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Research paper

Method of Autonomous Vehicle Control Using Simplified Reference Models and Regulators

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Abstract. This study contains a conceptual solution of the control method for an autonomous vehicle based on a simplified reference model and regulators. The whole issue is considered on the example of the execution phase of the lane change manoeuvre, in which the well-known 4WS (*Four-Wheel-Steering*) bicycle model and Kalman regulators based on the LQR (*Linear-Quadratic Regulator*) technique were used. The overall developed control mechatronic system was subjected to simulation studies in Matlab&Simulink. The simulation results showed correct functioning of the entire mechatronic control system and allowed us to determine further research directions. The developed control method can also find application in military vehicles.
Keywords: mechanical engineering, autonomous vehicle, 4WS vehicle, lane change process, bicycle model

1. INTRODUCTION

The 4WS system dates back to the early 20th century in its first developments. Among the vehicles produced in which it was used at that time, the military Nash Quad can be singled out. The truck was developed and built by Thomas B. Jeffrey in 1913. The Nash Quad was recognised as one of the most successful vehicles used during the First World War. The steering mechanism used in it caused the rear wheels to turn in the opposite direction to that of the front wheels, which provided a significant reduction in turning radius at low speed, thus increasing manoeuvrability [6]. The 4WS vehicles often feature in military applications, as it can be seen in the military designs used over the years up to the present day. Examples of vehicles are shown in the following figure (Fig. 1).

Fig. 1. The 4WS military vehicles [21, 22 and 23]

Nowadays, the 4WS system is mostly used in luxury cars. The idea behind 4WS vehicle control is to turn the rear wheels opposite to the front wheels at low speeds (increasing manoeuvrability, for example: during vehicle parking) and to turn the rear wheels in line with the front wheels at high speeds (increasing stability, for example: when driving on the highway). The speed at which the vehicle switches from manoeuvrability to stabilisation mode depends on the application adopted by the manufacturer.

The use of 4WS is therefore an element of driver assistance that translates into improved active safety. The authors decided to consider the issue of automatic control in a 4WS vehicle using a selected manoeuvre as an example.

The control problems of an autonomous vehicle are related to automatic navigation [1] and automatic obstacle avoidance [3]. The proposed control method for an autonomous vehicle is illustrated by the mechatronic control system diagram (Fig. 2).

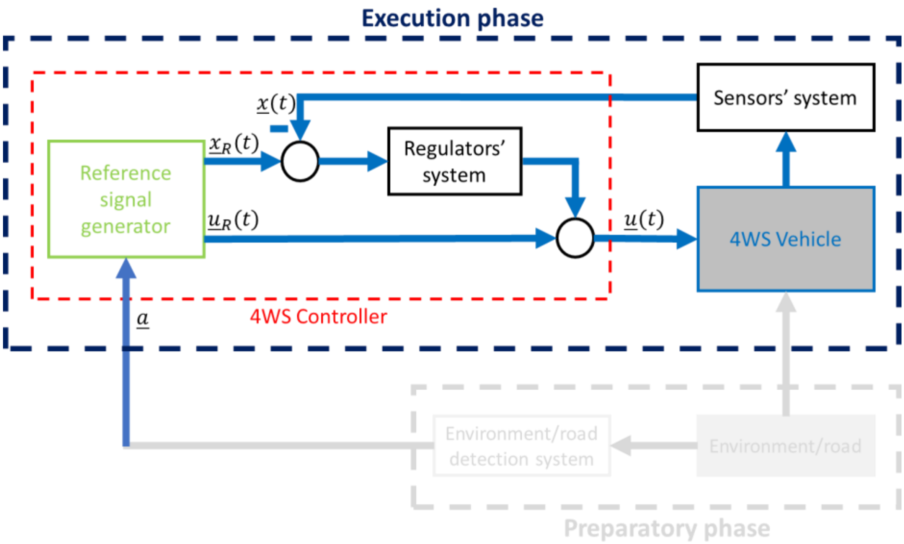

Fig. 2. Method of controlling an autonomous vehicle [4]

The mechatronic control system includes a digital controller with a reference signal generator and regulators' system, as well as detection systems (e.g. radar and lidar) and measurement systems (e.g. gyroscopic motion sensors) mounted on the vehicle [12, 15, and 16]. The generator generates, on the basis of a simplified model (reference model) [2], the applied control theory and data acquired from ambient recognition detection systems $\underline{a}$, the control signals $\underline{u}_R(t)$ appropriate for the traffic situation that has occurred and the corresponding model output signals $\underline{x}_R(t)$ describing the traffic trajectory [7 and 18]. The reference control signals are then corrected by the system of regulators resulting in the vehicle control signals $\underline{u}(t)$. The regulators use the deviation signals between the generated reference response signals and the signals measured by the motion sensors in the actual vehicle $\underline{x}(t)$ [16]. With the help of the regulators, the vehicle control signals can be adjusted so that, the movement trajectory of the actual vehicle is close to the reference trajectory suitable for the traffic situation [19].

The application of this mechatronic concept is presented using the example of a lane change manoeuvre [4-6] (Fig. 3).

