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MATHEMATICAL MODELING OF CHANGE OF STEERING WHEEL TOE-IN ANGLES WHEN VEHICLE MOVES IN A CURVED DIRECTION

V.G. Verbitsky [1] **A.V. Shcherbyna** [2] **O.M. Artyukh** [2] **O.V. Dudarenko** [2]
V.V. Kuzmin [3] **D.P. Ruban** [4] **A.V. Fasoliak** [5]

1. Software for Automated Systems Department, Engineering Institute, Zaporizhzhia National University, Zaporizhzhia, Ukraine, oxsidan@ukr.net
2. Automobiles Department, Zaporizhzhia Polytechnic National University, Zaporizhzhia, Ukraine avshcherbinaav@gmail.com, artyukh74@gmail.com, ovdudarenko@gmail.com
3. Social Work and Psychology Department, Zaporizhzhia Polytechnic National University, Zaporizhzhia, Ukraine kuzmin2v@gmail.com
4. Motor Vehicle Transport and Social Security Department, Open International University of Human Development, Kiev, Ukraine, ruban_dimon@ukr.net
5. Higher Mathematics Department, Zaporizhzhia Polytechnic National University, Zaporizhzhia, Ukraine antfas@ukr.net

Abstract- Scientists from all over the world have long agreed that vehicle tires are a serious environmental problem for the world around us. One way to reduce the harmful effects from tires to the environment is to exploit a vehicle with an optimum wheel toe-in angles setting. The front steering wheels of the vehicle, with any front suspension design, are fixed at certain angles of inclination in the vertical and horizontal planes, i.e. so-called camber and toe-in angles. These angles are provided to improve the vehicle's driving stability, ease of steering and have a significant impact on tire wear. More specifically, the toe-in angles of the wheels is the difference in size between the extreme front and rear points of the wheels at the level of their centers. The toe-in is necessary to prevent slippage caused by camber angle and to compensate for backlash in the steering joints and wheel bearings. Toe-in angles of wheel more or less than normal causes increased tire wear. Thus, proper setting toe-in angles of the wheel and its adjustment during the vehicle's exploitation is extremely necessary. The article presents a mathematical model of a vehicle, which allows us to study the dynamics of changes in wheel toe-in angles when the vehicle moves along a curvilinear trajectory. This mathematical model is used to determine the optimal toe-in angles of the steered wheels for a vehicle at different angles of rotation. The optimal toe-in angles allow to improve ecological and operational performance of the vehicle by reducing tire wear and fuel consumption.

Keywords: Vehicle, Wheel, Ecology, Toe-in Angles, Tire Wear.

1. INTRODUCTION

For many decades, all developed countries have been struggling with harmful emissions into the environment. As we know, one of the biggest polluters of the environment is the vehicle. In this case, if previously the biggest source of pollution in the vehicle was the engine, namely the exhaust gases from the exhaust system, now it is the tires and brake pads of the vehicle. This has happened due to the gradual and systematic strengthening of environmental standards for the content of harmful substances in exhaust gases, namely Euro 1 to Euro 6 standards.

Tires emit 5.8 grams of pollutants per kilometer, more than a thousand times the emissions of an internal combustion engine, which complies with the sixth environmental class, according to Emissions Analytics Company. Emission from the average modern engine is about 4.5 milligrams per kilometer [1]. Micro dust from tire wear is a serious modern environmental problem. This problem is exacerbated by the popularity of large and heavy vehicles, such as SUVs, as well as the growing market of electric vehicles, which are heavier than standard vehicles because of their batteries. Environmental problems have long gone beyond the borders of individual countries and can only be solved at the international level. Therefore, the problem of environmental pollution is dealt with at the level of the UN and other international organizations.

