the following: passengers board the means of transport at the head of line mayor, ensuring passengers, transporting passengers to the destination, shipment during the intermediate stations, as appropriate, disembarking passengers at intermediate stations of the race, landing line break passengers finally resumed the direction of the return path (figure 1).

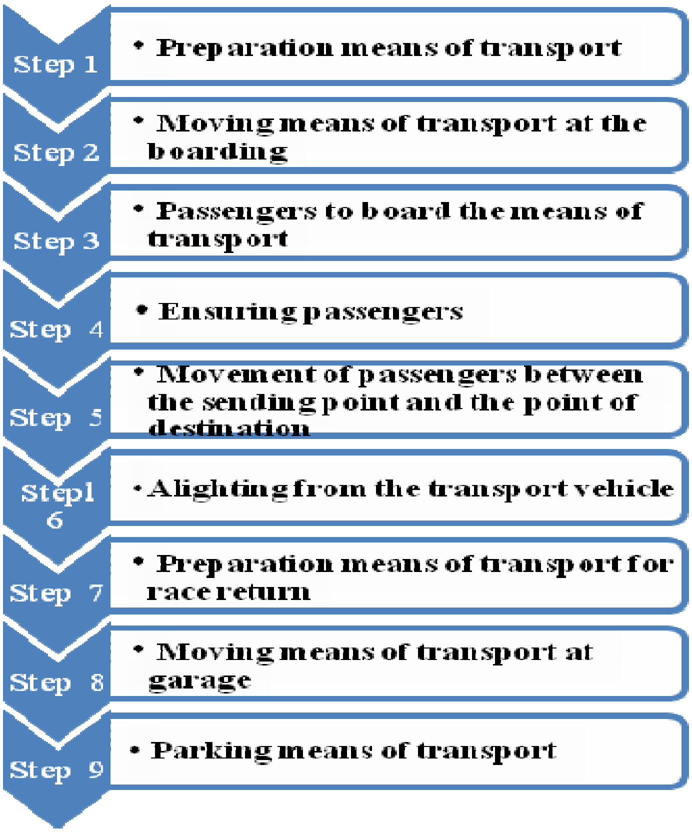

Figure 1. Flow of public passenger transport operations shown in the figure below.

Cycle (D_c) shall comprise the following times: during the preparation of the means of transport to leave for race (includes timings related technical preparation of vehicles for their adaptation to the characteristics of the journey, for action is the mechanization of operations of embarkation - landing passengers, preparation of documents (T_p) during the moving vehicle at primary boarding (end of line) (T_d); time to perform one or more races, including the time of embarkation - landing interim passengers at stations within race (T_e) time to return the vehicle after the last landing of passengers (last race) in point sidings, where due to start another cycle (T_i).
Transmission cycle is between the start of the first vehicle operations training and teaching time of conclusion of the last operation of the transport documents (after parking the vehicle):

$$D_c = T_p + T_d + T_e + Ti \tag{4}$$

Knowledge of transport operations which make up the cycle of special importance, since each of them consumes labor and materials; as these expenses are lower, the cost of transportation is less, knowing that often share the costs for conducting other operations than those for actual transportation operations is greater.
Therefore, it is a necessity reducing their weight at minimum limits in this regard, reducing downtime and eliminating as much as possible, unladen journeys we have a rational use of vehicles and transport costs minimal.
Means of transport in the public passenger transport, cycle or ride, carry journey laden and unladen journey traveled zero.

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Itinerary (circulation path) path to go through the means of transport, stations the journey, hour and minute crossing, commercial speed and downtime. Ideally, the entire public passenger transport activity to be carried out based on predetermined routes, with appropriate rhythm and to a timetable as accurately, with a maximum deviation of 30-40 seconds, but road is conducted under the influence of random factors, acting independently of the carriers, which do not always take place according to the schedule circulation timetable.
During loaded (l_l) the distance in kilometers covered by the vehicle laden (total or partial).
During unloaded (during empty) (l_u): distance in km traveled by the vehicle empty state (in during the race it does not have passengers).
During zero (l_z): the distance traveled by the means of transport from the point of loading garage, specifically primary line head and the disembarkation point, that final line head (last race) garage.
During the vehicle's total $l_t = l_u + l_l + l_z$ (km).
The races that make up the cycle of transport is based on a route (trail running) that vehicles must follow.In organization and management of public passenger transport system should be kept in mind that the road can create disturbances of public transport schedule for such situations must be clearly defined measures to be taken to ensure that disruption of traffic to be phased out in a time as small and return to the normal schedule as quickly transport.

4. METROPOLITAN AREAS MODELS

Establishment of metropolitan areas responds to needs and opportunities caused by organic development of settlements.
Urbanization European territory led to the development of metropolitan areas with interdependent localities in their area of influence, already forming metropolitan realities primary, if not practically operate metropolitan areas are designated as areas uniform, relatively independent.
Thus, "real country" urban-rural development realities calls for greater recognition and proper regulation of the "legal country" and those providing governance in Romania.
Many strategic issues of urban planning at European level cannot be treated directly than at metropolitan areas, to facilitate the development of production, trade and consumption of goods across Europe so as to avoid obstacles due to both localism excessive as centralism and nationally.
It is essentially the formation regions and metropolitan areas as poles of growth and development across Europe, included within national and transnational strategies.
Metropolitan areas can thus become bridgeheads to benefit from new ways of communication and trade relations, exchange of goods and services, know-how, as centers of capital movements as destinations for tourism and buyers.
The development of metropolitan areas facilitates integrate spatial planning at the regional level so as thereby to diminish the disparities between the center and the periphery area caused by dispersion (or marginalization of ghettoization, isolation of settlements deprived of opportunities), plan demographics, social and economic by imbalances transport, infrastructure financing, reserves space for living and commerce, the removal or mitigation of such imbalances would lead to an improvement in quality of life.
Strengthening the capacity of the settlements included in the metropolitan area to face competition from outside. Identifying common trends of development and cooperation within area as new forms of institutional organization and administration, representing the interests externally, will strengthen their capacity to cope with competition. Therefore shall ensure economic competitiveness settlements in the metropolitan area to neighboring regions. Implementation of development policies and planning through effective management and performance, the foundation of development policies to take account of market demand, but also-other requirements related to ensuring densities bearable, technical and social infrastructure (services), supply close to dwellings, the location of recreation areas etc.
Development of this policy is usually done through good cooperation between municipalities in land development, housing, infrastructure, economic development, environmental protection, human resource etc. The best performing shares are usually those related to transport, water supply, waste management and investment projects. Ensuring subsidiarity metropolitan areas by attracting public participation in various settlements in developing and implementing the strategy development of the area so that objectives and purposes planning to become more efficient.
As we know, democracy and subsidiarity cannot be applied if a decision maker is missing.