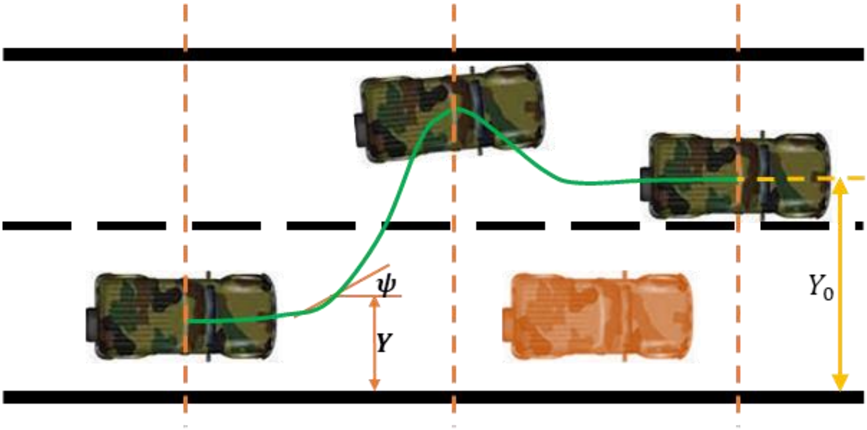

Fig. 3. Lane change process

There are two phases in the control process: the preparatory phase and the execution phase. In the preparatory phase, the traffic situation is identified, including determination of the required parameters of the lane change reference trajectory. In this phase, the steering angle („bang-bang") and the transverse and angular displacement of the vehicle are determined based on control theory and a simplified reference model [9 and 10]. The reference model used is based on of the so-called „bicycle model" [8, 13, and 14] known in car motion theory, expressed in transmittance form to facilitate its reduction [2-4]. In the execution phase, the actual control process takes place with the participation of regulators correcting the „bang-bang" course. This phase presents two main processes: the transposition process, which is the point at where the vehicle moves into the adjacent lane, and the stabilisation process, in which the vehicle stabilises its path after the manoeuvre. Examples of simulation calculations, illustrating the proper functioning of the system, are presented. In these calculations, a virtual, significantly more accurate vehicle model (taking into account, among other things, the parameters affecting the transitions and the steering system [20]) than the one used in the reference model is used as the control object. The method used will concern the execution phase of the lane change manoeuvre, allowing the mechatronic control concept to be represented as in Fig. 4.

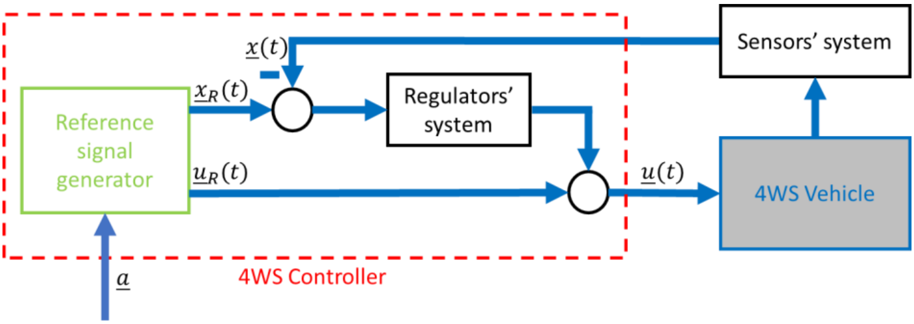

Fig. 4. Mechatronic control concept analysed

The figure above symbolically indicates the data supplied to the reference signal generator from the detection system (preparatory phase). On the other hand, it is not taken up here how these data are supplied. In this mechatronic system, the controller generates an electronic control voltage signal to the vehicle. The steering actuator converts the voltage signal into a mechanical signal, allowing the vehicle to perform the intended manoeuvre.

2. MODEL STRUCTURE

In the proposed mechatronic control method, a controller model was developed based on a „bicycle model" (Fig. 5) – Eqs. (1-5). It is described as a flat model with three degrees of freedom, requiring only seven parameters for its operation (speed, mass, front and rear axle distances from the centre of mass, and front and rear tyre cornering stiffness).

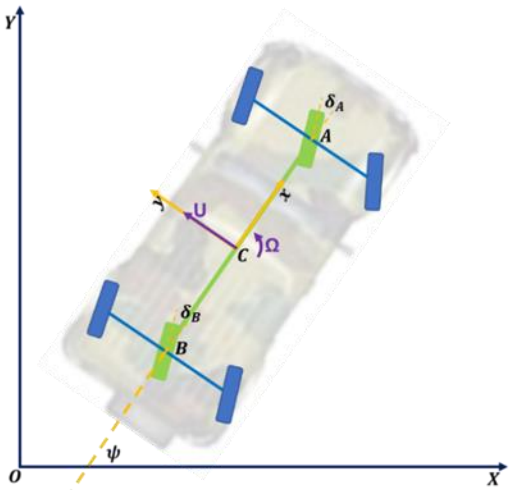

Fig. 5. Bicycle model 4WS

The longitudinal speed of the vehicle is assumed to be a constant parameter, allowing the mathematical form of the model to be written as follows:

$$m\dot{U}(t)+\frac{K_A+K_B}{V}U(t)+\frac{mV^2+K_AL_A-K_BL_B}{V}\Omega(t)=K_A\delta_A(t)+K_B\delta_B(t) \tag{1}$$

$$J\dot{\Omega}(t)+\frac{K_AL_A{}^2+K_BL_B{}^2}{V}\Omega(t)+\frac{K_AL_A-K_BL_B}{V}U(t)=K_AL_A\delta_A(t)-K_BL_B\delta_B(t) \tag{2}$$

where:

- V – the longitudinal vehicle speed in the local coordinate system (constant),
- U – the lateral vehicle speed in the local coordinate system,
- Ω – the angular vehicle speed in the local coordinate system,
- m – the vehicle mass,
- J – the mass moment of inertia (J_{ZZ} to p. C),
- L_A and L_B – the distances AC and BC between the centre of mass to the front / rear axis,
- K_A and K_B – the front and rear tyre cornering stiffness (yaw coefficients to p. A and B).

