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ALGORITHM OF AUTONOMOUS VEHICLE STEERING SYSTEM CONTROL LAW ESTIMATION WHILE THE DESIRED TRAJECTORY DRIVING

Sergey Sergeevich Shadrin and Andrey Mikhailovich Ivanov
Moscow Automobile and Road Construction State Technical University (MADI), Leningradskiy Pr. 64, Moscow, Russia
E-Mail: shadrin@madi.ru

ABSTRACT

The article discusses an estimation algorithm of control actions on steering system in order to provide vehicle driving along desired trajectory with accounting of non-steady (transient) driving modes. Minivan tests on MADI proving ground are described, the developed theory is verified.

Keywords: car, vehicle steering, autonomous vehicle, vehicle dynamics, vehicle motion simulation, automatic steering control.

INTRODUCTION

During driving of an autonomous vehicle [1] along a desired trajectory one of the most important tasks is to determine the exact control actions, including those on steering system [2]. In order to investigate into peculiar features of vehicle control the experiment was performed: GAZ-322132 minivan (VIN X9632213280611889) carried out maneuver “Lane change S = 20” (State standard GOST 31507-2012 [3]) at various speeds. Thus, the vehicle drove along the same approximate trajectory, but with different control actions on steering wheel. Figure-1 illustrates steering wheel angle as a function of traveled distance with various velocities.

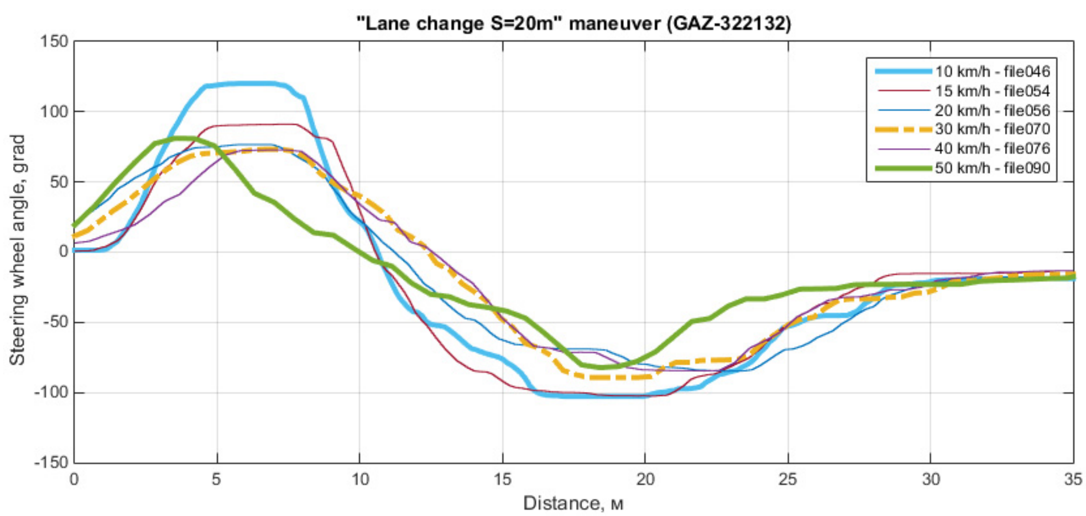

Figure-1. Vehicle driving along fixed trajectory.

Therefore, it can be stated, that with increase of vehicle velocity the driving along the same fixed trajectory is accompanied with decrease of steering wheel angle amplitude and phase advance shift.

It was required to develop estimation algorithm of steerable wheels turn angles as a function of time with given fixed trajectory, which has been recorded during road tests conduction.