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This means that the population of metropolitan areas, even at the periphery of them can have an influence on decisions from the metropolis that would affect their lives.
Ensuring sustainable development of the metropolitan area requires a strategy that takes into account the social and economic needs of the population and is based on the scheduling option, following an impact assessment compensates best waste of resources that cannot be immediately restored or replaced, so do not limit the future development of the area.
Sustainable development implies the existence of an overview on the environment, evaluating the improvement of the quality of the urban environment, urban renewal and regeneration of resources that would diminish the negative impact of development strategies of the area and ensure development in this context can determine the areas to be protected as the natural vegetation can be assessed values of the world and village places, due to cultural and ecological values should be helped to acquire the right of national heritage.
Disadvantages of the creation of the metropolitan area to consider about possible traffic jams due to increased circulation, environmental pollution caused by urban waste, and the use of non-organic means of transport.
In the plan there is a danger of limiting government involvement Townships at a formal collaboration or lack of trust of local governments settlements in local planning bodies.
Of course, there is the danger that certain interest groups to influence local planning zonal exaggerated. These issues are more abnormal, but they must be prevented.
In practice there will be difficulties in coordination between the instruments local and regional planning, to achieve a uniform policy for the use of reserve power, reconciliation between the proposals of local governments in terms of infrastructure and equipment planning, land use by different agents mayors and maintaining ecological balance etc.
The main indicators used are: location of population and employment, displacement volume when using motorized transport and public transport modal division, the level of pollution.
The scenario "Homo Tehnicus" envisages a hierarchy depending on the degree of training at the expense of a scale considerations native, given current trends influencing society.
They won two orientations, namely:

– development of infrastructure with its own running track and some modern vehicles, such as hybrid or fuel cell, in order to comply with directives on the environment and that retrofitting the fleet;
– qualitative and quantitative development of public transport is another way of economic progress consistent with the principles of sustainable development, bowing to the principle that the production of electricity to maintain the operation of public transport problematic environment.

In this scenario assumes service quality technical considerations related to: improve comfort and increase speed, optimize energy consumption and organizational considerations that take into account the frequency, frequency, reducing subsidies.
The scenario "Homo economicus" cost- and real prices of mobility and is designed to take into account on the one hand a failure of technical progress in reducing the environmental impact and on the other hand distrust in political actions with a political character.
The scenario "Homo Contractor" is linked to the control of mobility, in terms of reducing it, and constraints on mobility are accepted, but the values of democracy and individualism in modern society remain intangible.
The scenario "Homo Politicus" assumes control of space and is based on collective processes of the organization. Transport policies transupun collective decisions in favor of the individual and public transport is sustained presence anywhere is absolutely necessary, such as residential areas with high potential for jobs and housing owners isolated or peripheral, are taxed extra.
The scenario "Homo Civis" meets several criteria for sustainability: reducing individual motorized transport, pollution, reduction of the speed of expansion of the urban area, increased travel by collective transport modes "light" trucks.
Actions resulting from the simulations scenario "Homo Civis" involving the application of measures on urban space (the space distribution of population and jobs within the meaning of their proximity to the reorganization prices land) penalty go with the car individually implement a system vignetting urban to control the town center, reducing parking paid by companies, limiting and uniform maximum speed of vehicles on all types of roads, development offer collective transport, orientation behavior towards the use of collective transport and means "light" transport, reducing the number of daily movements, especially movements between home and work.

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These actions increase the attractiveness of collective transport compared to motorized means of transport and relocation habitat trends established in the suburban area.
The simulations have highlighted the key role of reducing travel speed for individual vehicles.
In a first step measure linked to a surge of collective transport supply movements and change the partition number increases modal shift from motorized transport.
In the second stage, with an accompanying increase in the supply of urban housing that offer, the price of land in the city center it will become more compact and reduces pollutant emissions.
Restoring blade size urban spatial, social and environmental brings another positive consequence: increased travel means "light" trucks.
The minimum requirements to be met for the introduction of an effective public transport and urban environmental protection are:

– adoption of integrated and complementary measures in all areas of public transport development;
– adoption of an integrated planning policy sustainable social and environmental protection, including the promotion of non-motorized transport and restrictive measures to traffic for particular;
– collaboration of all stakeholders in promoting public transport;
– public transport are an integral part of everyday life offers many opportunities inhabitants;
– achieving promotion of public transport, and especially those that perform environmental protection by adopting long-term policy.

During this period, the world is forced to review the future role of the automobile in cities, in one of the most serious transformations of the last half century, namely how to design transport.
It is a paradox that just cars and trucks that made possible the massive urbanization are now contributing to the degradation of cities.

Metropolitan governance models existing today in Europe are classified by METREX (Network of European Metropolitan Regions and Areas) in three categories:

1. metropolitan authorities have discretion as to the social, economic, infrastructure, environment and planning or land use. These authorities are responsible to plan and implement effective and comprehensive strategies for the harmonious development of metropolitan areas.
2. Authorities, appointed or elected, provided with essential selective powers through which to plan and apply strategies to solve key problems.
3. Agencies metropolitan called complementary or bodies entrusted with responsibilities for strategic planning and advisory enforcement.

5. CONCLUSIONS

Public passenger transport process envisages and includes all operations carried out by vehicles driving people to carry passengers into space.

In organization and management of public passenger transport system should be kept in mind that the road can create disturbances of public transport schedule for such situations must be clearly defined measures to be taken to ensure that disruption of traffic to be phased out in a time as small and return to the normal schedule as quickly transport.