The above equations are presented for the local (x, y) coordinate system. For further work on this model, these equations need to be transformed to a global coordinate system. This allows them to be written as follows:

$$X(t)=\int_0^t\dot{X}(\tau)d\tau=\int_0^t\big(V\cos(\psi(\tau))-U(\tau)\sin(\psi(\tau))\big)d\tau \tag{3}$$

$$Y(t)=\int_0^t\dot{Y}(\tau)d\tau=\int_0^t(V\sin(\psi(\tau))+U(\tau)\cos(\psi(\tau)))d\tau \tag{4}$$

$$\psi(t)=\int_0^t\Omega(\tau)d\tau \tag{5}$$

For the moment, the model consists of linear equations in the local system and non-linear equations in the global system. As mentioned earlier, the control method discussed was developed using the example of a lane change manoeuvre. This means in practice that the vehicle deflection angles during this manoeuvre are small. This allows for a partial simplification of the non-linear equations, i.e., their linearisation. The form of the equations after the linearisation process is as follows:

$$X(t)=\int_0^t\dot{X}(\tau)d\tau=Vt \tag{6}$$

$$Y(t)=\int_0^t\dot{Y}(\tau)d\tau=\int_0^t\big(V\psi(\tau)+U(\tau)\big)d\tau \tag{7}$$

The linearisation of the above equations allows the equations of the 4WS „bicycle model” to be represented in operator form. After the Laplace transform, the model will have the following form:

$$\left(ms + \frac{K_A + K_B}{V}\right)U(s) + \frac{mV^2 + K_A L_A - K_B L_B}{V}\Omega(s) = K_A\delta_A(s) + K_B\delta_B(s) \tag{8}$$

$$\left(Js + \frac{K_A {L_A}^2 + K_B {L_B}^2}{V}\right)\Omega(s) + \frac{K_A L_B - K_B L_B}{V}U(s) = K_A L_A \delta_A(s) - K_B L_B \delta_B(s) \tag{9}$$

$$Y(s) = \frac{V\psi(s) + U(s)}{s} \tag{10}$$

$$\psi(s) = \frac{\Omega(s)}{s} \tag{11}$$

The above interpretation of the model equations allows it to be expressed in transmittance form (Fig. 6).

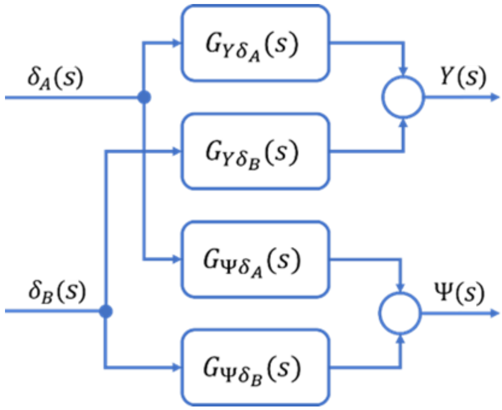

Fig. 6. Model in transfer function version [3]

$$G_{Y\delta_A}(s) = \frac{VK_0\left(T_{Y\delta_A}^2 s^2 + 2\xi_{Y\delta_A}T_{Y\delta_A}s + 1\right)}{s^2(T_0^2 s^2 + 2\xi_0 T_0 s + 1)} \tag{12}$$

$$G_{Y\delta_B}(s) = \frac{VK_0(T_{Y\delta_B}^2 s^2 + 2\xi_{Y\delta_B}T_{Y\delta_B}s - 1)}{s^2(T_0^2 s^2 + 2\xi_0 T_0 s + 1)} \tag{13}$$

$$G_{\psi\delta_A}(s) = \frac{K_0(T_{\psi\delta_A}s + 1)}{s(T_0^2 s^2 + 2\xi_0 T_0 s + 1)} \tag{14}$$

$$G_{\psi\delta_B}(s) = \frac{-K_0\left(T_{\psi\delta_B}s + 1\right)}{s(T_0^2 s^2 + 2\xi_0 T_0 s + 1)} \tag{15}$$

where:

$$K_0 = \frac{K_A K_B (L_A + L_B) V}{K_A K_B (L_A + L_B)^2 - mV^2 (K_A L_A - K_B L_B)} \tag{16}$$

$$T_0 = V \sqrt{\frac{mJ}{K_A K_B (L_A + L_B)^2 - mV^2 (K_A L_A - K_B L_B)}} \tag{17}$$

$$\xi_0 = \frac{m(K_A {L_A}^2 + K_B {L_B}^2) + J(K_A + K_B)}{2\sqrt{mJ(K_A K_B (L_A + L_B)^2 - mV^2 (K_A L_A - K_B L_B))}} \tag{18}$$

$$T_{\psi \delta_A} = \frac{m L_A V}{K_B (L_A + L_B)} \tag{19}$$

$$T_{\psi \delta_B} = \frac{m L_B V}{K_A (L_A + L_B)} \tag{20}$$

$$T_{Y \delta_A} = \sqrt{\frac{J}{K_B (L_A + L_B)}} \tag{21}$$

$$T_{Y \delta_B} = \sqrt{\frac{J}{K_A (L_A + L_B)}} \tag{22}$$

$$\xi_{Y \delta_A} = \frac{L_B}{2V} \sqrt{\frac{K_B (L_A + L_B)}{J}} \tag{23}$$

$$\xi_{Y \delta_B} = \frac{L_A}{2V} \sqrt{\frac{K_A (L_A + L_B)}{J}} \tag{24}$$

The above shows a very strong correlation between transmittance parameters with the mechanical parameters. However, the above interpretation is still incomplete, as the transmittances derived for the front wheels must be interdependent with the transmittances for the rear wheels. At this stage, it is necessary to introduce a transmission characteristic that will help achieve such a dependence (Fig. 7).

