[1.]Ali NAJARI, [2.]Fereshteh SHABANI, [3.]Mehdi HOSSEYNZADEH

INTEGRATED INTELLIGENT CONTROL SYSTEM DESIGN TO IMPROVE VEHICLE ROTATIONAL STABILITY USING ACTIVE DIFFERENTIAL

[1.]Sapienza University of Rome, Department of Computer, Control and Management Engineering, Roma, ITALY
[2,3.]Iran University of Science and Technology, Department of Electrical Engineering and Industrial Informatics, Tehran, IRAN

Abstract: This paper examines the improvement of the car's dynamic performance using a two-layer intelligent control system to control its direct torque. For this purpose, an active differential system is installed, which is located on the rear axle of the car. The top layer of this controller is produced using the optimal controller method, the values of the torque transmitted in the rear differential, and the torque produced. This system announces the necessary information by the front-wheel brake to track the desired values of the car to the subsystems. In the second layer, according to these two values, the transfer torque in the active differential and the brake torque is applied to the front wheels of the car. The simulated results of the 9-DOF model of the car show that the designed controller has the ability to maintain the car's stability in all driving and road conditions.
Keywords: car's dynamic performance, active differential system, simulation

INTRODUCTION

The advancement of technology in the automotive industry in the last three decades and the importance of safety for carmakers have led to the development of a variety of advanced electronic systems (mechatronics) that control the automobiles. Also, this impact can be considered as an influential factor in purchasing and choosing a car. The most considerable part of research and development in the field of the automotive industry occurs in order to sustain and improve vehicle handling in critical condition. To improve safety and reliability for semi-autonomous vehicles, there are two main aspects, the microscopic view, and macroscopic view [1-2]. In the microscopic view, steering control for lane-keeping/changing [3-4] and control the authority of the drivers [5] are studied. Before the advent of active safety systems, driver's driving skills was the only factor in preventing accidents, and the driver was responsible for maintaining and controlling vehicle stability. However, drivers have considered the installation of anti-lock brakes as a safety device to prevent accidents in the past two decades, today their expectations, in the existence of electronic controllers, have risen to the point where they are looking to balance their cars with these systems. Due to the fact that most of the forces on the car are generated by the tires, the force of each tire produces torque around the center of mass of the vehicle. Therefore, the control of the car's torque is closely related to the creation of longitudinal and transverse forces in the tires.

Longitudinal force in tires is created by applying thrust or brake torque and producing longitudinal slip in tires. While transverse force is produced by applying the steering angle and creating the slip angle in the tires, therefore, to control the torque in the car by creating an active chassis control of steering or thrust or integration of both systems, the control can be realized. One of the stability control systems using the transverse force of the tires is the active vehicle steering system.

The active steering controls the front or rear wheels or both according to the car's side dynamics and what the driver wants [6]. Asymmetric braking is the most effective and common method of creating torque in vehicles due to the use of an anti-lock brake subsystem and its effectiveness in maintaining vehicle stability in critical situations [7]. Another way to generate torque is to use an active differential system. As illustrated in Figure (1), the application of asymmetric driving torque to the drive cycles also creates torque. This mechanism during vehicle acceleration, in addition to system stability, causing the vehicle to accelerate further. This is the most important advantage of active differential compared to the conventional stability control systems, which cause longitudinal deceleration by braking [8].

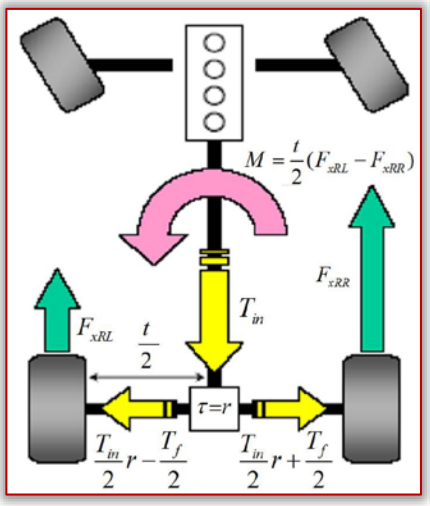

Figure 1. Vehicle stability with active differential [8]

Tomari et al. (2006) [9] explain how to simultaneously use the brakes and the active differential to apply the torque in a four-wheel-drive Honda. In this car, the central differential is of the planetary gear type, and the rear differential is of the limited electric slip type.