Thus, research aimed at reducing tire wear is a necessary and relevant direction. One of the ways to reduce tire wear is the correct selection of wheel mounting angles, in particular, their toe-in angles [2, 3, 4]. It should be noted that wheel toe-in angles have an impact on many of the vehicle's operating characteristics. In many works there is a direct correlation between the toe-in angles and tire wear and accordingly the resource of the tires [5, 6, 7].

Correctly selected toe-in angles wheels reduce the resistance to wheel rolling during the movement of the vehicle and, accordingly, tire wear and fuel consumption are reduced. When the wheel toe-in angle is incorrect, the loads on the suspension and the vehicle body increase [8, 9]. This generates a growth of sign-variable loads on the vehicle body and intensive formation of fatigue failure zones. Therefore, as a result of increased loads, the resource of the vehicle body also decreases. In [10], studies have shown that the toe-in angles of steering wheels change when the vehicle is moving. At the same time, the magnitude of change in the toe-in angles depends on the speed of the vehicle.

In order to keep the toe-in angles constant while the vehicle is moving, it is necessary to use automatic systems for adjusting the angles of wheel toe-in [11, 12, 13]. Such systems make it possible to set and maintain optimal toe-in angles while the vehicle is moving [12, 14, 15, 16]. The mathematical model presented in [10], makes it possible to obtain the optimal toe-in angles when the vehicle moves in a straight line at different speeds. It should be noted that most of the time the vehicle moves along a curvilinear trajectory with different angles of rotation of the steered wheels. Therefore, a mathematical model was developed to obtain optimal angles of wheel toe-in when the vehicle moves along a curvilinear trajectory.

2. RESEARCH TOE-IN ANGLES WHILE CURVED DRIVING

The analytical study of a vehicle, which is a mechanical system with a large number of mutually related elements, requires the creation of a mathematical model that provides a solution of the set tasks with the required accuracy and without excessive complication of the mathematical apparatus. The choice of a calculation model, which to some extent will correspond to the real vehicle, and the number of connections it takes into account is largely determined by the purpose of the research.

2.1. Assumptions and Simplifications of a Mathematical Model

For mathematical description of plane-parallel motion of a vehicle, we introduce the following assumptions and simplifications:

1. The vehicle is moving on a flat, horizontal surface.
2. The basic trajectory is that of the centre of mass.
3. The system maintains a constant value for the longitudinal component of the velocity for the vehicle's centre of mass.
4. Turning the wheel counter clockwise is considered a positive value, and vice versa, clockwise is considered a negative value.
5. The angles of rotation of the outer and inner wheels are equal to each other.
6. Gyroscopic moments and unbalanced moments of rotating parts are not considered.
7. There is no backlash in the steering joints.
8. The tracks of the front and rear axles are equal to each other.

For calculations, we will use a special coordinate system called the Koenig axis. The Koenig axis is a coordinate system that moves translationally, and its origin coincides with the investigated mass center of the system. Thus, we fix the *XYZO* coordinate system on the vehicle body so that its origin is in the center of mass of the vehicle, the *X* axis is directed along the longitudinal axis of the vehicle, and the *OZ* axis is perpendicular to the supporting surface and is directed upward (Figure 1).