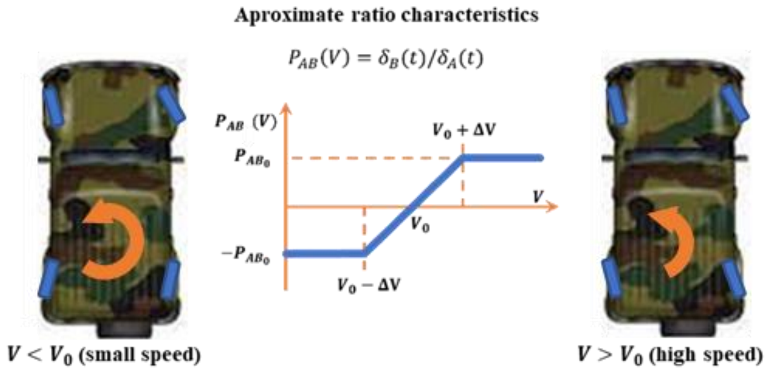

Fig. 7. Overall idea of steering front and rear wheels in 4WS vehicles

With the introduction of the above rear wheel ratio characteristics, the scheme of the transmittance form of the model (Fig. 8) and its equations will be slightly modified.

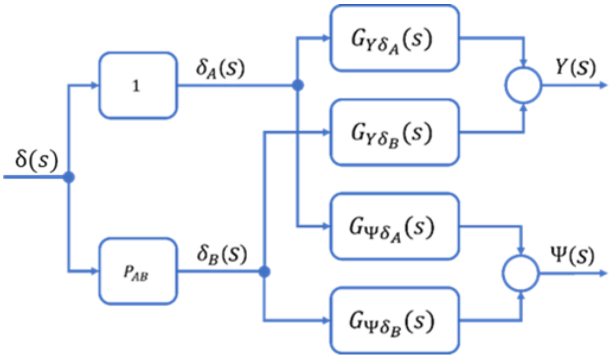

Fig. 8. Model including rear wheel ratio [3]

$$Y(s) = G_{Y\delta}(s)\delta(s) \tag{25}$$

where
$$G_{Y\delta}(s) = G_{Y\delta_A}(s) + P_{AB}(V)G_{Y\delta_B}(s) \tag{26}$$

$$\psi(s) = G_{\psi\delta}(s)\delta(s) \tag{27}$$

where
$$G_{\psi\delta}(s) = G_{\psi\delta_A}(s) + P_{AB}(V)G_{\psi\delta_B}(s) \tag{28}$$

from
$$G_{Y\delta}(s) = \frac{(1-P_{AB}(V))VK_0(T_Y^2 s^2+2\xi_{Y\delta}T_{Y\delta}s+1)}{s^2(T_0^2 s^2+2\xi_0 T_0 s+1)} \tag{29}$$

$$G_{\psi\delta}(s) = \frac{(1-P_{AB}(V))K_0(T_{\psi\delta}s+1)}{s(T_0^2 s^2+2\xi_0 T_0 s+1)} \tag{30}$$

where:

$$T_{Y\delta}=\sqrt{\frac{J\left(\frac{1}{K_B}+\frac{P_{AB}(V)}{K_A}\right)}{(L_A+L_B)\big(1-P_{AB}(V)\big)}} \tag{31}$$

$$\xi_{Y\delta}=\frac{L_B+P_{AB}(V)L_A}{2V\big(1-P_{AB}(V)\big)\sqrt{\frac{J\left(\frac{1}{K_B}+\frac{P_{AB}(V)}{K_A}\right)}{(L_A+L_B)\big(1-P_{AB}(V)\big)}}} \tag{32}$$

$$T_{\psi\delta}=\frac{mV\left(\frac{L_A}{K_B}-P_{AB}(V)\frac{L_B}{K_A}\right)}{(L_A+L_B)(1-P_{AB}(V))} \tag{33}$$

3. CONTROLLER STRUCTURE

The mechatronic control system is based on a controller that consists of a reference signal generator and regulators' system. When designing the reference signal generator for the lane change controller, particular attention should be paid to „bang-bang" input waveforms (Fig. 9), which are close to the actual waveforms during this manoeuvre.

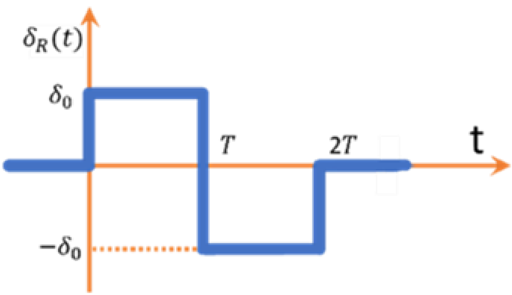

Fig. 9. "Bang-bang" signal [3]

$$\delta(t)=\delta_0(1(t)-2\cdot 1(t-T)+1(t-2T) \tag{34}$$

$$\delta(s)=\delta_0\frac{(1-e^{-sT})^2}{s} \tag{35}$$

The limits of the transmittance should be analysed at this point, as they will provide information on the final values for the individual displacements.

