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Simulation of the Effect of Turning Steering Wheel Intensity on the Vehicle Stability

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Abstract: ***In this article, a mathematical model has been developed to show the effect of the drivers' steering wheel turning intensity on the vehicle's stability. The developed mathematical model was compared with the results of experiment and its adequacy was evaluated. 3 conditional drivers turn the steering wheel of the vehicle at different speeds. When the conditional drivers were analyzed in the "J-turn" maneuver, it was determined that the indicators of 1,2,3 - conditional drivers are close to the standard. The conditional 2-driver recorded an indicator close to the standard. As for the "Single Lane Change" maneuver, the value of the smallest quadratic deviation from the trajectory of conditional 1-driver was recorded, the correlation index was equal to 0.102, respectively, 0.88***

Keywords: ***vehicle stability, "J-turn", "Single Lane Change" maneuver, trajectory, conditional driver, steering wheel turning intensity.***

I. INTRODUCTION

Traffic accidents are mainly caused by the lack of professional skills of vehicle drivers, improper maneuvers and neglecting of the speed mode. According to conducted research [1] more than 50% of the accidents occurred as a result of the vehicle's exit into another traffic lane of movement due to non-compliance of drivers with the motion speed mode and incorrect maneuver at the time of the turn. More than 45% cases of road traffic accidents with vehicle overtake occurred due to reasons such as incorrect turning maneuver or a sharp change in the corridor of motion. While driving and maneuvering, steering wheel is affected to a different level by the drivers. Especially, before making a maneuver drivers' steering wheel turning intensity plays an important role in maintaining the stability of the vehicle's motion. To model the effect of steering wheel turning intensity on the stability of the vehicle's motion, it is necessary to develop motion equations that sufficiently reveal the process. Many research studies are being conducted in these areas [2,3,4].

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Similar equations of motion have several degrees of freedom, and usually working with them creates different complexities. The reason is that as a result of experimental research it is necessary to determine the parameter values of many vehicle components. Therefore, this article, explores a compact, comprehensive and convenient mathematical model able to sufficiently reveal the modeling of the effect of steering wheel turning intensity on the vehicle's movement stability.

II. VEHICLE MODEL

The solution of the problem, which provides the desired characteristics of vehicle handling and stability, requires a fairly complete mathematical description, taking into account several degrees of vehicle freedom. An important condition for developing a mathematical description of a vehicle is to ensure that the simulation is adequate to the field experiment. To do this, the mathematical description of individual degrees of freedom must be sufficiently complete. The analysis of theoretical studies shows that to solve problems of vehicle handling and stability, various mathematical models are used in terms of completeness of description, starting with models that take into account two degrees of freedom of motion and including multi-mass models that take into account 4-14 degrees of freedom [2,3,4]. A very important and most difficult part in the study of the vehicle handling and stability is the "driver". To date, various authors have proposed mathematical descriptions of the driver as the control link of the "driver-vehicle" system, taking into account its complex actions to maintain a given direction of movement. These models differ in the complexity and completeness of accounting for various factors that characterize drive's behavior.

As a result, it is advisable to ensure first of all vehicle's stability and handling. That is, its ability to well counteract external disturbances, execute control signals with the necessary accuracy and speed when the driver is working, necessary to maintain a given direction of motion. These driving modes include the motion of the vehicle after a sudden turn of the steering wheel – "J-turn", "Single Lane Change", etc. For example, in "J-turn" maneuver, the driver's task is to quickly turn the steering wheel at a certain angle, hold it in the rotated position until the end of the transition process, and maintain the set speed. Therefore, this type of test is one of the main ones and is accepted in the standards [5,6,7]. In addition to this, the speed at which the steering wheel intense turn in the lane change maneuver is one of the important parameters while performing vehicle maneuver. Taking into account the above-mentioned points, the calculation scheme of the curve linear motion of the vehicle was developed. The scheme of calculation is presented in Figure 1.

