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ПРИМЕНЕНИЕ MPC И PID ДЛЯ УПРАВЛЕНИЯ ДВИЖЕНИЕМ АВТОМОБИЛЯ ПО ЗАДАННОЙ ТРАЕКТОРИИ

THE USE OF MPC AND PID TO CONTROL THE MOVEMENT OF THE CAR ALONG A GIVEN TRAJECTORY

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В эпоху технологических достижений исследование и применение эффективных методов управления в автомобилях становятся важнейшей областью, охватывающей не только автономные транспортные средства, но и другие интеллектуальные функции. Данная работа рассмотривается применении комбинации техник PID (пропорционально-интегрально-дифференциальное управление) и MPC (управление на основе модели предсказания) для повышения эффективности управления движением автомобилей, особенно при отслеживании заданных траекторий. В исследовании, приведется динамическая модель автомобиля, на основании которой дается оценка эффективности работы комбинации PID и MPC о минимизации отклонения между фактической траекторией движения и заданной траекторией при поддержании постоянной скорости автомобиля. Результаты показывают, что этот интегрированный подход обеспечивает значительно лучшую эффективность по сравнению с использованием только методов PID или MPC.

In the era of technological advancements, the research and application of efficient control techniques in automobiles are emerging as a crucial field, not only focusing on autonomous vehicles but also encompassing other intelligent functionalities. A recent study has employed a *combination of PID (Proportional-Integral-Derivative) and MPC (Model Predictive Control) techniques to augment the performance of motion control in vehicles, particularly in tracking predefined trajectories. By utilizing a dynamic kinematic model, the study assessed the efficacy of the PID and MPC amalgamation in minimizing the deviation between the actual motion trajectory and the predetermined reference trajectory, while maintaining a constant vehicle velocity. The findings reveal that this integrated approach yields superior effectiveness compared to solely employing either PID control or MPC methods.*

Ключевые слова: *траектория движения, PID, MPC, динамика поворота автомобиля.*

Keywords: *motion trajectory, PID, MPC, vehicle cornering dynamics.*

INTRODUCTION

The research and application of effective control techniques on automobiles is becoming an important field, not only for autonomous vehicles but also for other intelligent functions that are being developed. One of the important problems in this field is to ensure that the vehicle's motion follows a predetermined trajectory, which is determined by the sensors installed on the vehicle (Radar, Lidar, Camera, GPS and others) and the model predictive controller (MPC) combined with the PID controller. When controlling a vehicle to follow a predetermined trajectory with an allowable error (according to SAE is 150 mm), it is necessary to take into account the effect of the lateral tire force generated during turning [1].

When turning, the vehicle's motion follows certain rules, affected by the steering angle and the lateral deformation of the tire during turning [2–3]. When there is a lateral force, the vertical reaction force from the road surface acting on the two wheels of the same axle changes, affecting the ability to transmit traction and braking force, and even causing dangers such as skidding or tipping the vehicle sideways. The lateral deformation of the tire during steering can cause the vehicle's trajectory to deviate when it is controlled automatically. This paper studies the ability of the vehicle to follow a predetermined trajectory in the case of parking and double lane change at different speeds, taking into account the lateral deformation of the tire during steering. The modeling method is chosen to carry out the above research.

MODELING DEVELOPMENT

Modeling of the cornering dynamics of the vehicle.

Based on prior research in electric vehicles [4–5], the paper presents the methodology for constructing a dynamic model of a front-wheel-drive electric vehicle based on the vehicle's dynamic equations.

The bicycle model can be used to investigate the cornering dynamics of vehicles in motion trajectory problems [2–4] (fig. 1).

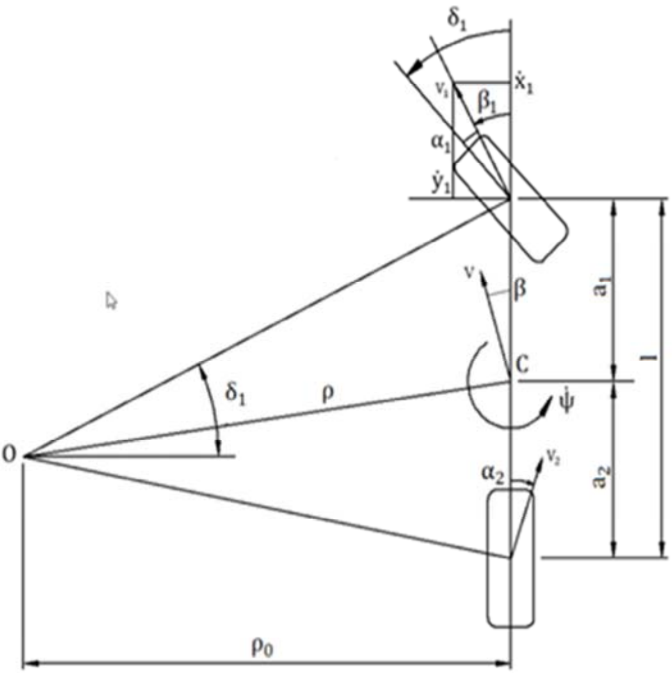

Figure 1 – Vehicle cornering model

Notation in fig. 1: β – yaw angle; δ – steering angle; v_1, v_2 – front and rear wheel velocities; a_1, a_2 – distances from the vehicle's center of gravity to the front and rear axles; l – wheelbase; ψ – vehicle body angle; ρ – turning radius; C – center of gravity; α_1, α_2 – front and rear lateral tire slip angles.