$$\delta_A(t) = \delta(t) \tag{36}$$

when
$$\delta_B(t) = P_{AB}(V)\delta(t) \tag{37}$$

$$\lim_{t\to\infty}\big(\psi(t)\big) = \lim_{s\to 0}\big(s\psi(s)\big) = \lim_{s\to 0}\Big(s(G_{\psi\delta_A}(s)\delta_A(s) + G_{\psi\delta_B}(s)\delta_B(s)\Big) = 0 \tag{38}$$

$$\begin{gathered}\lim_{t\to\infty}\big(Y(t)\big) = \lim_{s\to 0}\big(sY(s)\big) = \lim_{s\to 0}\Big(s\Big(G_{Y\delta_A}(s)\delta_A(s) + G_{Y\delta_B}(s)\delta_B(s)\Big)\Big) \\ = \big(1 - P_{AB}(V)\big)\delta_0 T^2 V K_0 = Y_0\end{gathered} \tag{39}$$

A real-time (online) controller must have a relatively simple model in its structure in order to be able to process all the information on the basis of the set parameters. Due to this fact, it was decided to simplify the existing transmittance form of the model even further. In this way, an extremely simplified model was obtained, omitting members that only affect transient states, which enables us analytical synthesis of the control of classic 4WS vehicles.

$$G_{Y\delta_R}(s) = \frac{\big(1 - P_{AB}(V)\big)VK_0}{s^2} \tag{40}$$

$$G_{\psi\delta_R}(s) = \frac{\big(1 - P_{AB}(V)\big)K_0}{s} \tag{41}$$

In the case of lateral displacement, the final value is the final value of the lateral displacement of the vehicle and it is Y_0. In the case of angular displacement, on the other hand, the final value is 0 and it indicates that the vehicle has aligned itself parallel to the original track after the manoeuvre. Unfortunately, this does not provide information on by what maximum angle the vehicle has deviated during the manoeuvre. In the case of angular displacement, information is mainly needed about the peak value, because here it does not coincide with the final value as in the case of lateral displacement. Therefore, the analysis was performed again at $t = T$ using (41). This provided information on the angular displacement value $\psi(T) = \psi_0$ for the maximum (peak) value ψ_0. This procedure made it possible to provide information on the basic parameters of the control signal (δ_0 and T) and thus the form of the other reference signals (Fig. 10).

$$T = \frac{Y_0}{V\psi_0} \tag{42}$$

$$\delta_0 = \frac{V\psi_0^2}{K_{\psi\delta}Y_0} \tag{43}$$

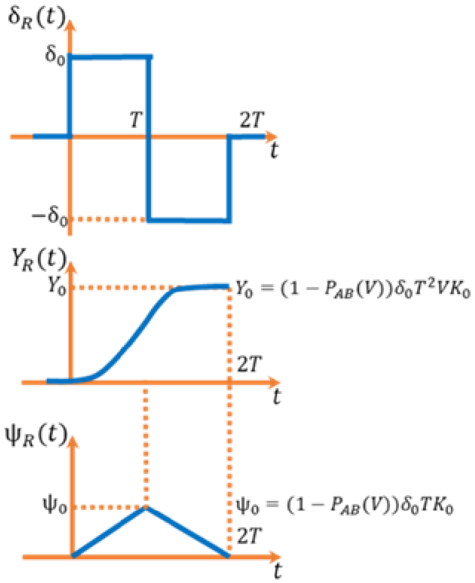

Fig. 10. Reference signals in the control system [4]

The above theoretical analysis allows us to show the final form of the reference signal generator in the 4WS vehicle lane change mechatronic controller (Fig. 11).

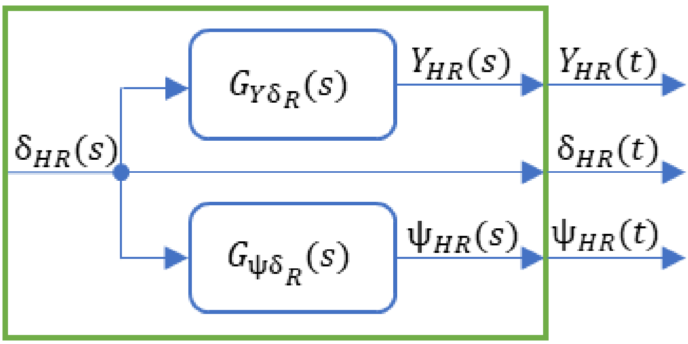

Fig. 11. Reference signal generator [4]

$$\delta_{HR}(s) = p \cdot \delta_R(s) \tag{44}$$

where p is the steering ratio, and the signals $Y_{HR}(s)$ and $\psi_{HR}(s)$ are the effect of the signal $\delta_{HR}(s)$ on the extreme simplified transmittances. The following figure shows the representation of the steering signal in a lane change manoeuvre (Fig. 12) [4].

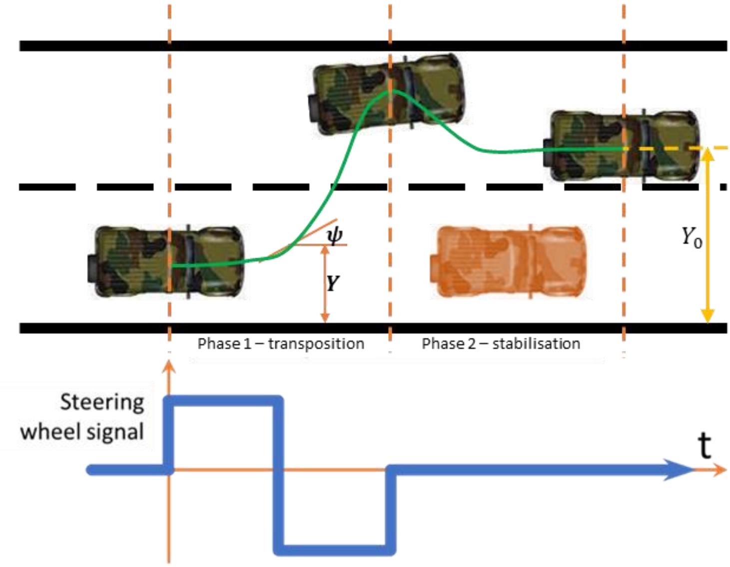

Fig. 12. Lane change process including control signal

The next stage in mechatronic controller design is to calculate the form of the regulators' system. As already mentioned, this stage will be carried out based on the linear quadratic regulator (LQR) technique. This problem was solved by Kalman, hence the customary designation of the regulators as „Kalman's regulators". Reduced transmittance forms will be used again for this study as, due to their simplified form, the calculations will be much less complicated and fully solvable. The mathematical notation of LQR theory in the form of the equations of state is presented below:
Model:

$$\dot{\boldsymbol{x}}(t) = \boldsymbol{A}\boldsymbol{x}(t) + \boldsymbol{B}\hat{\boldsymbol{u}}(t),\ \boldsymbol{x}(0) = \boldsymbol{0} \tag{45}$$

Quality indicator

$$J = \frac{1}{2}\int_0^{\infty} (\boldsymbol{x}(t)^T\boldsymbol{Q}\boldsymbol{x}(t) + \hat{\boldsymbol{u}}(t)\boldsymbol{R}\hat{\boldsymbol{u}}(t))dt \tag{46}$$

where: Q, R – the positively determined weight matrices.
Solution:

$$\hat{u}(t) = -\boldsymbol{R}^{-1}\boldsymbol{B}^T\boldsymbol{K}\boldsymbol{x}(t) \tag{47}$$

where: K – the symmetric matrix satisfying the Riccatti's equation:

$$-\boldsymbol{K}\boldsymbol{A} - \boldsymbol{A}^T\boldsymbol{K} + \boldsymbol{K}\boldsymbol{B}\boldsymbol{R}\boldsymbol{B}^T\boldsymbol{K} = \boldsymbol{Q} \tag{48}$$

The process of determining the form of the regulators was carried out separately for the transposition process and stabilisation of the lane change manoeuvre.

At the outset, it is necessary to determine the designations of the individual elements of the above equations for the transposition process.

$$x_1(t) = \Delta Y(t) \tag{49}$$

$$x_2(t) = \Delta \dot{Y}(t) \tag{50}$$

$$\hat{u}(t) = K_{Y\delta}\Delta\delta_Y(t) \tag{51}$$

The form of the model must then be presented in terms of the equations of state.

$$\dot{x}_1(t) = x_2(t) \tag{52}$$

$$\dot{x}_2(t) = \hat{u}(t) \tag{53}$$

$$x(0) = 0 \tag{54}$$

The form of the quality indicator should then be defined.

$$J = \frac{1}{2}\int_0^{\infty} \left(p_1 x_1^2(t)^T + p_2 x_2^2(t) + \hat{u}^2(t)\right) dt \tag{55}$$

After substituting all elements into the solution and calculating the complex Riccatti's equation, the final form of the first regulator equations for the transposition process is obtained.

$$\Delta\delta_Y(t) = -\frac{\sqrt{p_1}}{K_{Y\delta}}\Delta Y(t) - \frac{\sqrt{p_2 + 2\sqrt{p_1}}}{K_{Y\delta}}\Delta\dot{Y}(t) \tag{56}$$

The above form of regulator consists of two members – a proportional and a differential – resulting in a PD regulator. In a similar way, calculations should be made for the stabilisation process in order to determine the form of the regulator.

Designations:

$$x(t) = \Delta\psi(t) \tag{57}$$

$$\hat{u}(t) = K_{\psi\delta}\Delta\delta_\psi(t) \tag{58}$$

Model:

$$\dot{x}(t) = \hat{u}(t) \tag{59}$$

$$x(0) = 0 \tag{60}$$

Quality indicator:

$$J = \frac{1}{2}\int_0^{\infty}\left(p_3 x^2(t) + \hat{u}^2(t)\right)dt \tag{61}$$

After substituting all the necessary elements into the solution, the regulator form for the stabilisation process is obtained.

$$\Delta\delta_{\psi}(t) = -\frac{\sqrt{p_3}}{K_{\psi\delta}}\Delta\psi(t) \tag{62}$$

The regulator for the stabilisation process contains only a proportional member, which indicates that it is the *P* regulator. It is worth noting that an additional parameter is defined for each of the regulator settings (p_1, p_2, and p_3). Among other things, this allows the individual regulator settings to be adjusted independently of each other and thus, the values of the regulators themselves.

4. STRUCTURE OF THE 4WS VIRTUAL VEHICLE MODEL

Since the authors do not have an actual 4WS vehicle, there is a need to create a virtual facility. It is assumed that the vehicle, when changing lanes, travels at a constant speed on a level road without excessive skidding and it does not brake in front of an obstacle, but avoids it immediately. As a result of this assumption, the influence of the braking and suspension systems can be ignored in the construction of the virtual model due to the small nature of the excursions. In addition, during lane changes, yaw angles of the vehicle are small (on the order of a few degrees), making side-to-side tilt of the vehicle of little importance, so this element can also be ignored. On the other hand, the steering system plays an important role when vehicle deflection is small, hence the 4WS virtual vehicle model consists of a steering system and a vehicle movement system. The vehicle movement system is described by a non-linearised 4WS bicycle model with the effect of crosswind forces F_p (as a result of the assumptions described earlier) – Fig. 13.