Rajesh Rajmani et al. (2006) [10] have used central coupling and active rear differential to control the car's dynamics. Their research focuses on differential modeling in a front-wheel-drive vehicle that is converted into the four-wheel drive as needed. In fact, torque control is done by distributing the thrust force on each wheel. The central coupling and rear differential, which have electric clutches, transmit torque to each wheel.
Lee et al. (2006) introduced a four-wheel-drive system using the central differential slip differential without the CC-LSD clutch to improve control of thrust force. In this system, a planetary gear and a Gerotor are used to divide the torque between the front and rear axles of the vehicle. A PI controller is used to calculate the control input, which in this system is the inlet pressure to the pump. The simulations show that cars equipped with the CC-LSD system have more thrust force than rear-wheel drive vehicles [11-12].
Canale et al. (2007) introduced a robust control for the car's rotational stability control system based on the Internal Motor Control (IMC) method. Simulations with the 14-degree freedom model of car show a significant improvement in vehicle stability in hazardous conditions such as low-friction road traffic in a vehicle equipped with this system [13].
Rubin et al. (2013) designed a sliding mode controller for an active differential system to control the car's rotational stability. Simulation with CarSim software showed that the designed control increases vehicle stability in acceleration at the corners [14].
The purpose of this paper is to design an integrated intelligent control system for direct torque resulting from the active differential in the rear wheels and the braking in the front wheels. So that this controller can track the desired values of the rotational angle speed and the side sliding angle of the car. In addition, by eliminating the weaknesses caused by the two active steering control systems and direct torque, their performance and thus the stability and maneuverability of the car will be improved.
Using the rotational angular speed and the actual side angle of the car as the input of the controller, the rear differential torque and the brake torque determine the front wheels to track the desired values. In fact, the car is rear-wheel drive, and the braking difference is only made at the front wheels. The torque of the rear wheels is determined by the engine torque and the transmission torque in the differential. In fact, the clutch mounted on the differential allows the total engine torque to be transmitted to the left or right wheel.

VEHICLE MODELING

A dynamic model with nine degrees of freedom has been used for car modeling. This model has degrees of freedom including longitudinal velocity (u), lateral velocity (v), rotational angle velocity (r), body roll angle and spinning mass (ϕ), four degrees of freedom related to the rotational speed of each wheel (ω_i) and one degree of freedom of the system is the front angle steering angle (δ_f). Figure (1) shows a general schematic of the vehicle. The equations of the vehicle motion are given in detail in reference [14].

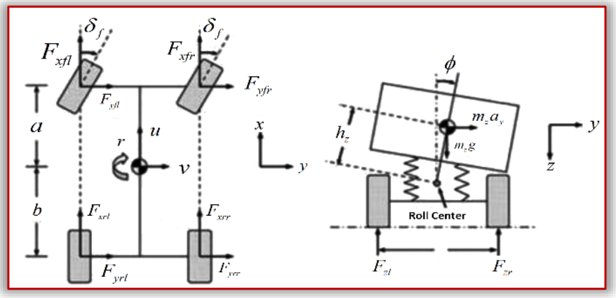

Figure 2. Nine degree model of car

— Tire modeling

To model the tire from the nonlinear model, the formula using longitudinal slip, slip angle, camber angle, and vertical tire force as the input of the longitudinal, transverse, and torque alignment of the tire is considered as the best nonlinear model for tire modeling. A description of its details is used in reference [15]. The reason for using the hybrid model is to limit the production of tire power when the longitudinal and lateral forces of the tire are produced simultaneously.

— Differential modeling

The differential used in this study is a conventional differential with two dry electromagnetic clutches. The use of these clutches enables the differential to independently control the torque of each axle. The schematic of this system is shown in Figure (2). In this research, information about the torque of a gasoline engine in the whole field of work is used.