According to the dynamic model, the vehicle is depicted in the form of a two-axle crew, the front wheel of which is located at the center of the front axle at the point O_1, and the rear wheel at the point O_2 in the center of the rear axle.

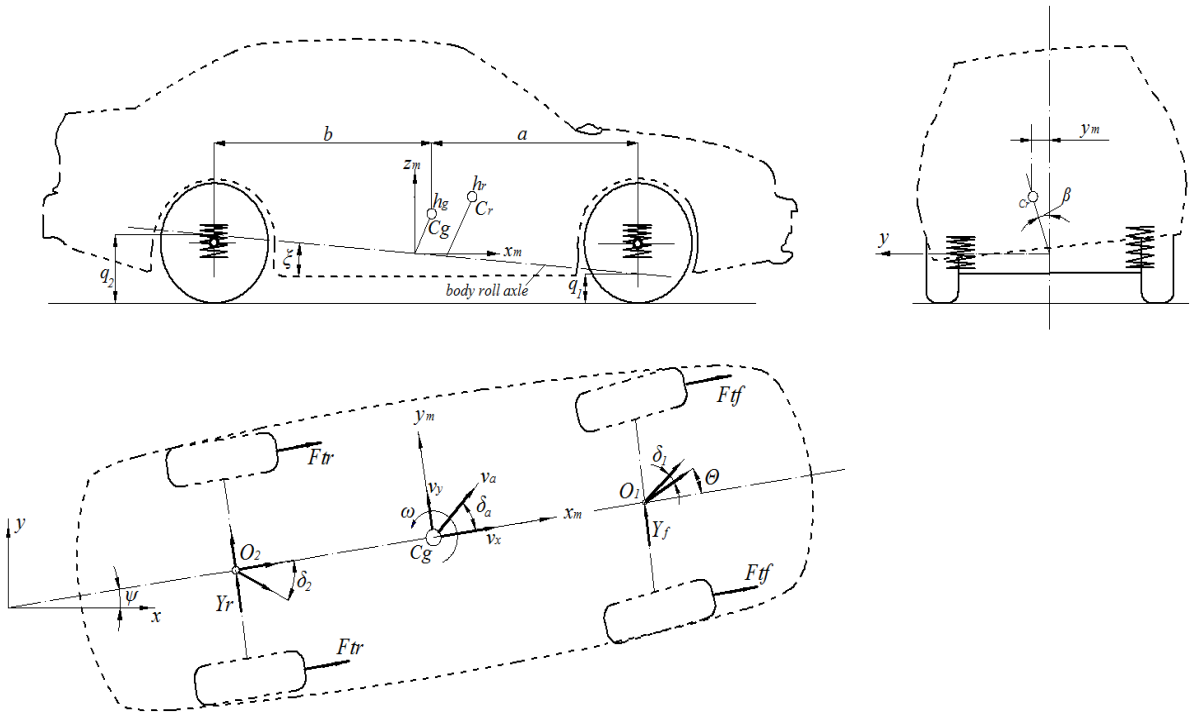


Fig. 1. Dynamical model of the vehicle (bicycle model).

The vehicle is placed in a system of coordinates with two points of inertia (x,y,z) and drive (x_m,y_m,z_m) to assess the linear motion. The longitudinal axis of the vehicle is located at an angle of ψ relative to the x axis. The angle of front axes wheels is - Θ.

The equation of motion of the vehicle in the direction of the Y axis can be written as follows:

$$M_a(\dot{\psi}+\dot{\delta}_a)v_a - m_n h_n \ddot{\beta} = \sum Y_1 + \sum Y_2 \qquad (1)$$

where M_a - is the total mass of the vehicle, ω - is the yaw rate of the vehicle, δ_a - the slip angle of the vehicle (the angle between the velocity vector of the center of gravity of the machine and its longitudinal axis), v_a – vehicle speed, m_n – sprung mass of the vehicle, h_r – distance from the center of gravity of the vehicle to the roll axis, β – angle of body roll, ΣY_f and ΣY_r - total lateral force on the front and rear axles.