The organization of the process of passenger or passengers decisive influence both efficiency and its quality, so it is necessary to pay special attention to preparing the transport process itself and the preparation (training / education) staff engaged in -a such complex and diversified to meet the rules of the road (road discipline) and compliance of employment (work discipline) to ensure the regularity of public service transport, traffic safety.

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THE INFLUENCE OF THE SUSPENSION UPON THE AXLE WEIGHT DISTRIBUTION FOR HEAVY TRUCKS

Bogdan Adrian ȚOLEA[1]*, Ionuț RADU[2], Dragoș Sorin DIMA[2], Horia BELEȘ[1]

[1)] University of Oradea, Str. Universitatii Nr. 1, 410087 Oradea, Romania
[2)] Transilvania University of Brasov, Str. Politehnica, No. 1, 500024 Brasov, Romania

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Abstract: *Due to the increasing number of heavy trucks, the number of the overweight trucks have increased, and most of them, were recorded with axle overweight. This may lead to fines for the transportation companies, and to the road damage. One of the main reasons of axle over-weight is the COG displacement, which is influenced by several factors. The main objective of this paper is to determine the way that a suspension setting can influence the cargo load distribution inside a truck-semi-trailer assembly.*
In order to achieve the proposed objective, a calculation principle in regarding to the influence of the suspension setting upon the axle weight distribution for heavy trucks is presented. This calculation method was validated through experimental tests.
The results indicates that the suspension setting plays a great role in the uniformly distribution of the cargo load, and using a wrong suspension setting, overloading on a certain axle may appear.

Key-Words: *heavy trucks, suspension, axle weight distribution*

1. INTRODUCTION

The number of heavy commercial trucks is growing every day, and on this background, the number of overweight trucks has the same tendency, leading to a great challenge not only for the roadway maintenance but also for the truck weight enforcement [9].
During its movement, the heavy trucks are subjected to high external forces during braking and steering, while the cargo loading plays a great role in the truck stability [10]. The movement of the cargo, and the way it is positioned in the trailer/semi-trailer, may influence the coordinates of the COG (center of gravity). The COG modification can overload a certain axle, which not only may lead to significant fines for the transportation companies, but the road infrastructure will be damaged [7].
A study conducted in USA reported that in Tennessee more than 50% of the heavy trucks were recorded with axle overload; while for a gross overweight a percentage of 16.8% was reported [8].
Subjects such as the influence of load upon the rolling resistance respectively the fatigue assessment upon the rear axle due to the service loading were approached by J.Ejsmont et.al, respectively Zhao et. al. but, very few literature approach the subject of the influence of the suspension settings upon the axle weight distribution. The weighting process of a truck with a semitrailer is made on each axle, and the allowable values for each axle are the following: for the truck front axle is 7500 kg, the rear axle 11500 kg, while for the semi-trailer the total allowable mass is 8000 kg for each axle of the trident. In case of exceeding, the transportation company receives a fine, proportional with the exceeding [1][2][4].
In this study, will be presented a study case where we will show the influence of the suspension settings upon the overload on the driving.

2. TESTING PROCEDURE METHODOLOGY

For the measurements of the axle loadings, there can be used two types of measurement capable to determine this parameter. One of the measurements is measuring the static loading, while the second one is capable to determine the axle loading dynamically.
Both of the measurements have an allowable exceeding value of 4%.

* *Corresponding author e-mail: victor.bratu@upit.ro*

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Therefore the two procedures are as follows [6]:
- The loaded truck is stopped with approximately 4 meters before the entrance to the measuring device. He is slowly accelerated on an inclined plane, such that the truck can be lifted on a horizontal platform, in order to enter with the first axle on the device for the measurements. As soon as the first axle is positioned horizontally on the device, and a short amount of time (approximately 1 minute) passed since the first axle was positioned properly such that the cargo will be stabilized, the first measurement is taken. This procedure will be repeated for all the axles, and in the end the full report will be printed.
- In the second type of procedure, the truck is moving with a constant speed that may not exceed 5 km/h, on a special platform capable to determine the axle loadings, dynamically.

3. CASE STUDY

A loaded truck is heading to the loading measurement device, but before entering to the platform, the driver observes that due to the inclined plane that is heading to the horizontal platform, he has to lift the truck's pneumatic suspension, positioned on the rear axle, due to the fact that truck's ground clearance was too small. The main objective here is to determine the influence of the lifted pneumatic suspension. In order to determine this influence, the following sketch was drawn, using the notations:

h_{gt} - Truck COG Height
h_{cgr}- Semi-trailer COG Height
O_s - Coupling point position
h_{os}- Coupling point height
C_{gr} - semi-trailer COG location
C_{grt} - Truck COG location
C_g- Truck/trailer assembly COG location

a –Distance between the truck's front axle (axle 1) to the C_{gr}
b – Distance between the truck's rear axle (axle 2) to C_{gt}
$Z_{1,}$, Z_2 - Normal reactions of the ground, for the axle number 1 respectively axle number 2 of the truck
$Z_{r3,}$ $Z_{r4},$ Z_{r5} - Normal reactions of the ground, for the axle number 3, axle number 4 respectively axle number 5 of the semi-trailer