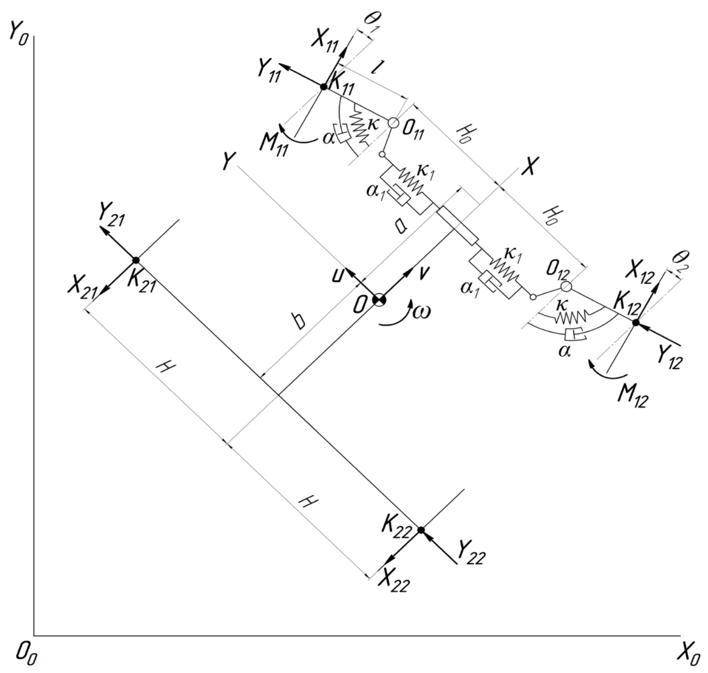

Figure 1. Vehicle calculation diagram

2.2. Mathematical Model Motion of a Vehicle

Taking into account the above-mentioned assumptions and the vehicle calculation diagram shown in Figure 1, we write the system of Equations (1) for the dynamics of plane-parallel motion of a wheeled vehicle containing six equations. Moreover, the first four equations are the equation of perturbed motion of the wheel module, namely, the equation of rotational motion relative to the turn center and the equation of transverse displacement of the center of rotation of the wheel, and the last two equations take into account the inertia of the entire vehicle.

$$\ddot{\theta}_1 = \frac{-\dfrac{l \times X_{11}^*}{\cos\theta_1} - M_{11} - \kappa \times (\theta_1 - \theta_{10}) - \alpha \times \dot{\theta}_1}{I_1 + m_1 \times l^2}$$
$$\ddot{y}_1 = -\frac{1}{m_1}(\kappa_1 \times y_1 + \alpha_1 \times \dot{y}_1 + Y_{11} \times \cos\theta_1 +$$
$$+ X_{11} \times \tan\theta_1 + m_1 \times l \times \cos\theta_1 \times \dot{\theta}_1^2)$$
$$\ddot{\theta}_2 = \frac{\dfrac{l \times X_{12}^*}{\cos\theta_2} - M_{12} - \kappa \times (\theta_2 - \theta_{20}) - \alpha \times \dot{\theta}_2}{I_1 + m_1 \times l^2}$$
$$\ddot{y}_2 = -\frac{1}{m_1}(\kappa_1 \times y_2 + \alpha_1 \times \dot{y}_2 + Y_{12} \times \cos\theta_2 + \qquad (1)$$
$$+ X_{12} \times \tan\theta_2 - m_1 \times l \times \cos\theta_2 \times \dot{\theta}_2^2)$$
$$m \times (\omega \times v + \dot{u}) = X_{11} \times \tan\theta_1 + X_{12} \times \tan\theta_2 +$$
$$+ Y_{11} \times \cos\theta_1 + Y_{12} \times \cos\theta_2 + Y_{21} + Y_{22}$$
$$J \times \dot{\omega} = -X_{11} \times H_0 + X_{11} \times \tan\theta_1 \times a + X_{12} \times H_0 + X_{12} \times$$
$$\times \tan\theta_2 \times a + Y_{11} \times \cos\theta_1 \times a + Y_{11} \times \sin\theta_1 \times H_0 + Y_{12} \times$$
$$\times \cos\theta_2 \times a - Y_{12} \times \sin\theta_2 \times H_0 - b \times (Y_{21} + Y_{22}) = 0$$

In the system of Equations (1) the following designations are taken:

X_{11}^*, X_{12}^* are longitudinal component of the force (projection on the OX axis) acting, respectively on the left and right wheel;
Y_{11}, Y_{12} are transverse force acting, respectively on the left and right wheel;
M_{11}, M_{12} are the stabilizing moment of the withdrawal forces on the left and right wheel, accordingly;
l is wheel trunnion length;
θ_1, θ_2 are the current turning angle of left and right wheel;
θ_{10}, θ_{20} are left and right wheel turning angle produced by steering;
$\dot{\theta}_1$, $\dot{\theta}_2$ are angular velocity of the left and right wheels;
α is angle damping;
κ is torsional stiffness of the suspension;
I_1 is axial moment of inertia for the one wheel;
m_1 is mass of the one wheel;
y_1, y_2 are transverse offset of the left and right wheel pivot axis;
$\dot{y}_1$, $\dot{y}_2$ are the transverse offset velocity of turn center for the left and right wheel;
α_1 is damping by the transverse offset of turn center for the left and right wheel;
κ_1 is transverse stiffness of the suspension by offset of turn center for the left and right wheel;
m is mass of the vehicle;
J is vehicle axial moment of inertia;
ω is angular velocity of the vehicle mass center about the vertical axis;
v is straight-line speed of the vehicle mass center;
u is sideline speed of the vehicle mass center;
$\dot{u}$ is sideline acceleration of the vehicle mass center;
a is position of the vehicle mass center from the front axle;
b is position of the vehicle mass center from the rear axle;
H_0 is distance from the longitudinal axis of the vehicle to the wheel-spin center.

The transverse forces and withdrawal moments were determined on the basis of empirical relationships as functions of the withdrawal angles. In the study, the dependences of the withdrawal force (2) and the withdrawal moment (3) are used in accordance with the results in the research [10, 17]:

$$Y_{ij} = k_i \cdot \delta_{ij} \sqrt{1 + \left(\frac{k_i \cdot \delta_{ij}}{\phi_{ij} \cdot Z_{ij}}\right)^2} \qquad (2)$$

where, k_i is withdrawal resistance coefficient of the ith axis of the vehicle;
δ_{ij} is withdrawal lateral angle of the jth wheel of the ith axle of the vehicle;
φ_{ij} is coefficient of traction of the wheel with the road;
Z_{ij} is vertical reaction of the jth wheel of the ith axis of the vehicle, at the point of contact of the wheel with the road.

$$M_{1j} = \frac{\sigma_{1j} \times \delta_{1j}}{39122.65 \times \delta_{1j}^{\ 4} + 71.45 \times \delta_{1j}^{\ 2} + 1} \qquad (3)$$

where, σ_{1j} is coefficient determining the linear stabilizing moment of the jth wheel.

The wheel traction coefficient for the rear axle wheels is taken as the maximum value for the given bearing surface, and for the front axle wheels we calculate by the Equation (4).

$$\phi_{1j} = \phi_{\max} \cdot \sqrt{1 - \left(\frac{X_{1j}}{Z_{1j}}\right)^2} \qquad (4)$$

Let us determine the withdrawal angles at the front and rear wheels of the vehicle through the lateral and longitudinal components of the speed of the vehicle mass center, as well as through its angular speed relative to the vertical axis, according to Figure 1, as follows:

$$\delta_{11} = \arctan\left(\frac{(-H_0 \times \omega + v)\sin\theta_1 - (a \times \omega + u)\cos\theta_1}{-l \times \dot{\theta}_1 + (-H_0 \times \omega + v)\cos\theta_1 + (a \times \omega + u)\sin\theta_1}\right) \qquad (5)$$

$$\delta_{21} = \arctan\left(\frac{-u + \omega \times b}{v - \omega \times H}\right) \qquad (6)$$

$$\delta_{12} = \arctan\left(\frac{(H_0 \times \omega + v)\sin\theta_2 - (a \times \omega + u)\cos\theta_2}{l \times \dot{\theta}_2 + (H_0 \times \omega + v)\cos\theta_2 + (a \times \omega + u)\sin\theta_2}\right) \qquad (7)$$

$$\delta_{22} = \arctan\left(\frac{-u + \omega \times b}{v + \omega \times H}\right) \qquad (8)$$

The vertical reactions of the wheels are determined by Equations (9) and (10).