The relationship between the slip angles and the lateral forces is set as follows:

$$Y_i = K_{yi}\delta_i \qquad (2)$$

where K_{yi} is the slip resistance coefficient i - axes of the vehicle; δ_i – is the slip angle of the axes.

The equivalent coefficient of resistance to tire deflection can be represented as [8]:

$$K_y = q_Z q_T K_{y0} \qquad (3)$$

where q_z, q_T - corrective factors taking into account the impact of the change in vertical load, tractive force and compressibility of soil on tire slip; K_{y0} – slip resistance coefficient takes on a linear plot of the dependence of slip angle from the lateral force.

The correction factor q_z is determined using the following formula:

$$q_Z = 2{,}4\frac{R_z}{G_0} - 1{,}8\left(\frac{R_z}{G_0}\right)^2 + 0{,}4\left(\frac{R_z}{G_0}\right)^3 \qquad (4)$$

where R_z is the vertical load on the wheel; G_0 is the vertical load on the wheels, corresponding to the maximum dependence on the slip resistance from the lateral force.

To account for the effect of traction force on the coefficient of slip resistance, the following formula is proposed [6]:

$$q_T = \sqrt{1-\left(\frac{P_{ti}}{R_Z\varphi}\right)^{1.5}}, \qquad (5)$$

where, P_{ti} is the traction force on the i wheel; φ is the coefficient of adhesion to the road.

The equation of motion of machines relative to the Z axis is equal to:

The equation of motion of machines relative to the Z axis is equal to:

$$J_z\dot{\omega} + (J_{xz} - J_{z\xi})\ddot{\beta} = \sum Y_1 a - \sum Y_2 b - \\ - \sum M_{cu} - \sum M_{cy} - \Delta M_F - M_g \qquad (6)$$

where, J_z is the moment of inertia wheeled vehicles along the z-axis, I_{xz} – centrifugal moment of inertia of the machine relative to the axes x and z, $I_{x\xi}$ is the moment of inertia of the sprung parts of machines relative to the x axis, ΣM_{cu} total elastic stabilizing moment of the tire; ΣM_{cy} - total stabilizing moment due to the longitudinal tilt of the king pin; ΔM_F - point of the difference between the force of resistance to rolling of the right and left wheels; M_g is the gyroscopic moment generated by changes of camber angle; a, b – distance to the center of gravity from center of the front and rear axle wheels. The equation of moments relative to the X axis for sprung masses has the form:

$$J_{xn}\ddot{\beta} + J_{xz}(\dot{\omega} - \ddot{\beta}\xi) = m_n h_r(\omega + \dot{\delta}_a)v_a - \sum M_{cn} - \\ - \sum M_{\lambda} \pm M_{mpn} \quad (7)$$

where $\sum M_{cn}, \sum M_{\lambda}, M_{mpn}$- represent respectively the moments of roll resistance of the sprung masses of elastic suspension elements, shock absorbers and dry friction in suspensions.

The equations of motion of the controlled wheels relative to the axis of rotation can be presented in the following form:

$$J_K\ddot{\Theta} = M_c + \mathrm{M}_{\lambda} + c_p(\frac{\alpha}{i} + \Theta) \quad (8)$$

where, J_z is the moment of inertia of the wheels vehicle, M_c – the moment caused by the split tire disposal and removal axis of the driven wheel, M_{λ} is the moment of viscous friction, is given to the axis of the driven wheel rack, c_p – rigidity steering, α - is the angle of the steering wheel, i – gear ratio steering, Θ is the rotation angle of the steered wheels.