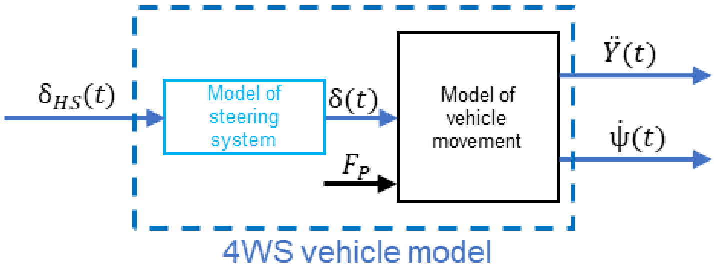

Fig. 13. Virtual 4WS vehicle [4]

The model of steering system is presented below [20]:

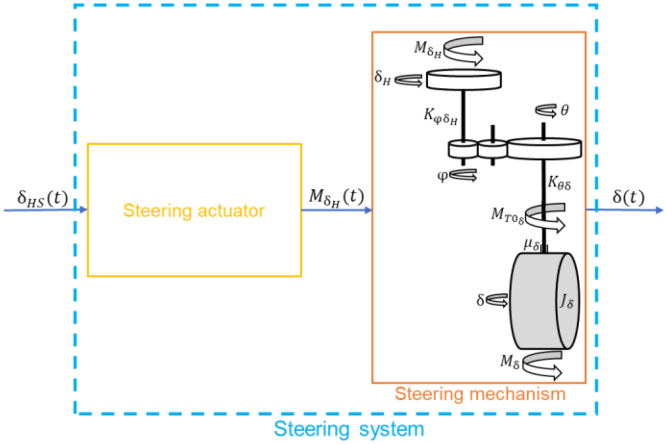

Fig. 14. Steering system of the 4WS vehicle [4]

$$G_{M_{\delta_H}\delta_{HS}}(s) = \frac{K_{M_{\delta_H}\delta_{HS}}}{T^2_{M_{\delta_H}\delta_{HS}}s^2 + 2\xi_{M_{\delta_H}\delta_{HS}}T_{M_{\delta_H}\delta_{HS}}s + 1} \tag{67}$$

(steering actuator - second order inertial member)

$$J_\delta\ddot{\delta}(t) = -\mu_\delta\dot{\delta}(t) + M_\delta(t) + \frac{(K_{\varphi\delta_H} + K_{\theta\delta})(\delta_H(t) - p\delta(t))}{pK_{\varphi\delta_H} + K_{\theta\delta}} \tag{68}$$

(steering mechanism)

where: J_δ – the moment of inertia of the steering knuckle with the wheel, μ_δ – the damping factor in the steering knuckle bearing, M_δ – the moment of external force acting on the steering knuckles, $M_{T0\delta}$ – the moment of friction of the dry bearing of the steering knuckle, $K_{\varphi\delta H}$ – the stiffness coefficient of the steering shaft, $K_{\theta\delta}$ – the stiffness coefficient of the shaft representing the rod, p – the gear ratio, δ – the angle of rotation of the road wheel knuckle, θ – the angle of rotation of the gear wheels from the knuckles, φ – the angle of rotation of the gear wheel from the steering wheel side, and δ_{HS} – the handlebar rotation angle.

5. SIMULATION STUDIES

The considerations presented so far allow the full structure of the case study mechatronic control system to be determined (Fig. 15).

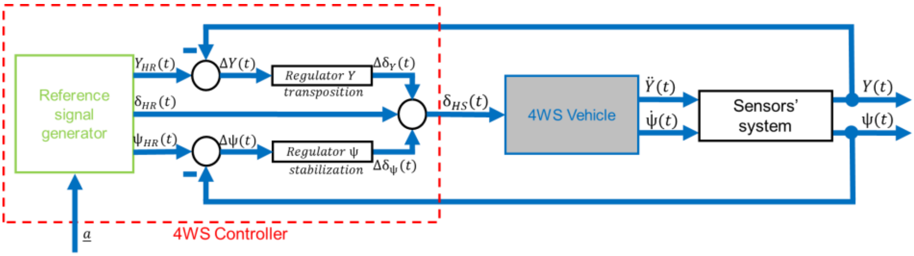

Fig. 15. Full structure of mechatronic control system [4]

As a result of the unavailability of a real vehicle, simulation studies were carried out based on a data record partly from literature [20] and partly from own research (not mentioned here). The necessary data for the simulation are shown in the following table (Table 1).

Table 1. Data of simulations [3, 4, 11, and 20].

Parameter	Unit	Value
V	[m/s]	21.7
m	[kg]	1627
J	[kg·m^2]	2893
L_A	[m]	1.15
L_B	[m]	1.56
K_A	[N/rad]	57719
K_B	[N/rad]	80723
V_0	[m/s]	15
ΔV	[m/s]	5
P_{AB0}	[-]	0.1
Y_0	[m]	3.5
Ψ_0	[rad]	0.17
J_δ	[kg·m^2]	0.2
μ_δ	[Nms/rad]	100
M_δ	[Nm]	0
$M_{T0\delta}$	[Nm]	0
$K_{\varphi\delta H}$	[Nm/rad]	100
$K_{\theta\lambda}$	[Nm/rad]	1×10^{11}
$K_{M\delta H\delta HS}$	[-]	1
$T_{M\delta H\delta HS}$	[-]	0.1
$\xi_{M\delta H\delta HS}$	[-]	0.7
p	[-]	16.4

At first, the test was carried out in an open-loop (without feedback) system for one speed (21.7 m/s ≈ 80 km/h), in order to confirm the necessity of the proposed mechatronic system (Fig. 16)