$$Z_{1j} = \frac{m \cdot g \cdot b}{2 \cdot (a+b)} - \frac{X_{ij}^* \cdot h_m}{a+b} \mp \frac{m \cdot v \cdot \omega \cdot b \cdot h}{2 \cdot H \cdot (a+b)} \quad (9)$$

$$Z_{2j} = \frac{m \cdot g \cdot a}{2 \cdot (a+b)} - \frac{X_{ij}^* \cdot h_m}{a+b} \mp \frac{m \cdot v \cdot \omega \cdot a \cdot h}{2 \cdot H \cdot (a+b)} \quad (10)$$

where, m is mass of the vehicle;
a is position of the center mass from the front axle;
b is position of the center mass from the rear axle;
h_m is height of the metacenter;
H is half of the vehicle track;
± is minus is taken for the left wheel, plus for the right wheel.

Thus, the mathematical model of a front-wheel-drive vehicle is described by a system of Equations (1), which allows us to determine the wheel toe-in angles for curvilinear motion.

The analysis of system dynamics was based on numerical integration of differential equations using Maple 18 software.

2.3. Determination of Dynamic Toe-in Angles in Curvilinear Motion

The dynamic toe-in angles in curvilinear motion were determined based on the fact that the angle of rotation of the wheels set by the driver using the steering control differs from the final angle of rotation of the wheels due to their installation with some initial angle of toe-in ε_0, as shown in Figure 2.

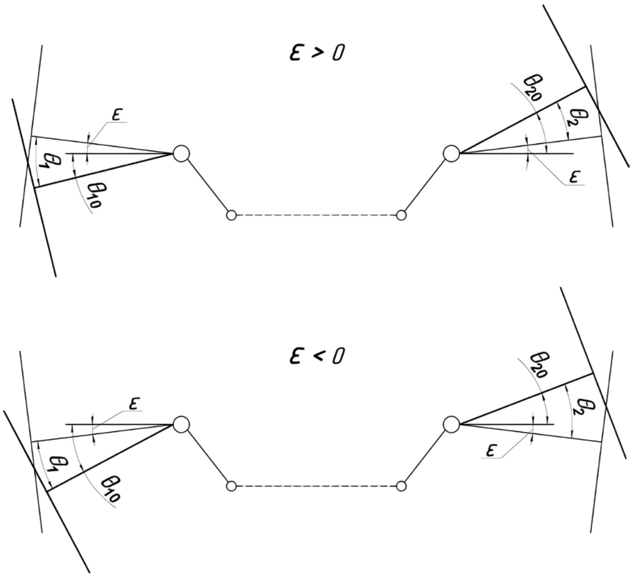

Figure 2. Principle diagram of the car wheels turn taking into account the initial angle of toe-in

Taking into account the diagram in Figure 2, we can write:
when, $\varepsilon_0 > 0$

$$\theta_{10} = \theta_1 - \varepsilon_0, \quad \theta_{20} = \theta_2 + \varepsilon_0 \quad (11)$$

when, $\varepsilon_0 < 0$

$$\theta_{10} = \theta_1 + \varepsilon_0, \quad \theta_{20} = \theta_2 - \varepsilon_0 \quad (12)$$

Using Equations (11) and (12) and data obtained by integrating the system of Equation (1), we can obtain dynamic wheel toe-in angles, applying the following solution scheme. Consider the case where the wheels are mounted with toe-out, i.e. $\varepsilon_0 < 0$. Taking into account the previously accepted simplification that the left and right wheel angles are equal to each other, let us introduce some average angle of their rotation θ_0 in the calculation. Considering that, the initial toe-in angle has a negative sign, we can write:

$$\theta_{10} = \theta_0 - \varepsilon_0, \quad \theta_{20} = \theta_0 + \varepsilon_0 \quad (13)$$