Equations of motion of the vehicle taking into account (1), (6), (7) and (8) can be kept in mind:

$$\begin{cases} M_a(\dot{\psi} + \dot{\delta}_a)v_a - m_n h_r \ddot{\beta} = \sum Y_1 + \sum Y_2 \\ J_z\dot{\omega} + (J_{xz} - J_{z\xi})\ddot{\beta} = \sum Y_1 a - \sum Y_2 b - \\ - \sum M_{cu} - \sum M_{cy} - \Delta M_F - M_g \\ J_{xn}\ddot{\beta} + J_{xz}(\dot{\omega} - \ddot{\beta}\xi) = m_n h_r(\omega + \dot{\delta}_a)v_a - \\ - \sum M_{cn} - \sum M_{\lambda} \pm M_{mpn} \\ J_K\ddot{\Theta} = M_c + \mathrm{M}_{\lambda} + c_p(\frac{\alpha}{i} + \Theta) \end{cases} \quad (9)$$

In the modeling of the effect of the driver's steering wheel turning intensity on the movement stability of the vehicle, the control of the steering wheel and steering wheel play are not taken into account.

III. EXPERIMENTAL RESULTS

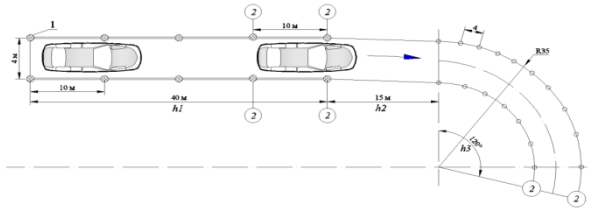
The purpose of experimental research was to determine the transient and steady-state characteristics of the machine for the control effect and obtain the necessary characteristics of the machine to check the adequacy of the mathematical model.

Table-1

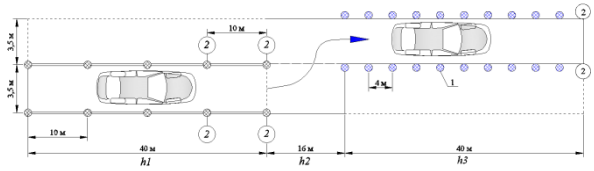
1	**Gyroscopic sensors**	
1.1	Dynamic range	±450 ^{0}/c
1.2	Non linearity in the full range	0.01 %
1.3	Mixing stability	±0035 ^{0}/c
2	**Accelerometers**	
2.1	Dynamic range	±5 g
2.2	Non-linearity in the full range of	0.03 %
3	Rated power consumption	1.3 watt
4	Temperature Range	$-20 \div 70^0 C$

The "J-turn" and "Single Lane Change" tests were selected to determine the transient responses of the vehicle to the driver's driving influence. This type of testing is included in the interstate standard [5,6,7], which regulates technical requirements for vehicles and test methods. As well as the characteristics of transient reactions of vehicles are normalized by setting the permissible range of transient and steady values for the angular speed of rotation, i.e., the sensitivity of the vehicle to the control effect from the position of the controllability of the vehicle is normalized.

A measuring system with special software was used for conducting experiments. Technical characteristics of the measuring system are given in table-1.



"J-turn" maneuver



"Single Lane Change" maneuver

Fig. 2. Geometrical dimensions of the experiment passageways: 1-vertical elements separating the passageway; 2-speed - measuring sensors; h1-acceleration section; h2-starting section of the passageway; h3-closing range of the passageway.

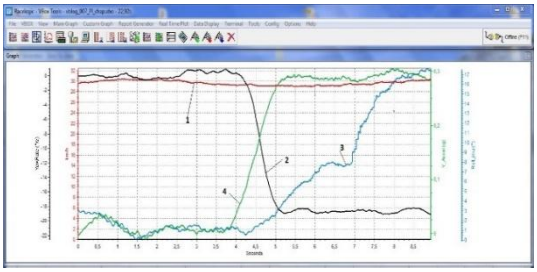


Fig. 3. "J-turn" maneuver (turn to the left) 1-the speed of the vehicle; 2-vehicle yaw rate; 3-the roll of the body; 4-lateral acceleration

Comparison of calculated and experimental data. Experiment was carried out with different modes of movement of the vehicle. The results of the experiment are presented in Fig. 3 and 4.