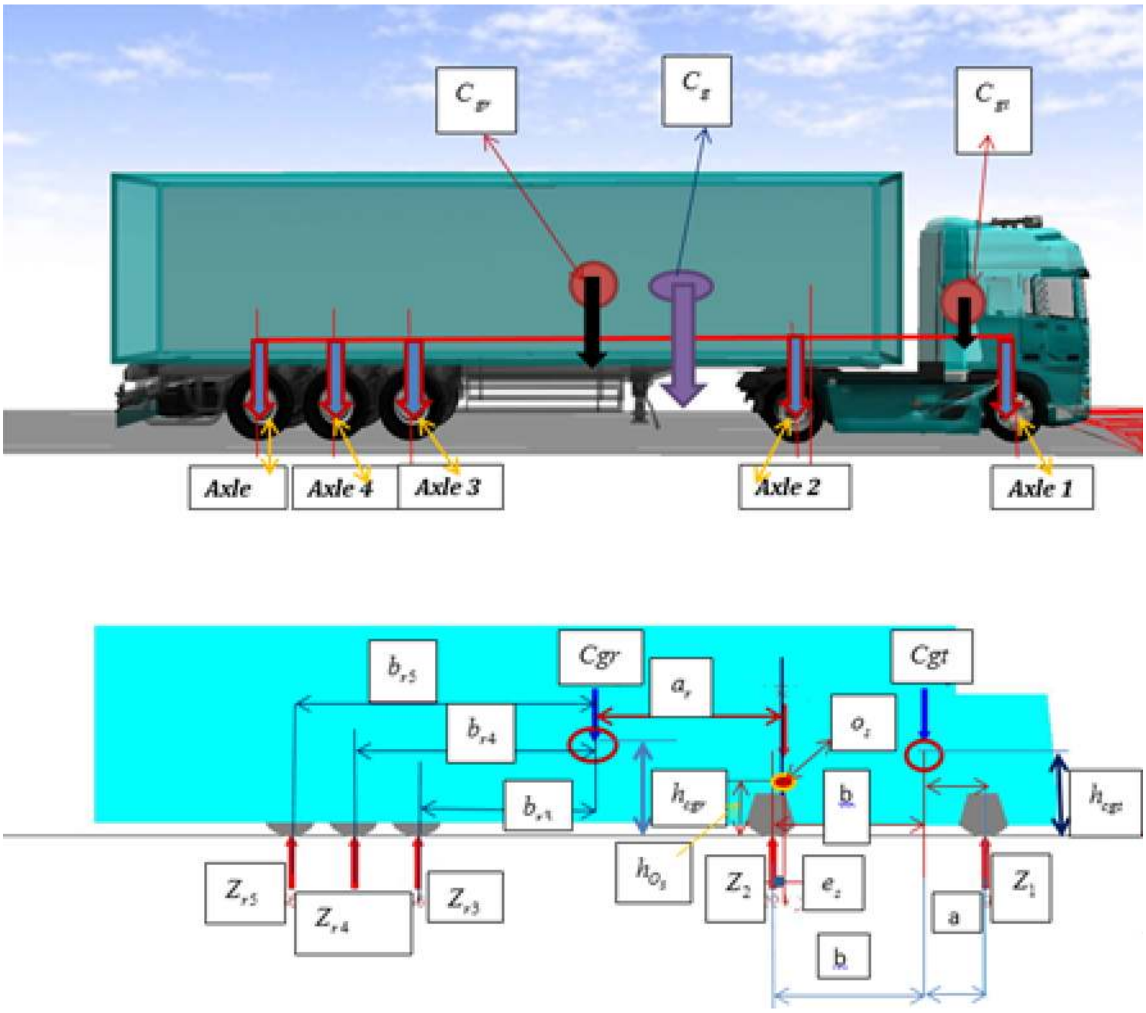

Figure 1. Sketch of the forces that are subjecting the static truck and semi-trailer

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At the semi-trailers with 3 axles, the suspension is provided with two air cushions for each axle, being fueled with compressed air.
These cushions take over and distribute uniformly the cargo's weight on each axle, ensuring the correct equilibrium, according to the approval of vehicle's technical rules set.
The used semi-trailer, has the following specifications:
M_0 = 7220 kg – Unladen mas;
M_{TMA} = 39000 kg – Maximum allowable mass;
M_{UMA} = 31780 kg – Maximum allowable payload capacity.
Distribution of maximum allowable mass per each axle:
Rear axles (axles 3, 4, 5) – M_{rs} = 24000 kg.
For the trucks with two axles, the suspension is provided with two air cushions for the rear axles, capable to take over and distribute uniformly the weight, ensuring equilibrium.
According to the approval of vehicle's technical rules set, the specifications are:
M_0' = 8456 kg – Unladen mass,
M_{TMA}' = 18600 kg – Maximum allowable mass;
M_{UMA}' = 12920 kg – Maximum allowable payload capacity.
Distribution of maximum allowable mass per each axle:
Front axle (axle 1) – M_{tf} = 7500 kg;
Rear axle (driving axle 2) – M_{ts} = 11500 kg;
M_U = 23800 kg – Maximum cargo load limit.
Knowing the value of maximum cargo limit, we were able to determine the correct load distribution on each axle.
This procedure was made by determining the percentage of each maximum load capacity on each axle, and adapting it to our maximum cargo limit (e.g. the driving axle load percentage from the maximum total truck-semi-trailer assembly mass of 40 000 kg represents 28.75%, leading to a load re-distribution using the total mass of 38680 kg, on the driving axle of 0.2875* 38680=11120.5 kg).
Using this procedure we obtained:
M_{TOTAL} = 38680 kg < 40 000 - Total mass of the truck and semi-trailer assembly;
M_{AXLE1} = 7252.5 kg, M_{AXLE2}= 11120.5 kg; M_{AXLE3} = M_{AXLE4} = M_{AXLE5} = 6769 kg
By checking the semi-trailer suspension specification provided by the manufacturer, a lifting of the suspension can reach a maximum value of 80 mm. In our calculation, we will consider a value of *H*= 70 mm (DAF, 2013).
Using the measured values presented in the Figure 2, we were able to write the momentum equilibrium equations restricted to the truck and semi-trailer assembly' COG as follows:

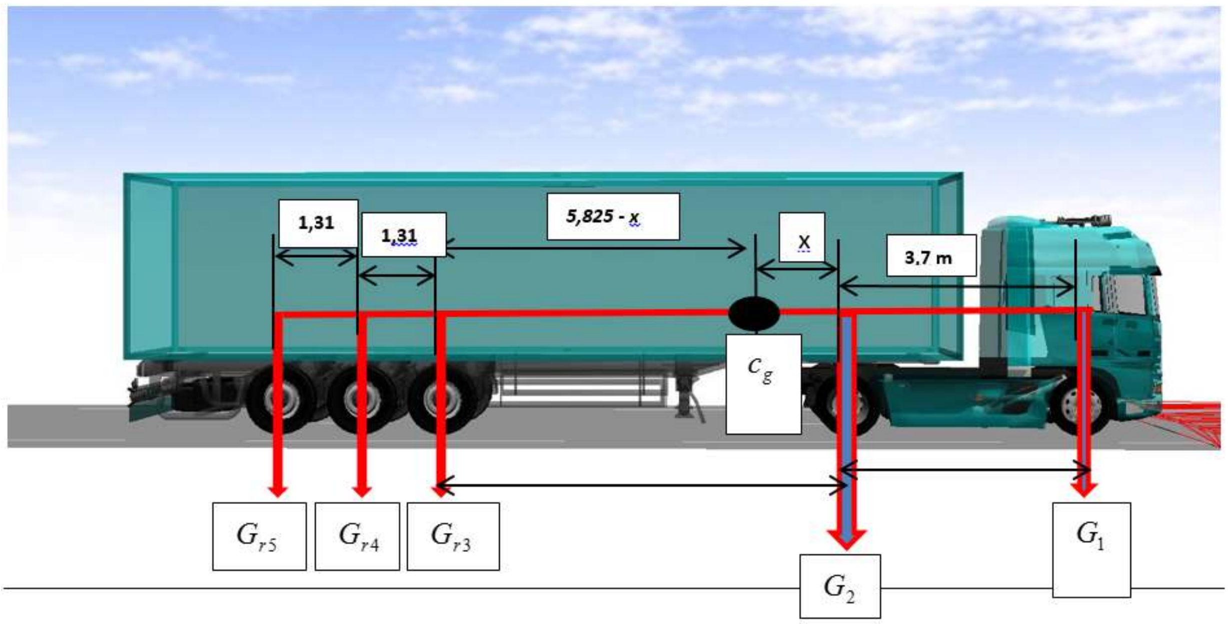

Figure 2. The sketch of the truck-semi-trailer assembly with the force distribution per axles

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$$G_1 \cdot (x+3.7) + G_2 \cdot x - G_{r3} \cdot (5.825 - x) - G_{r4} \cdot [(5.825+1.31) - x] - G_{r5} \cdot [(5.825 + 2 \cdot 1.31) - x] = 0 \qquad (1)$$

The geometric parameters taken into account in order to calculate the truck-semi-trailer assembly such as the distances between axles respectively truck's front and rear overhang were measured (Figure.2). The load distribution on each axle was determined as follows (g = 9.81 m/s^2):

$$\begin{cases} G_1 = M_{AXLE1} \cdot g = 71150N \\ G_2 = M_{AXLE2} \cdot g = 109100N \\ G_{r3} = G_{r4} = G_{r5} = 66400N \end{cases} \qquad (2)$$

Solving equation (1), the value of x was obtained as x=3.05 m, x representing the initial location of the truck-semi-trailer assembly COG location along X axis, before suspension lifting.
Distance from axle I to the COG: *D1* =3.84+x = 6.89 m
Distance from axle II to the COG: *D2* = x = 3.05 m
Distance from axle III to the COG: *D3*= 5.823 -x = 2.773 m
Distance from axle IV to the COG: *D4*= 5.823 - x + 1.31 = 4.083 m
Distance from axle V to the COG: *D3*= 5.823 -x + 2.62 = 5.393 m
In Figure 3 it is presented a sketch of the forces after the elevation caused by the suspension lifting leading to a modification of the center of gravity (denoted by C2).

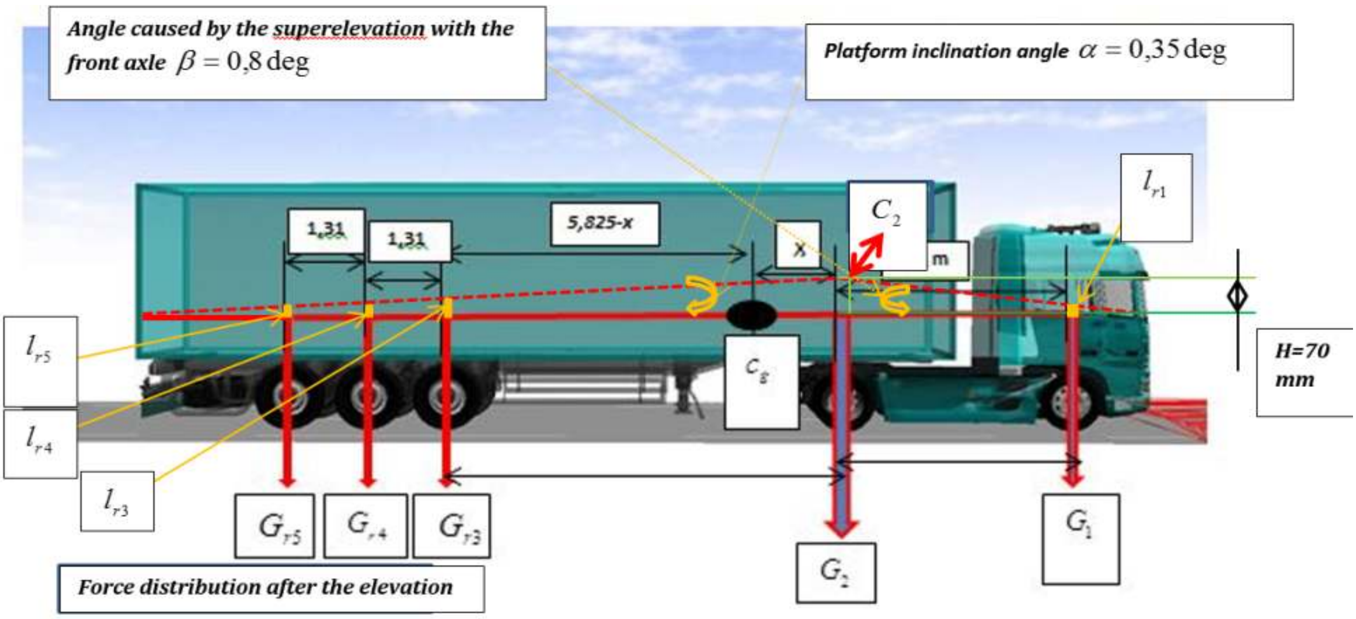

Figure 3. The sketch of the forces after the elevation occurred

The next step was to determine the values of lr1, 3, 4, 5, which represent the displacement of the axles 1, 3, 4 and 5, caused by driving axle suspension lifting.
These values were determined geometrically, using similar triangles presented in Figure 4.
Using the notations from Figure 4, the equations for the *lr* can be written:

$$\begin{cases} lr1 = Cf \cdot \dfrac{H}{A+Cf} = 0.012\,m \\ lr3 = H \cdot \dfrac{Cs + 2 \cdot D34}{Ds} = 0.033\,m \\ lr4 = H \cdot \dfrac{Cs + D45}{Ds} = 0.025\,m \\ lr5 = H \cdot \dfrac{Cs}{Ds} = 0.017\,m \end{cases} \qquad (3)$$