Then, when integrating the system of Equation (1) at the initial moment of time, the angles of rotation of the wheels will be equal:

$$\theta_1(0) = \theta_{10}, \quad \theta_2(0) = \theta_{20} \quad (14)$$

or

$$\theta_1(0) = \theta_0 - \varepsilon_0, \quad \theta_2(0) = \theta_0 + \varepsilon_0 \quad (15)$$

Then for wheel rotation angle at some point of time we can write;

$$\theta_1(t) = \theta_0 - \varepsilon_1(t), \quad \theta_2(t) = \theta_0 + \varepsilon_2(t) \quad (16)$$

From Equation (16) dynamic toe-in angles of left and right wheels will be equal:

$$\varepsilon_1(t) = -(\theta_1(t) - \theta_0), \quad \varepsilon_2(t) = \theta_2(t) - \theta_0 \quad (17)$$

Thus, solving the system of Equations (1), taking into account expression (17), we can study the dynamics of changes in wheel toe-in angles during curvilinear motion of the vehicle (Figure 3).

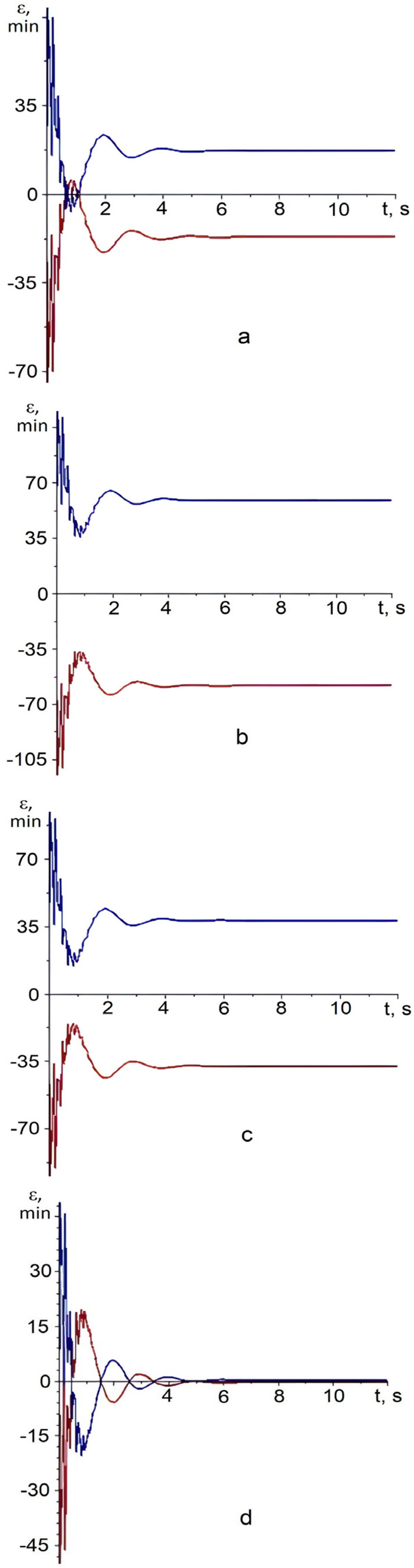

Figure 3. Change of wheel toe-in angles when the vehicle moves at a speed of V = 90 km/h on a circle of radius R = 90 m, with different initial toe-in angles, a. ε_0 = -20 min, b. ε_0 = 20 min, c. ε_0 = 0 min, d. ε_0 = -36 min

3. CONCLUSIONS

The results obtained in Figure 3, clearly show how different initial angles of wheel toe-in change when driving a front-wheel drive vehicle in a circle. For example, Figure 3d shows that at the initial angle of toe-in ε_0 = -36 min. the wheels will have a current angle of toe-in equal to 0 (ε = 0 min) when the vehicle moves. It is at this current angle of toe-in that the wheels of the vehicle move with the least rolling resistance, and therefore with the least tire wear and fuel consumption. Based on the data obtained, the optimal toe-in angles of the vehicle during its curvilinear motion with different angles of steerable wheels θ were determined (Figure 4).