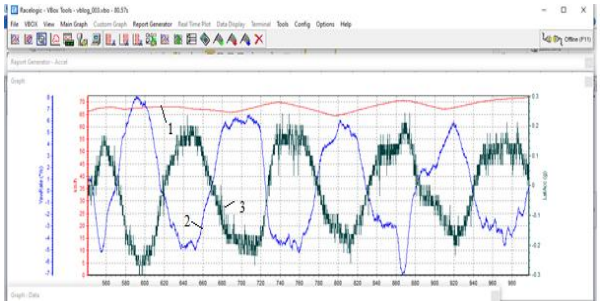


Fig. 4. "Single Lane Change" maneuver 1-speed of the vehicle; 2-vehicle yaw rate; 3-lateral acceleration of the vehicle

Determination of the degree of compliance of the vehicle's characteristics obtained by calculation with those obtained experimentally was carried out based on the transient characteristics of the angular speed of the vehicle when the steering wheel turns abruptly. Fig. 5 and 6 show the transient characteristics of the angular speed of turning of the vehicle, obtained by calculation and experimental methods. The graphs show the calculated curves obtained using a model that takes into account (ω, v_y, Θ) - (I) and takes into account ($\omega, v_y, \beta, \Theta$) - (II) in the mathematical model.

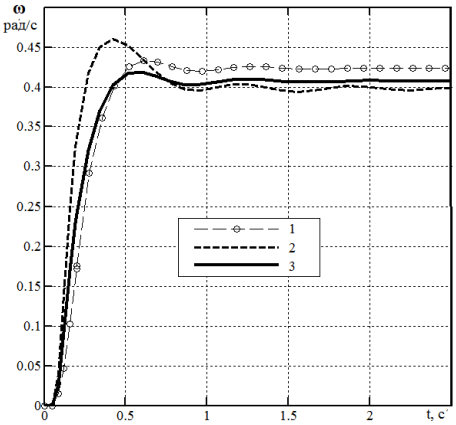


Fig.5. "J-turn" results, turn to the right. 1-experiment values; 2-model (I), 3-model (II), v =30 km/h.

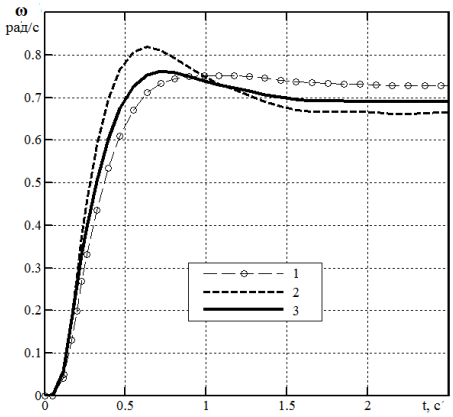


Fig.6. "J-turn" results, turn to the right. 1-experiment values; 2-model (I), 3-model (II), v=70 km/h.

The model that takes into account (ω, v_y, Θ) gives satisfactory results for small values of transverse accelerations that do not exceed 1,0–2,0 m/s^2. With the growth of transverse accelerations and the speed of movement, the nature of the transition characteristics differs both qualitatively and quantitatively from the experimental ones.

The model taking into account ($\omega, v_y, \beta, \Theta$) both qualitatively and quantitatively, gives a good match with the experimental characteristics at transverse accelerations up to 4,0 m/s^2. The difference between the calculated and experimental characteristics in the speed range of 36-72 km/h does not exceed 6-8%. Thus, further computational studies were carried out using a mathematical model with three degrees of freedom.