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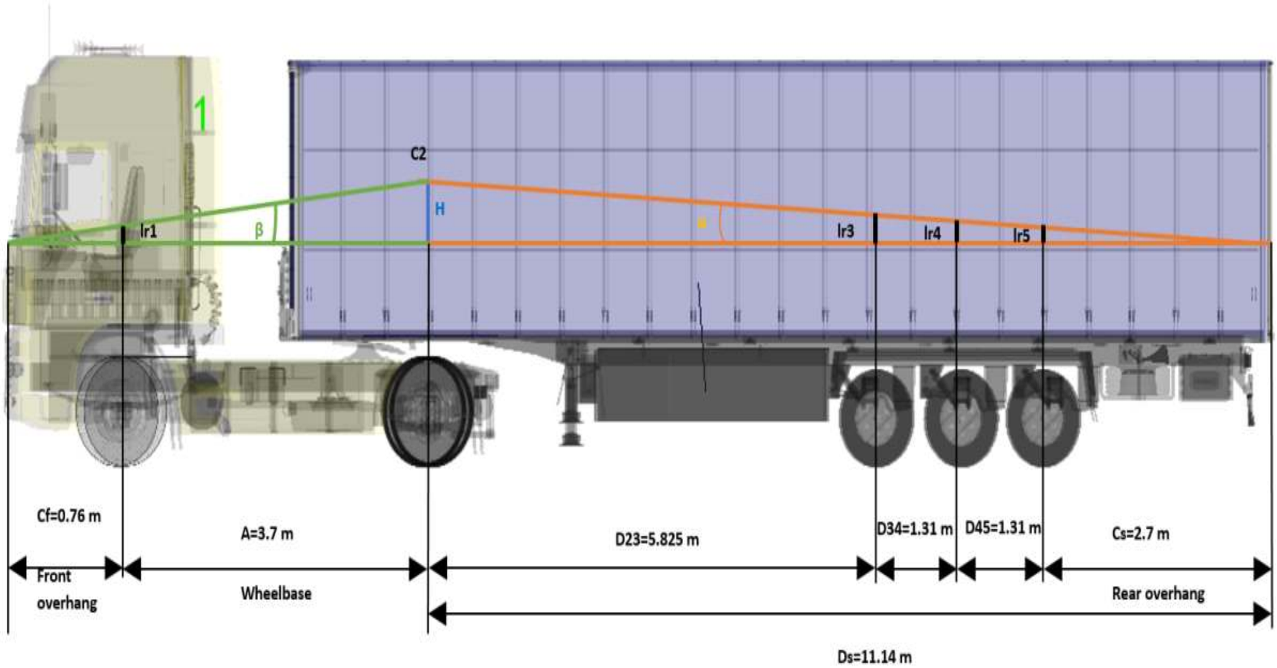

Figure 4. The triangles described by the suspension lifting

In this case, by writing the moment equilibrium equation in point C2, we obtained the following equation in function of *α*:

$$H \cdot F_{1C_2} = (H - l_{r3}) \cdot G_{r3} \cdot \sin\alpha + (H - l_{r4}) \cdot G_{r4} \cdot \sin\alpha + (H - l_{r5}) \cdot G_{r5} \cdot \sin\alpha \tag{4}$$

The moment equilibrium equation can be written in function of *β* in point C2, as follows:

$$M_{2C_2} - M_{G1} = 0 \tag{5}$$

where:
M_{2C2} – represents the moment on axle 2 applied in the point C2,
M_{G1} – represents the moment on axle 1 caused by weight G_1

$$H \cdot F_{2C_2} = (H - l_{r1}) \cdot G_1 \cdot \sin\beta \tag{6}$$

By adding relation 3 with 5, it is obtained:

$$F_{1C_2} + F_{2C_2} = \frac{1}{H} \cdot [(H - l_{r3}) \cdot G_{r3} \cdot \sin\alpha + (H - l_{r4}) \cdot G_{r4} \cdot \sin\alpha + (H - l_{r5}) \cdot G_{r5} \cdot \sin\alpha + (H - l_{r1}) \cdot G_1 \cdot \sin\beta] \tag{7}$$

Therefore, knowing parameters such as *α, β, H, Gr* we determined the overloading of the driving axle:

$$F_{1C_2} + F_{2C_2} = 2007\,N \tag{8}$$

- overloading *g* on the axle 2 (driving axle)

Representing an overloading mass on the measuring device of M_{ov} = 204.58 kg.

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By comparing the weight distribution on axle 2, in normal equilibrium conditions with G2, results:

$$\frac{F_{1C_2} + F_{2C_2}}{G_2} = \frac{1915}{109100} \cong 1.84\% \tag{9}$$

Therefore the influence of the suspension lifting with 7 cm in this case, can lead to an increasing of weight distribution on the driving axle of 1.84 %, which may cause significant fines for the transportation companies.

5. RESULTS AND DISCUSSION

In order to test the influence of the suspension settings upon the axle load distribution, there were performed both types of measurements on a heavy truck, with the same cargo.
In one of the test, the truck had the air cushion set at 0 level, while for the second test the driving axle suspension air-cushion was lifted.
For the first test the static measurement was performed, while for the second one we used the dynamic measurement. The data obtained is summarized in the table 1.