The following measuring equipment was used:

a) Car scales “Intercomp Racing SW500 E-Z Weigh Cabled Scale Systems”;

b) Tri-Axial Navigational Sensor (Kistler) of accelerations and angular velocities, mounted approximately in the vehicle center of gravity;

c) 100Hz CDS GPS-Glonass (Kistler) data recorder;

d) Wire Potentiometer, D8.3A1.0125.A223.000 (Kistler), rigidly fixed on front suspension lateral axle beam and measuring the variation dynamics of distance to flexible coupling of transverse arm with inclined wheel hub steering lever of front right wheel for subsequent calculation of steerable wheels positions and steering wheel angle upon manipulations;

e) Hand-Lever Force Sensor (Kistler): force sensor, was used as digital signal of road tests starts;

f) CSM AD-Scan MiniModul (Kistler) analog-to-digital converter;

g) power distribution unit with independent battery.

CALCULATION OF STEERABLE WHEELS TURN ANGLE

Average wheel turn angle as a function of kinematic parameters of vehicle motion can be calculated, for instance, as follows [4]:

$$\delta_W = \frac{l}{R} + m_{CoG} \cdot \frac{V_{CoG}^2}{R} \cdot \left(\frac{l_R}{l \cdot K_{yF}} - \frac{l_F}{l \cdot K_{yR}} \right) \tag{1}$$

where

δ_W = wheel turn angle;

l = wheelbase;

R = turning radius (respectively, R^{-1} is the trajectory curvature);

V_{CoG} = velocity of vehicle center of gravity;

l_F = distance from vehicle center of gravity to front axle;

l_R = distance from vehicle center of gravity to rear axle;

K_{yF} = front wheels tire side slip constant;

K_{yR} = rear wheels tire side slip constant.

In Bosch handbook [5], while describing principles of operation of dynamic stability system, the equation is used which can be applied for calculation of wheel turn angle:

$$\dot{\psi} = \frac{V_{CoGX}}{l} \cdot \delta_W \cdot \frac{1}{1 + \left(\frac{V_{CoGX}}{V_{ch}} \right)^2}$$

$$\delta_W = \frac{\dot{\psi} \cdot l \cdot \left(1 + \left(\frac{V_{CoGX}}{V_{ch}} \right)^2 \right)}{V_{CoGX}} \tag{2}$$

where

$\dot{\psi}$ = yaw rate;

V_{CoGX} = longitudinal velocity of vehicle center of gravity;

V_{ch} = characteristic vehicle velocity, that is, the parameter which generalizes geometrical and physical properties of vehicle.

The performed tests demonstrated that Equation (1) provides acceptable results if, using the procedure described in article [6], the coefficients K_{yF} and K_{yR} are substituted with the functions of velocities (see Figure-2), and moreover, such calculations, as Eq. (2), can describe only steady (quasi-stationary) driving modes. Comparisons of wheel turn angles according to the developed procedure with those according to Eq. (2) will be presented below, Equation (2) will be mentioned in the plots as the Bosch equation.

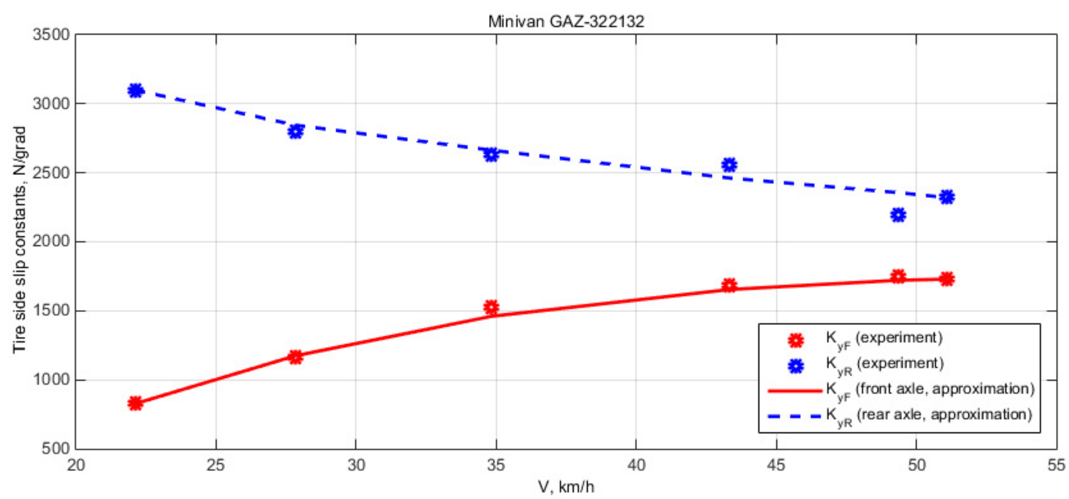

Figure-2. Tires side slip constants as a functions of vehicle velocity.