IV. SIMULATION RESULTS

Computational modeling was performed using the "Matlab-Simulink" program. In solving the mathematical model the following boundary values of the vehicle with the category *N1* were accepted, and the most basic parameters of them were as follows: v_a*=70 km/h;* M_a*=1050 kg;* m_n*=850 kg;* K_{y01}*=75000 N/rad;* K_{y02}*=1.15x*K_{y01} *(understeer); hr=0.41 m;* J_Z*=1820 kgm*^{2} *; a=1.23 m; b=1.32 m;* J_K*=8.8 kgm*^{2}*; 1,2,3 – the weight of the drivers is 75 kg;*

The results of the modeling were initially included in the **"J-turn" maneuver** for modeling the stability of the vehicle at the speed of rotation of the steering wheel, respectively α_1=300, α_2=250, α_3=200 rad/h to 1,2,3 – conditional drivers. After a sharp turn on the steering wheel, 2-3 c is held until the current angular velocity ω_{cur} of the vehicle at the turn is stabilized, that is ω_{stab} to the value and is compared with the standard.

Table-2 shows the transient characteristics of the vehicle obtained for a speed of 70 km / h. The lateral acceleration of the N1 j_y=4 м/с^2 Category vehicle 90% should not exceed reaction time 0.3s [5,6,7]. And relative angular velocity drop should not exceed 30%. When conditional drivers are analyzed (relative angular velocity drop), the indicators of 1,2,3 conditional drivers are close to the standard, that is, they are close to the values indicated by the driver with a mal. Only the conditional driver -2 recorded an indicator close to the standard when the indicators on the 90% time reaction were analyzed.

Table-2

Conditional drivers	Relative casting angular velocity,% (j_y=4 m/s^2)	90% reaction time, with (j_y=4 m/s^2)
1	26.2	0.45
2	28.1	0.38
3	13.3	0.59

The normalized characteristics of the vehicle are shown in Fig.7. The dotted line (R) in the same figure shows the range of acceptable values of the normalized transition characteristic according to the standard. 1,2 and 3 Conditional drivers respectively.

As can be seen in the Fig. 7, the 3-Conditional driver is leaving the area of the normalized transition process. According to the indicators of the 1conditional driver, the vehicle is initially stable when entering the circle, but is approaching the edge border of the sphere, which is limited by the increase in the current value of the vehicle's angular speed. And the indicator of the 2conditional driver remains within the normalized sphere and indicates the stability of the movement of the vehicle in the position after the end of the maneuver.

For modeling the **"Single Lane Change"** maneuver the middle line of the road corridor R is set as:

$$y_0 = y_0^1 + y_0^2 \quad (10)$$

$$y_0^1 = \begin{cases} \alpha v_a t, & \text{at } t \succ 0 \\ 0, & \text{at } t \prec 0 \end{cases} \quad (11)$$

$$y_0^2 = \begin{cases} -\alpha v_a (t - \frac{l}{v_a}), & \text{at } t \succ \frac{l}{v_a} \\ 0, & \text{at } t \prec \frac{l}{v_a} \end{cases} \quad (12)$$

For modeling of the maneuver, 1,2,3-conditional drivers are provided with the condition that the steering wheel rotates at a speed of α_1=120, α_2=140, α_3=200 rad/s, respectively. Movement speed 70 km/h. The results of the modeling of the "Single Lane Change" maneuver are presented in Fig. 8.

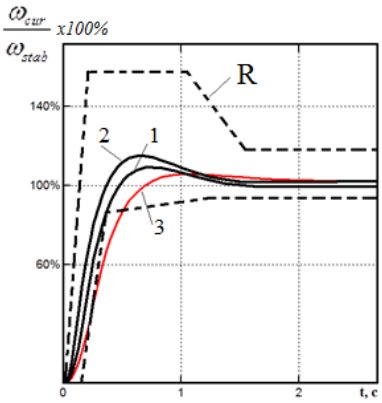


Fig. 7. "J-turn" results:

R-range of acceptable values of the normalized transition characteristic according to the standard, 1,2 and 3 conditional drivers respectively.