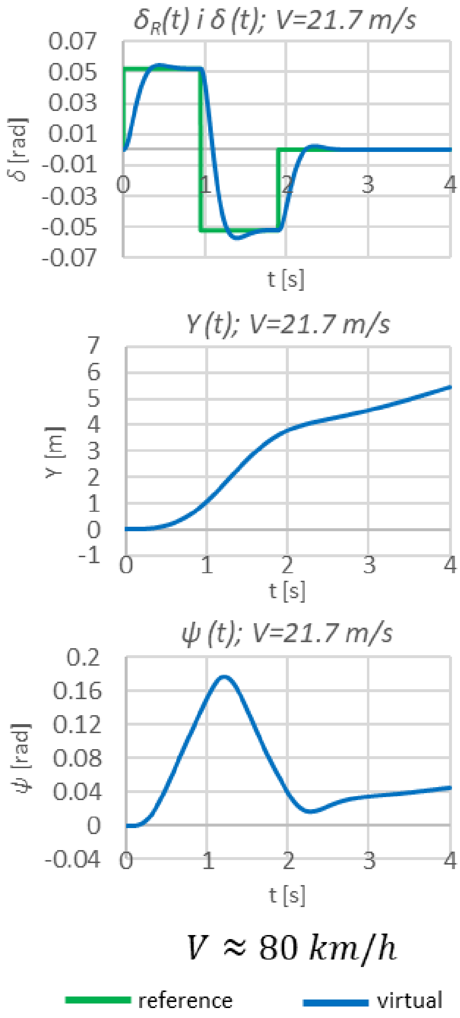

Fig. 16. Result of simulations in open-loop system [4]

In an open-loop system, the vehicle is unable to achieve the assumed steady-state values, hence the need for a closed-loop system controller (as proposed so far).

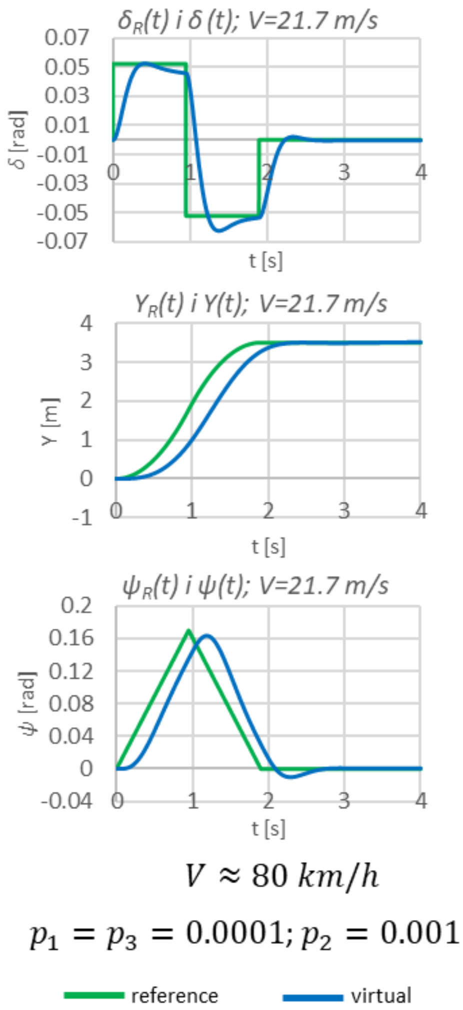

Fig. 17. Result of simulations in closed-loop system

The proposed mechatronic control system functions correctly. The responses of the system reach the assumed peak and final values for the manoeuvre.

6. SUMMARY

The proposed mechatronic control method is important because it considers the control of a vehicle at the highest level of automation (level 5 according to NHTSA – National Highway Traffic Safety Administration) based on the fundamental manoeuvre, which is changing lanes and out of which other even more complicated ones are composed.

This method is effective and reasonable in application. This is confirmed by simulation studies conducted for an example speed. The advantage of using this method is that the calculated regulators are able to properly correct the failures occurring in the system, so that the actual signals are as close to the reference ones as possible. This represents a superiority over open-loop system, which without determining the appropriate reference signals describing the trajectory of movement and a properly selected system of regulators, may not be able to correct the trajectory due to crosswinds etc. It is worth noting that the form of the regulators was not assumed in advance, but it results from the calculations performed (solution of the LQR task). The disadvantage of this mechatronic system may be the low sensitivity, to any imperfections that may occur, such as noise or offset. In future work, sensitivity studies should be conducted to find out what imperfections the algorithm is sensitive to. Such studies will show what elements need to be improved in order to make the mechatronic system function fully correctly.

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Metoda sterowania pojazdem autonomicznym z wykorzystaniem uproszczonych modeli referencyjnych oraz regulatorów

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Streszczenie. Niniejsze opracowanie zawiera rozwiązanie koncepcyjne metody sterowania pojazdem autonomicznym w oparciu o uproszczony model referencyjny oraz regulatory. Całe zagadnienie jest rozważane na przykładzie fazy wykonawczej manewru zmiany pasa ruchu, w którym to wykorzystano powszechnie znany model rowerowy 4WS (Four-Wheel Steering) oraz regulatory Kalmana oparte na technice LQR (Linear-Quadratic Regulator). Całościowo opracowany mechatroniczny układ sterowania poddano badaniom symulacyjnym w programie Matlab&Simulink. Wyniki symulacji pokazały poprawne funkcjonowanie całego mechatronicznego układu regulacji oraz pozwoliły na określenie dalszych kierunków badań. Opracowana metoda sterowania może również znaleźć zastosowanie w pojazdach wojskowych.

Słowa kluczowe: inżynieria mechaniczna, pojazd autonomiczny, pojazd 4WS, zmiana pasa ruchu, model rowerowy.