It should be mentioned, that in order to reproduce the control law of steering wheel it is not sufficient to predefine the trajectory as a discrete set of Cartesian coordinates of vehicle center of gravity *(x, y)*, it is also required to define the travelling velocity, for instance, as a parameter of time *t* in each point of the trajectory *(t, x, y)*, however, these initial parameters are also insufficient for calculations, since the vehicle center of gravity in general case can travel along one and the same trajectory, driving with various body sideslip angles β. Thus, let us consider the discrete data set *(t, x, y, β)* as initial data for calculations, which is equal to the fact that in the tests the travelling trajectory of two different vehicle body points were recorded.

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The main difficulty in simulation of curvilinear vehicle driving is the calculation of lateral forces, acting in tires contact areas, since tire due to its elastic properties reacts to disturbances with some delay.

Basic calculated equation of lateral force F_S on the i^{th} wheel is presented by empirical equation from [7] and [8]:

$$F_{Si}(\alpha, F_Z, k) = \left(1 - \frac{F_{Zi}}{k_{y1}}\right) \cdot F_{Zi} \cdot \arctan\left(k_{y2} \cdot \alpha(t)\right) \quad (3)$$

where:

k_{y1}, k_{y2} = dynamic tire side slip constants;

α = tire side slip angle;

F_Z = vertical wheel load.

Analysis of Equation (3) revealed that it describes well the decrease of amplitudes of lateral force under dynamically varying external impacts (vertical force and side slip angle), but it does not describe in terms of physics the delay in increment of lateral force stipulated by tire elasticity. Thus, modifying Equation (3) with accounting for empirical dependence from [9] and [10], we obtain the final equation for calculation of lateral force:

$$\frac{\partial F_{Si}}{\partial t} = \frac{V_W}{L_{y0}} \cdot \left(\left(1 - \frac{F_{Zi}}{k_{y1}}\right) \cdot F_{Zi} \cdot \arctan\left(k_{y2} \cdot \alpha(t)\right) - F_{Si} \right) \quad (4)$$

where

V_W = wheel velocity;

L_{y0} = relaxation length of pneumatic tire.

Further on, the calculations were based on the vehicle “bicycle” model, the tire side slip angles and dynamic weight distribution over axles were determined by equations from [7]. The calculation procedure of steerable wheel turn angle while driving along desired trajectory is illustrated in Figure-3.