The equations of motion are obtained as follows:

$$\dot{y} = v_y + v_x \cdot \psi ; \tag{1}$$

$$\begin{bmatrix} \dot{v}_y \\ \ddot{\psi} \end{bmatrix} = \begin{bmatrix} \dfrac{-C_{\alpha 1} - C_{\alpha 2}}{m v_x} & \dfrac{-a_1 \cdot C_{\alpha 1} + a_2 \cdot C_{\alpha 2}}{m v_x} - v_x \\ \dfrac{-a_1 \cdot C_{\alpha 1} + a_2 \cdot C_{\alpha 2}}{I_z \cdot v_x} & \dfrac{-a_1^2 \cdot C_{\alpha 1} - a_1^2 \cdot C_{\alpha 2}}{I_z \cdot v_x} \end{bmatrix} \cdot \begin{bmatrix} v_y \\ \ddot{\psi} \end{bmatrix} + \begin{bmatrix} \dfrac{C_{\alpha 1}}{m} \end{bmatrix} \cdot \delta ; \text{ (2))}$$

$$\begin{bmatrix} \dot{\beta} \\ \ddot{\psi} \end{bmatrix} = \begin{bmatrix} \dfrac{-C_{\alpha 1} - C_{\alpha 2}}{m \cdot v_x} & \dfrac{-a_1 \cdot C_{\alpha 1} + a_2 \cdot C_{\alpha 2}}{m \cdot v_x^2} - 1 \\ \dfrac{-a_1 \cdot C_{\alpha 1} + a_2 \cdot C_{\alpha 2}}{I_z} & - \dfrac{-a_1^2 \cdot C_{\alpha 1} - a_1^2 \cdot C_{\alpha 2}}{I_z \cdot v_x} \end{bmatrix} \cdot \begin{bmatrix} \beta \\ \dot{\psi} \end{bmatrix} + \begin{bmatrix} \dfrac{C_{\alpha 1}}{m} \\ \dfrac{a_1 \cdot C_{\alpha 1}}{I_z} \end{bmatrix} \cdot \delta . \tag{3}$$

The Vinfast VF e34 car was chosen as the research subject, and its technical specifications are presented in tab. 1.

Table 1 – Input simulation parameters

Parameter	Value	Unit	Parameter	Value	Unit
a_1	1,125	m	I_z	2875	kg/m^2
a_2	1,486	m	$C_{\alpha 1}$	500	N/degree
m	1850	kg	$C_{\alpha 2}$	833	N/degree

Modeling of MPC, PID, and combined controllers.

At each sampling time, the future control signals are calculated by optimizing the objective function *J(U)* in a similar form to the following equation:

$$J = \sum_{k=1}^{N} L(X(t+k/t), Y(t+k/t), u(t+k/t)) .$$

The PID control scheme is named after its three corrective terms, the sum of which is formed by the manipulated variable:

$$u(t) = MV(t) = K_p e(t) + K_i \int_0^t e(\tau) d\tau + K_d \frac{d}{dt} e(t) .$$

The predefined trajectory and actual parameters of the Y-coordinate and roll angle of the vehicle body are fed into the PID controller, and the output of the controller is the steering wheel angle. With the obtained steering wheel angle, the position of the vehicle's center of gravity can be determined through the dynamic model of the vehicle's yaw motion.

MPC-PID Combined Control Model

The combination of the two MPC and PID models improves the system by overcoming the drawbacks of each individual model. The model of the system with the combination of MPC and PID is presented in fig. 2.

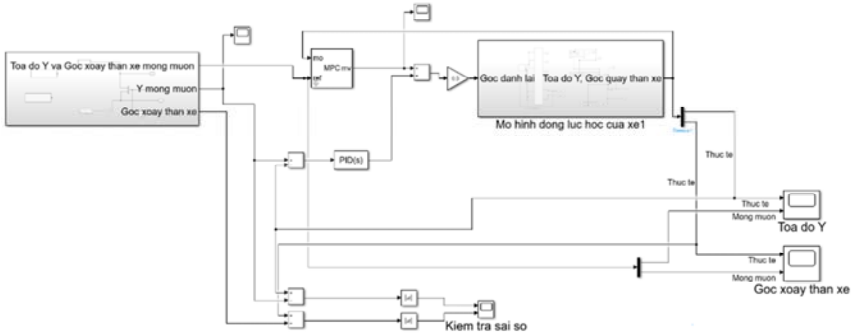

Figure 2 – Simulation Model of the Combined PID and MPC Controller

EXPERIMENTAL RESULTS AND ANALYSIS

The simulated double lane change trajectory is based on the «ISO double lane change test» [7]. Simulation results of Double Lane Change at Vehicle Speed of 120 km/h are present in fig. 3 and tab. 2.