Table 1.
The measurements obtained for both tests

Load measured	Test 1 - Static measurement [kg]	Test 2 - Dynamic measurement - Lifted suspension [kg]	Maximum allowable value [kg]	Maximum allowable value with 4% exceeding value [kg]	Exceeding value - Test 1 [kg]	Exceeding value - Test 2 [kg]
Total mass	38680	38680	40000	-	0	0
Mass on axle 1	7510	7780	7500	7800	0	0
Mass on axle 2	11800	11980	11500	11960	0	20
Mass on axle 3	6550	6340	8000	8320	0	0
Mass on axle 4	6490	6220	8000	8320	0	0
Mass on axle 5	6330	6360	8000	8320	0	0

It can be observed from Table 1, that in the first test the cargo load on each axle is within the allowable values, while for the second test (suspension lifted) the exceeding value of 20 kg on the driving axles occurs and represents an exceeding of 1.53% in comparison with the static test measurement, where the suspension's air cushion was at 0 level.
Was observed that in the test where the suspension was lifted, the masses on the semi-trailer trident were reduced, and the main reason of this is that by moving the COG backwards along the *X* axis, the length of the force lever arm is reduced, leading to a reduced torque value on the trident.
Using the presented calculation principle, the overloading on the driving axle can be determined using as input value the measured driving axle load of 11800 kg, when the suspension setting was set to 0 level, and we can determine the height of the suspension setting for the second measurement (dynamic measurement), where the overloading occurred.
By performing this calculation, we obtained a value for the suspension air cushion height, *H* of 61 mm, representing an overloading mass of 180 kg.
In order to obtain a diagram of the overloading force in function of the suspension displacement, was developed a Mathcad program. In this program, the suspension displacement *(H)* was used as a range variable, with values from 10-80 mm, with a step increment of 10.
The obtained diagram of the overload force in function of the suspension displacement is presented in Figure 4. Another diagram that was extracted was the one including the overload percentage of the driving axle in function of the suspension displacement. This diagram is presented in Figure 5.
It can be observed that a small suspension modification leads to significant overloading on the truck's driving axle.

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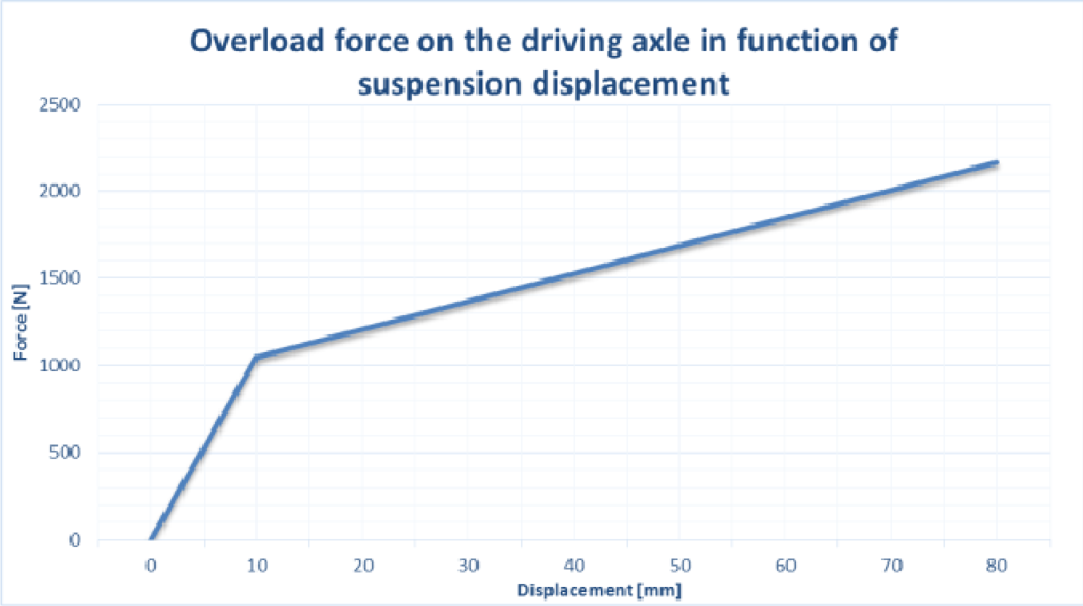

Figure 4. Overload force on the driving axle in function of suspension displacement

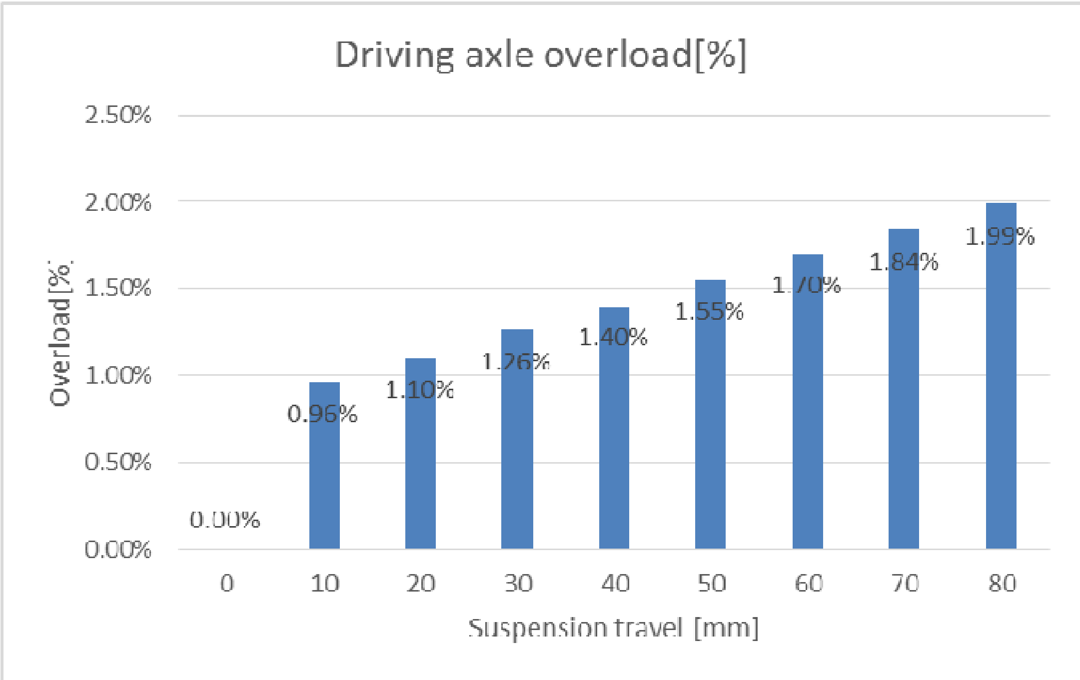

Figure 5. Driving axle overload in function of suspension displacement

At a modification of 10 mm, we can discuss of an overload of nearly 1%, considering the reference value the maximum allowable mass on the driving axle (11500 kg), while for the case where the suspension is set at a height of 80 mm, the overload is doubled.
The main reason of this phenomena, is the fact that the suspension modification leads to a COG modification, and the higher is the suspension lifted, the greater is the truck-semitrailer COG modification

6. CONCLUSION

This paper presents a calculation principle, meant to determine the influence of a truck's pneumatic suspension upon the load distribution, and the way that this influences the overloading on the driving axle. By analyzing the data obtained in the case study, we can argue that the suspension setting plays an important role on the axle load distribution.