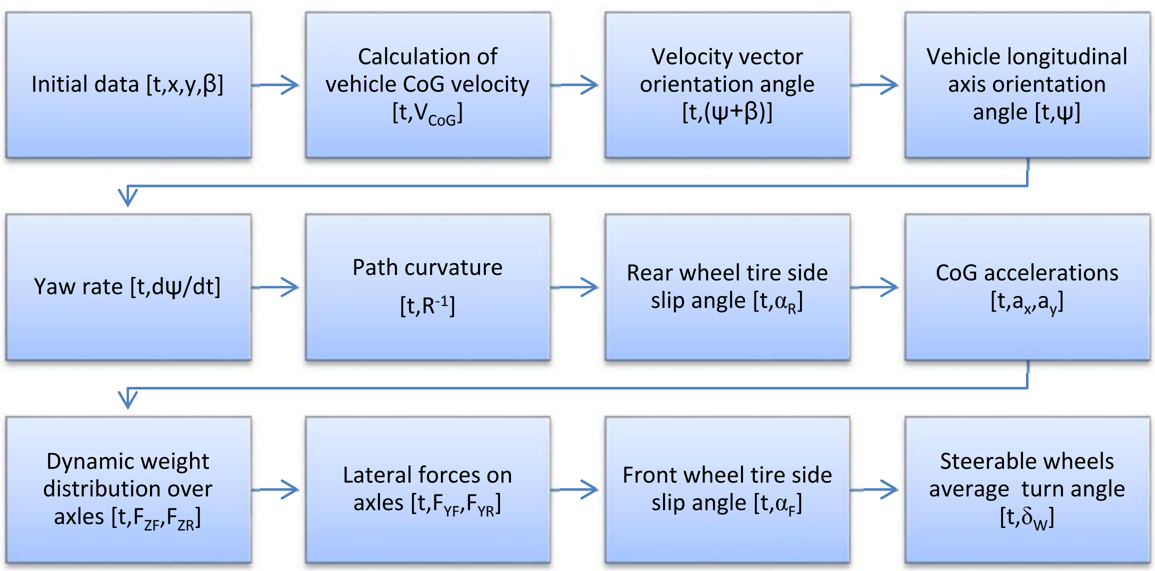

Figure-3. Calculation procedure of steerable wheels average turn angle upon driving along desired trajectory.

Figure-4 and Figure-5 illustrate comparisons between acquired upon road tests steerable wheels average turn angles with calculations according to the developed procedure (Figure-3) and by Equation (2).

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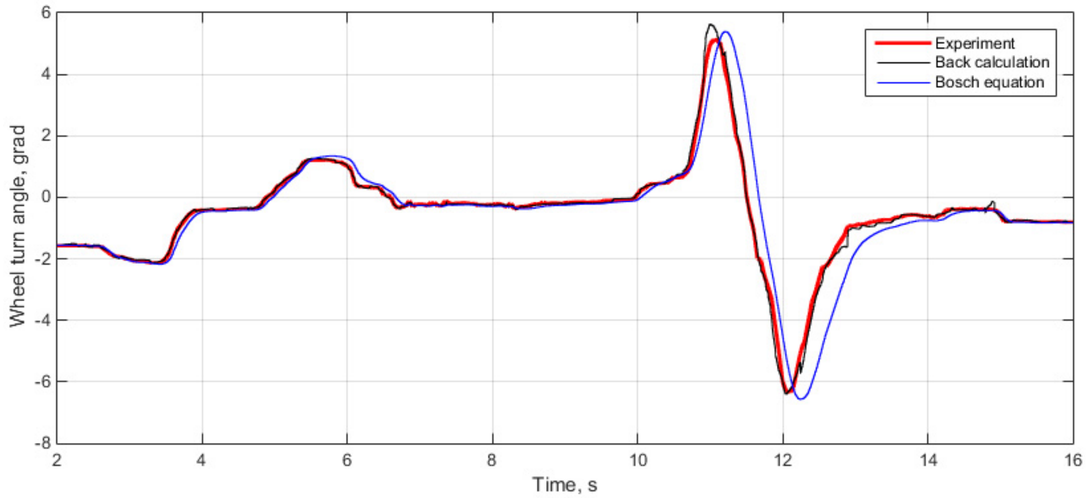

Figure-4. GAZ-322132 minivan, “Lane change S = 20”, velocity: 48 km/h.

It can be seen, that the calculated wheel turn angle according to the proposed procedure is in good agreement with experimental data, coinciding both in phase and in amplitude of actions.