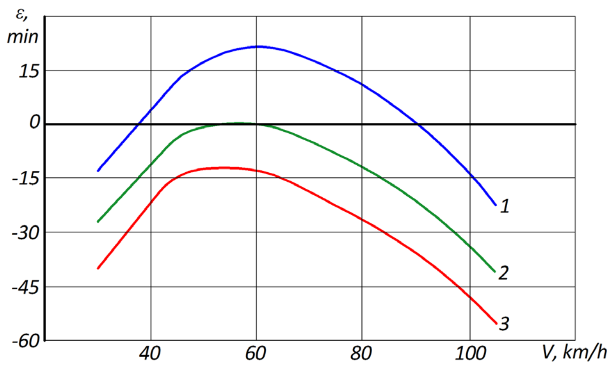

Figure 4. Optimal toe-in angles for curvilinear motion of the vehicle 1. θ=10°, 2. θ=20°, 3. θ=30°

This data, combined with automatic systems to adjust wheel toe-in angles while the vehicle is in motion, can improve performance, namely, reduce the rolling resistance of the wheels. The proposed mathematical model allows to get the optimum toe-in angles for the wheels of the vehicle. These toe-in angles allow to get the least rolling resistance with different angles of the steered wheels when the vehicle moves along the curvilinear trajectory. This in turn will improve the operating and environmental performance of the vehicle, in particular to reduce tire wear, which leads to an increase in tire resource and reduce fuel consumption, thereby reducing the emission of harmful materials into the environment.

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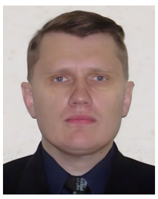
BIOGRAPHIES

Name: **Volodymyr**
Middle Name: **Gryhorovych**
Surname: **Verbitsky**
Birthday: 16.10.1953
Birthplace: Zaporizhzhia, Ukraine
Bachelor: Mathematical Analysis, Department of Mathematical Analysis, Faculty of Mechanics and Mathematics, Lomonosov Moscow State University, Moscow, Russia, 1977
Doctorate: Doctor of Physio-Mathematical Sciences Degree, S.P. Timoshenko Institute of Mechanics, Kiev, Ukraine, 1999
The Last Scientific Position: Prof., Department of Software for Automated Systems, Engineering Institute of Zaporizhzhia National University, Zaporizhzhia, Ukraine, Since 2016
Research Interests: Modeling of Transport Systems, Dynamics of Wheeled Vehicles, Application of the Theory Stability and Bifurcation to Problems of Dynamics Wheeled Vehicles
Scientific Publications: 150 Papers, 2 Books, 3 Patents, 100 Theses

Name: **Andrii**
Middle Name: **Vasyliovych**
Surname: **Shcherbyna**
Birthday: 30.11.1979
Birthplace: Zaporizhzhia, Ukraine
Bachelor: Automobiles, Faculty of Transport, Zaporizhzhia National Technical University, Zaporizhzhia, Ukraine, 2004
Doctorate: Automobiles and Tractors, National Transport University, Kiev, Ukraine, 2017
The Last Scientific Position: Assoc. Prof., Automobile Department, Zaporizhzhia Polytechnic National University, Zaporizhzhia, Ukraine, Since 2018
Research Interests: Research of Controllability and Stability of Vehicles, Solution of Scientific and Technical Problems Aimed at Improving Performance Characteristics of Vehicles
Scientific Publications: 80 Papers, 5 Patents, 30 Theses