Table 2 – Simulation error results at 120 km/h

Parameter	MPC + PID Controller	PID Controller	MPC Controller
Maximum Lateral Deviation of the Vehicle Trajectory	0,127 m	0,256 m	0,661 m
Maximum Yaw Angle Error	0,251 rad	1,011 rad	0,245 rad

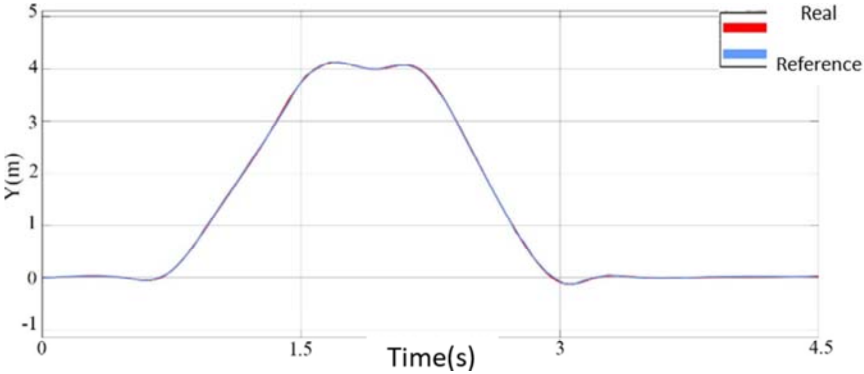

Figure 3 – Double Lane Change at 120 km/h using the Combined MPC and PID Controller

Simulation results of Right-Angle Turn at a Vehicle Speed of 100 km/h are present in fig. 4 and tabl. 3.

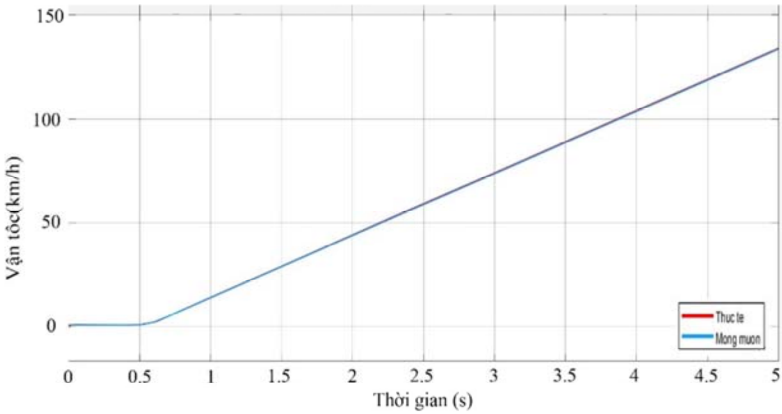

Figure 4 – Right-Turn Maneuver at 100 km/h using the Combined MPC and PID Controller

Table 3– Right-Turn Maneuver Error Results at 100 km/h

Parameter	MPC + PID Controller	PID Controller	MPC Controller
Maximum Lateral Deviation of the Vehicle Trajectory	0,130 m	0,415 m	0,710 m
Maximum Yaw Angle Error	0,320 rad	0,767 rad	0,326 rad

The results show that when performing double lane change and right-angle turn with constant longitudinal velocity, the lateral deviation using PID and MPC controllers is significant. According to the SAE standard for deviation of 150 mm, this indicates that at this speed, the performance of the PID or MPC controller alone is no longer guaranteed. Meanwhile, using the combined MPC and PID controller results in a maximum lateral deviation of 130 mm. This error falls within the SAE recommended range.

CONCLUSION

The construction of a combined MPC and PID controller for trajectory tracking leads to the conclusion that the controller satisfies the tracking of a predefined trajectory in the case of parking or lane changing at low speed, and is also effective compared to using only the MPC or PID controller alone. However, the controller does not operate completely accurately when the vehicle is moving at high speed. Based on this conclusion, future research will focus on optimizing or combining other controllers to achieve accurate trajectory tracking at high speeds.

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ВЛИЯНИЯ ИЗМЕНЕНИЯ ЖЕСТКОСТИ ШИН ПРИ ИЗМЕНЕНИИ НАГРУЗКИ НА КОЛЕБАНИЯ ЛЕГКОВЫХ АВТОМОБИЛЕЙ

EVALUATING THE IMPACT OF TIRE STIFFNESS VARIATION DUE TO LOAD CHANGES ON PASSENGER CAR OSCILLATIONS

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Tires are the components that make contact with the road surface, and their condition is one of the factors affecting the smoothness of the vehicle. This paper investigates the influence of tire stiffness variation on the performance of the suspension system. The study employs a combination of experimental and simulation methods to examine how changes in pressure and load lead to changes in tire stiffness, thereby affecting vehicle body oscillation. Experiments were conducted to determine tire stiffness, and the experimental results were incorporated into a quarter-car suspension system simulation model for further investigation. The