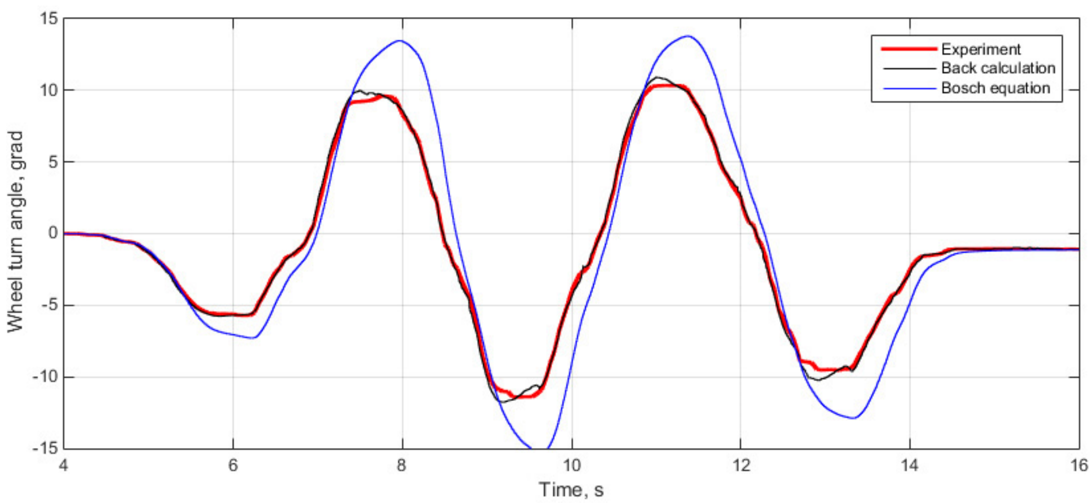

Figure-5. GAZ-322132 minivan, Slalom, 18 m; velocity: 37 km/h.

The road test in Figure-5 was performed with lateral accelerations of about 5.5 m/s^2. Calculated and experimental lateral forces acting on vehicle are compared in Figure-6.

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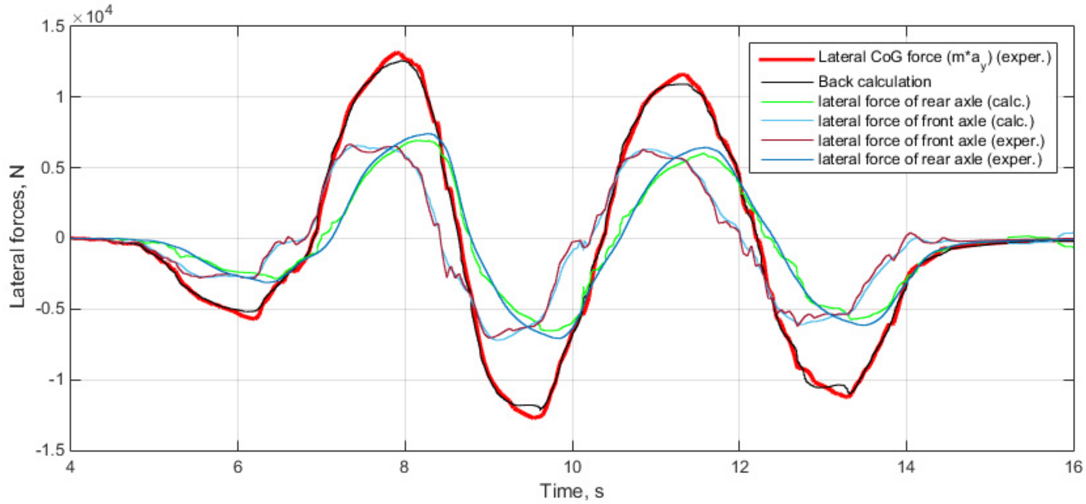

Figure-6. GAZ-322132 minivan, Slalom, 18 m; velocity: 37 km/h.

As expected, Eq. (2), valid only for steady driving modes, should not be used in the tasks of vehicle steering control automation. Therefore, the proposed procedure can be applied in prediction steering control system of driving of autonomous wheeled vehicles.

RESULTS AND CONCLUSIONS

a) The estimation algorithm of control actions on steering system has been developed, which provides driving of a vehicle of 3--5 SAE automation levels along desired trajectory.

b) An estimated equation is proposed for determination of lateral force in the pneumatic tire contact area, which describes non-steady (transient) driving modes.

c) Road tests with a minivan have been performed in the MADI proving ground; the validity of the developed procedure has been verified.

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