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By lifting the air cushion provided by the driving axle suspension with 80 mm, can lead to the axle overload of 2%.
The calculation method was validated through measurement tests, and it is an accurate method to predict the overloading on the driving axle in function of the air cushion suspension settings.
Transportation companies can easily receive fines in case of the drivers will not set the suspension properly before they reach the cargo weighing device.
The suspension setting represent an important factor to distribute correctly the cargo load, but may be influenced by other factors. Future research will be conducted on the suspension modification in case the cargo will be hanging of the semi-trailer sealing, and during the vehicle motion, this cargo will produce a pitch movement of the truck/semi-trailer assembly.

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Standing as it does at the cross-roads of Physics, Chemistry, Mechanics, Engineering Design and Materials Sciences, automotive engineering is experiencing considerable growth as a result of recent technological advances. The Romanian Journal of Automotive Engineering, by providing an international medium of communication, is encouraging this growth and is encompassing all aspects of the field from thermal engineering, flow analysis, structural analysis, modal analysis, control, vehicular electronics, mechatronics, electro-mechanical engineering, optimum design methods, ITS, and recycling. Interest extends from the basic science to technology applications with analytical, experimental and numerical studies.

The emphasis is placed on contribution that appears to be of permanent interest to research workers and engineers in the field. If furthering knowledge in the area of principal concern of the Journal, papers of primary interest to the innovative disciplines of „Automotive Technology, Science and Engineering” may be published.
No length limitations for contributions are set, but only concisely written papers are published. Brief articles are considered on the basis of technical merit. Discussions of previously published papers are welcome.

Notes for contributors
Authors should submit an electronic file of their contribution to the ***Production office:*** www. siar.ro. All the papers will be reviewed and assessed by a series of independent referees.

The Journal of the Society of Automotive Engineers of Romania
www.ro-jae.ro www.siar.ro
ISSN 2457 – 5275 (Online, English)
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RoJAE
Romanian
Journal of Automotive Engineering

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The Scientific Journal of SIAR
A Short History

The engineering of vehicles represents the engine of the global development of the economy.
SIAR tracks the progress of the automotive engineering in Romania by: the development of automotive engineering, the development of technologies, and road transport services; supporting the work of the haulers, supporting the technical inspection and of the garage; encouraging young people to have a career in the automotive engineering and road haulage; stimulation and coordination of activities that promote an environment that is suitable for continuous education and improving of knowledge of the engineers; active exchange of ideas and experience, in particular for students, master students, PhD students, and young engineers, and dissemination of knowledge in the field of automotive engineering; cooperation with other technical and scientific organizations, employers' and socio-professional associations through organization of joint actions, of mutual interest.
By the accession to FISITA (International Federation of Automotive Engineering Societies) since its establishment, SIAR has been involved in achieving an overall professional community that is homogeneous in competence and performance, interactive, dynamic, and competitive at the same time, oriented towards a balanced and friendly relationship between people and the environment; this action will be constituted as a challenge worthy of effort and recognition.
The insurance of a favorable framework for the initiation and the development of cooperation of the specialists in this field of activity allows for an efficient and easy exchange of information, specific knowledge and experience; it supports the cooperation between universities and between research centers and industry; it speeds up the process of implementing the new technologies, it simplifies the identification of training and specialization needs of the personnel involved in the engineering of motor vehicles, transport, and road safety.
In order to succeed, ever since its founding, SIAR has considered that the stress should be put on the production and distribution, at national and international level, of a publication of scientific quality.

Under these circumstances, the development of the scientific magazine of SIAR had the following evolution:
1. RIA – Revista inginerilor de automobile (in English: ***Journal of Automotive Engineers***)
ISSN 1222 – 5142
Period of publication: 1990 – 2000 | Format: print, Romanian
Frequency: Quarterly | Electronic publication on: www.ro-jae.ro
Total number of issues: 30 | Type: Open Access
The above constitutes series nr. 1 of SIAR scientific magazine.

2. Ingineria automobilului (in English: ***Automotive Engineering***)
ISSN 1842 – 4074
Period of publication: as of 2006 | Format: print and online, Romanian
Frequency: Quarterly | Electronic publication on: www.ingineria-automobilului.ro
Total number of issues: 47 | Type: Open Access
(including the June 2018 issue)
The above constitutes series nr. 2 of SIAR scientific magazine (Romanian version).

3. Ingineria automobilului (in English: ***Automotive Engineering***)
ISSN 2284 – 5690
Period of publication: 2011 – 2014 | Format: online, English
Frequency: Quarterly | Electronic publication on: www.ingineria-automobilului.ro
Total number of issues: 16 | Type: Open Access
(including the December 2014 issue)
The above constitutes series nr. 3 of SIAR scientific magazine (English version).

4. *Romanian Journal of Automotive Engineering*
ISSN 2457 – 5275
Period of publication: from 2015 | Format: online, English
Frequency: Quarterly | Electronic publication on: www.ro-jae.ro
Total number of issues: 14 (June 2018) | Type: Open Access
The above constitutes series nr. 4 of SIAR scientific magazine (English version).

Summary – on June 30, 2018

Total of series:	*4*
Total years of publication:	*24 (11=1990 – 2000; 13=2006-2018)*
Publication frequency:	*Quarterly*
Total issues published:	*77 (Romanian), out of which, the last 30 were also published in English*