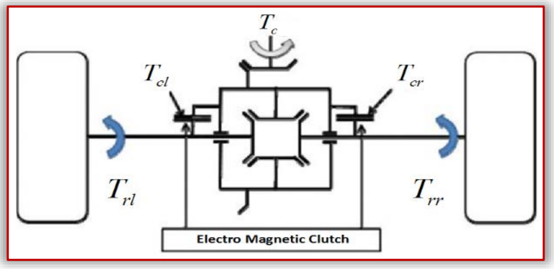

Figure 3. Schematic of active differential

The equations governing the system are shown in Equations (1) to (3). If the right-hand clutch transmits torque T_{cr} to the right-wheel drive torque increases by $\frac{T_{cr}}{2}$ and the left-wheel drive torque decreases by $\frac{T_{cr}}{2}$. According to the figure (2), the torque on the rear wheels and the torque between the two wheels can be calculated as follows:

$$T_{rr} = \frac{T_c - T_{cl} + T_{cr}}{2} \quad (1)$$

$$T_{rl} = \frac{T_c + T_{cl} - T_{cr}}{2} \quad (2)$$

$$T_{diff} = T_{cl} - T_{cr} \quad (3)$$

CONTROLLER DESIGN

The four-degree linear freedom model is used to design the controller. The degrees of freedom of the model are the transverse vehicle speed, rotational speed, two degrees of freedom related to the rotational speed. The system's control input is T_{diff} and direct rotational torque due to differential braking is M_{zb}. State equation of the system is

$$[\boldsymbol{M}]_{4\times4}[\dot{\boldsymbol{X}}]_{4\times1} = [\boldsymbol{A}]_{4\times4}[\boldsymbol{X}]_{4\times1} + [\boldsymbol{B}]_{4\times2}[\boldsymbol{U}]_{2\times1} \quad (4)$$

$$\boldsymbol{X} = [v \quad r \quad \omega_{rl} \quad \omega_{rr}]^T, \quad \boldsymbol{B} = \begin{bmatrix} 0 & 0 \\ 0 & 1 \\ 1 & 0 \\ -1 & 0 \end{bmatrix}, \quad \boldsymbol{U} = \begin{bmatrix} T_{diff} \\ M_{zb} \end{bmatrix} \quad (5)$$

The controller is a two-layer type that the upper layer uses the optimal controller technique to announce the torque values of the rear differential clutch and the torque generated by the front-wheel brake system to track the car's optimal dynamic behavior to the subsystems. The goal of the first layer controller is to maximize the adaption of the actual car's behavior to the desired behavior. This is possible by minimizing the following integral value (performance index), which is the same mathematical form of the measurement of lateral dynamic measurement [16]:

$$J = \frac{1}{2}\int_0^\infty \left((r - r_d)^2 + w_1 v^2 + w_2 T_{diff}^2 + w_3 M_{zb}^2 \right) dtc \quad (6)$$

Where T_{diff} the differential clutch torque and $M_{zb}s$ is the external torque generated by the brake, w_i are the desired weight coefficients, and the value (r_d) is equal to the stable value of the torque. The desire values of the states are the response of the stable state of the two freedom degree model of the car. In the second layer, according to these two sent values, the clutch force is applied to the active differential and the brake torque of the front wheels to the car. To solve and implement a real-time solver for an optimal problem with an associated cost function, we can use data compression for the send and receive data in the communication link. The time for producing compressed samples is an essential factor when we consider ambulatory devices, considering that data should be transferred to the controller in the real-time mode [17]. It is supposed that the structural monitoring and the system identification approach for the vehicle provides an accurate model for the system [18]. The details about the communication link, data compression, and system monitoring are considered in the ideal condition in the simulation section of this paper.

SIMULATION

In this section, simulation of vehicle motion in controller-free modes with integrated controller and active differential controllers (AD) and differential braking (ESP) is performed virtually, with an initial longitudinal velocity of 30 m/s and a fixed steering angle of 30 degrees. Three seconds after starting the simulation, the car suddenly enters the slippery road with a coefficient of μ = 0.2 from a dry road with a coefficient of friction μ = 0.9, the results are shown in Figures (4) to (7).

As can be seen in Figure (4), in a car without control, the speed has decreased due to the increase in longitudinal slip. While in AD controller mode, the car is still accelerating. ESP and integrated controllers have reduced longitudinal speed due to braking on the front wheels.