Name: **Olexander**
Middle Name: **Mykolaiovych**
Surname: **Artyukh**
Birthday: 27.08.1974
Birthplace: Zaporizhzhia, Ukraine
Bachelor: Automobiles, Faculty of Transport, Zaporizhzhia National Technical University, Zaporizhzhia, Ukraine, 1998
Doctorate: Agricultural Engineering, National Center of Science Institute of Mechanization and Electrification of Agriculture, Kiev, Ukraine, 2002
The Last Scientific Position: Assoc. Prof., Automobile Department, Zaporizhzhia Polytechnic National University, Zaporizhzhia, Ukraine, Since 2018
Research Interests Study of Systems Active and Passive Safety of Vehicles, CAD Systems in Automotive Industry
Scientific Publications: 80 Papers, 3 Patents, 10 Theses

Name: **Olga**
Middle Name: **Vasylivna**
Surname: **Dudarenko**
Birthday: 31.12.1966
Birthplace: Zaporizhzhia, Ukraine
Bachelor: Automobiles, Faculty of Transport, Zaporizhzhia National Technical University, Zaporizhzhia, Ukraine, 1989
Doctorate: Mechanics of a Deformable Solid Body, Zaporizhia National Technical University, Zaporizhzhia, Ukraine, 2002
The Last Scientific Position: Assoc. Prof., Automobile Department, Zaporizhzhia Polytechnic National University, Zaporizhzhia, Ukraine, Since 2005
Research Interests: Research of Systems Active and Passive Safety of Vehicles, Theory of Technical Systems Vehicles
Scientific Publications: 70 Papers, 3 Patents, 20 Theses

Name: **Viktor**
Middle Name: **Volodymyrovych**
Surname: **Kuzmin**
Birthday: 12.06.1983
Birthplace: Zaporizhzhia, Ukraine
Bachelor: Automobiles, Faculty of Transport, Zaporizhzhia National Technical University, Zaporizhzhia, Ukraine, 2007
Doctorate: Sociological Science, Classic Private University, Zaporizhzhia, Ukraine, 2013
The Last Scientific Position: Assoc. Prof., Social Work and Psychology Department, Zaporizhzhia Polytechnic National University, Zaporizhzhia, Ukraine, Since 2015
Research Interests: Career Strategies in Modern Ukrainian Society; Traffic Safety Management, Safe Behavior of Road Users, Scientific Provision of Road Traffic Injury Prevention
Scientific Publications: 120 Papers, 2 Books, 30 Theses

Name: **Dmytro**
Middle Name: **Petrovych**
Surname: **Ruban**
Birthday: 20.10.1980
Birthplace: Cherkasy, Ukraine
Bachelor: Metal Cutting Machines, Faculty of Mechanical Engineering, Faculty, Cherkasy State Technological University, Cherkasy, Ukraine, 2004
Doctorate: Automobiles and Tractors, National Transport University, Kiev, Ukraine, 2013
The Last Scientific Position: Assoc. Prof., Motor Vehicle Transport and Social Security Department, Open International University of Human Development, Kiev, Ukraine, Since 2021
Research Interests: Study of Systems Active and Passive Safety of Bus, Technology of Bus Bodies, Production Process of Bus Bodies
Scientific Publications: 70 Papers, 25 Theses

Name: **Anton**
Middle Name: **Volodymyrovych**
Surname: **Fasoliak**
Birthday: 07.06.1992
Birthplace: Zaporizhzhia, Ukraine
Bachelor: Algebra and Number Theory, Faculty of Mathematics, Zaporizhzhia National University, Zaporizhzhia, Ukraine, 2014
Doctorate: Mechanics of a Deformable Solid Body, Zaporizhzhia National Univ., Zaporizhzhia, Ukraine, 2018
The Last Scientific Position: Assist. Prof., Higher Mathematics Department, Zaporizhzhia Polytechnic National University, Zaporizhzhia, Ukraine, Since 2018
Research Interests: Non-Stationary Dynamic Problems of Thin Cylindrical Shells in Three-Dimensional Elastic Media, Solution of Scientific and Technical Problems Aimed at Improving Performance Characteristics of Vehicles
Scientific Publications: 40 Papers, 15 Theses