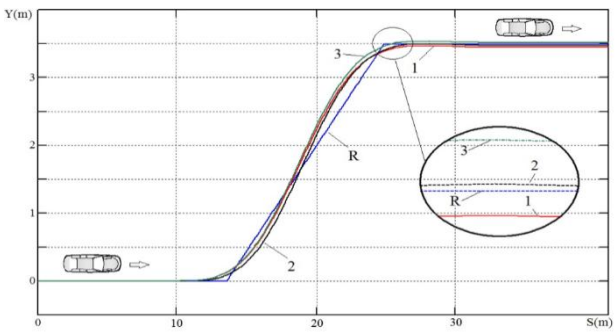


Fig. 8. "Single Lane Change" maneuver. Desired (R) and realized vehicle trajectory (1,2,3) maneuver

As can be seen in the Fig. 8 above, conditional drivers are performing different types of maneuvers, depending on the turning intensity of the different wheel.

The following formula was used to determine the square deviation $E^2_{r(y)}$ of the conditional drivers from the given (R):

$$E^2_{r(y)} = \sum (d_i - R)^2 \quad (13)$$

where $d_i - i$ the value of the deviations from the drivers' coordinate (i=1,2,3), R- the coordinate of the given road

The smaller the value $E^2_{r(y)}$ in the equation (13), the lesser the errors.

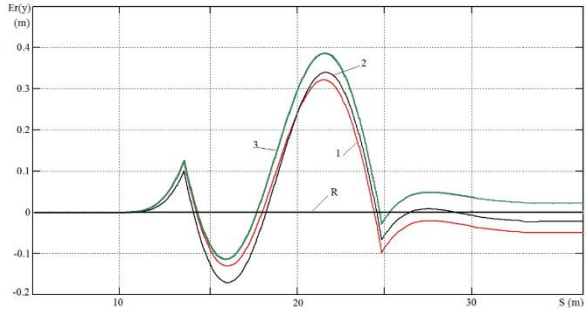


Fig. 9. "Single Lane Change" maneuver. Deviation of the trajectory of the vehicle from the driver *(1,2,3)* from the set one (R)

There is an indicator that characterizes the closeness of the connection d_i and R, which is called the correlation coefficient k and is calculated by the formula:

$$k = \sqrt{1 - \frac{\sum_{i=1}^{n}(y_i - R_i)^2}{\sum_{i=1}^{n}(y_i - M_y)^2}} \quad (14)$$

where, y_i values of the realized trajectory of the 1 driver, values of the desired trajectory (R) and M_y – the average of y values.

The correlation index in its absolute value ranges from 0 to 1. For smaller deviations, the correlation index is close to k 1.

Table-3

Conditional drivers	$E^2_{r(y)}$	k
1	0.102	0.88
2	0.115	0.76
3	0.144	0.69

As can be seen from the 3-table above, the conditional 1drive's smallest square deviation value is equal to 0.102, respectively, the correlation index is 0.88.

V. CONCLUSION

According to the theoretical and experimental results considered above, the intensity of turning the vehicle's steering wheel while performing 2 different maneuvers has different effects on its stationary motion. On the analysis of conditional drivers in the "J-turn" maneuver (Relative angular velocity drop) the indicators of 1,2,3 conditional drivers are close to the standard. Only the conditional 2 driver recorded an indicator close to the standard when the indicators on the 90% reaction time were analyzed. On the "Single Lane Change" maneuver, the conditional 1 driver's smallest square deviation value is equal to 0.102, respectively, the correlation index is equal to 0.88.

The method of modeling, considered above, allows us to predict the stability of movement in the "vehicle-driver" system in advance. With the help of this method it is also possible to improve the constructive features of vehicles by analyzing them.

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