Table 1. Car information in the simulation

The distance between the two axles of the car	$L = 2.5(m)$
The distance from the front axle to the center of gravity	$L_f = 1.2(m)$
The distance from the rear axle to the center of gravity	$L_r = 1.3(m)$
Tire hardness front and rear	$C_{af}, C_{ar} = 45000(N/rad)$
Car mass	$M = 1300(kg)$
Moment of inertia around the z-axis	$I_z = 2500(kg.m^2)$

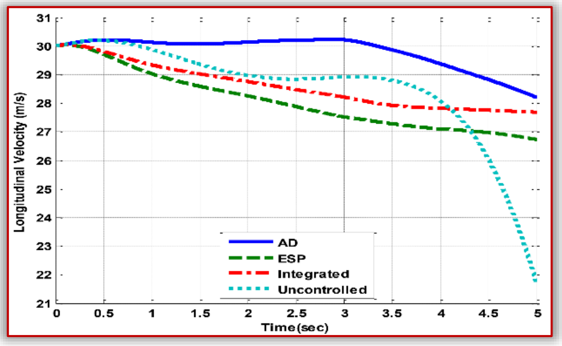

Figure 4. Longitudinal speed in the maneuver of sudden entry into the low friction road

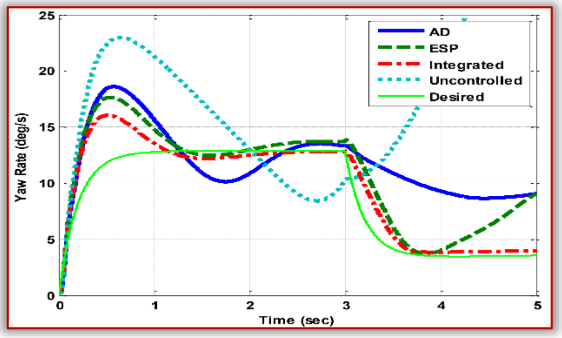

Figure 5. Rotational angular velocity

According to Figure (5), it is observed that a car without a controller, after a sudden entry into a low friction road, becomes unstable due to saturation of the transverse forces of the tire and its side sliding angle and rotational angle speed increase rapidly. While in a car with an integrated controller, the rotating angle speed and the side sliding angle of the car converge to the desired values. As can be seen, the braking has brought the actual values of the car closer to the desired values than the active differential.

Figure (6) shows the active differential transfer torque that results in asymmetric torque distribution in the right and left wheels. Since this torque depends on the engine torque, the smooth areas in this curve indicate the saturation of the clutch torque. Figure (7) shows the path of a car with an integrated controller without a controller when it suddenly enters the slippery road. As can be seen in the figure, an uncontrolled vehicle is unable to change lanes properly due to the slippery road and deviates to the left in the second cycle of lane changes. The controller prevents the car from turning around, but it is not able to correct the path, while the car with the ESP and integrated controller changes the line with the least deviation from the path.

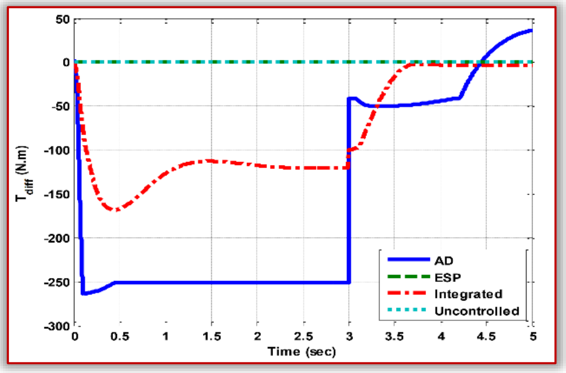

Figure 6. Transfer torque in active differential

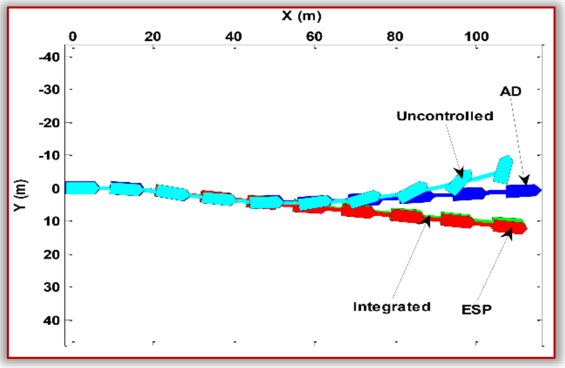

Figure 7. The trace of the vehicle

CONCLUSION

In this study, the control of the vehicle rotational stability is discussed using the active rear axle differential and the front wheel brake system. According to the simulations, it is concluded that the integrated control system has the ability to maintain stability and track the desired values of rotational angle speed and side sliding angle of the car. Due to the high cost of the active steering system in the car and the intangible role of this system in improving transverse dynamic behaviour, the use of this system is uneconomical practically. Although the ESP system is able to stabilize the car, it destroys the longitudinal dynamics. While active differential covers this defect. In addition, the active differential has a significant effect on improving the vehicle's longitudinal dynamics, especially at high speeds. Therefore, the use of an integrated controller in the form of an active differential combination and ESP system can be useful in improving longitudinal dynamics and vehicle stability in hard maneuvers as well.

Note: This paper is based on the paper presented at International Conference on Applied Sciences – ICAS 2020, organized by University Politehnica Timisoara – Faculty of Engineering Hunedoara (ROMANIA) and University of Banja Luka, Faculty of Mechanical Engineering Banja Luka (BOSNIA & HERZEGOVINA), in Hunedoara, ROMANIA, 09–11 May, 2020.

References

[1] Karimi Shahri P, Chintamani Shindgikar S, HomChaudhuri B and Ghasemi A, 2019 *Optimal Lane Management in Heterogeneous Traffic Network*, In Dynamic Systems and Control Conference (Vol. 59162, p. V003T18A003). American Society of Mechanical Engineers.

[2] Izadi V, Ghasemi A H and Shahri P K, 2020 *Negotiating the Steering Control Authority within Haptic Shared Control Framework* (No. 2020-01-1031). SAE Technical Paper.

[3] Shahri PK, Ghasemi, A H and Izadi V, 2020 *Optimal Lane Management in Heterogeneous Traffic Network Using Extremum Seeking Approach* (No. 2020-01-0086). SAE Technical Paper.

[4] Izadi V, Bhardwaj A and Ghasemi A H, 2020 *Impedance Modulation for Negotiating Control Authority in a Haptic Shared Control Paradigm*, arXiv preprint arXiv:2001.07779.

[5] Izadi V, Yeravdekar A and Ghasemi A, 2019. *Determination of roles and interaction modes in a haptic shared control framework*. In Dynamic Systems and Control Conference (Vol. 59148, p. V001T05A005). American Society of Mechanical Engineers.

[6] Ackermann J, Bunte T, and Odenthal D 2003 *Advantages of active steering for vehicle dynamics control*, in Proceedings of the 32nd International Symposium on Automotive Technology and Automation, Vienna, Austria, pp. 263-270

[7] Esmailzadeh E, Goodarzi A, Vossoughi G R 2010 *Optimal yaw moment control law for improved vehicle handling*, Mechatronics, Vol. 13, pp. 659-675

[8] Resta F, Teuschl G, Zanchetta M, and Zorzutti A 2005 *A new control strategy for a semi-active differential (Part II)*, 16th International Federation of Automatic Control (IFAC) World Congress, Prague

[9] Tomari T, Mori A, Shibahata Y 2006, *Development of SH-AWD Based on DC (Direct Yaw Control) Concept*, 8th International Symposium on Advanced Vehicle Control (AVEC), Taipei, Taiwan, August 20-24

[10] Rajmani R, Piyabongkarn D, Grogg J, Yuan Q, and Lew J 2006, *Dynamic modeling of torque-biasing devices for vehicle yaw control*, SAE Word Cingress, Detroit, MI, USA, SAE paper 2006-01-1963

[11] Lee H 2006 *Four-wheel drive control system using a clutchless centre limited slip differential", Proceedings of the Institution of Mechanical Engineers*, Part D: Journal of Automobile Engineering, Vol. 220, No. 6, pp. 665-682

[12] Canale M et al 2007 *Robust Vehicle Yaw Control using an Active Differential and IMC techniques*, Control Engineering Practice, no.15, pp. 923-941,

[13] Rubin D and Arogeti S 2013 *Vehicle Yaw Stability Control Using Rear Active Differential via Sliding Mode Control Methods*, In Proc. of 21st Mediterranean Conference on Control & Automation (MED), Platanias-Chania, Crete, Greece, pp. 317-322

[14] Lanzotti A, Renno F, Russo M, Russo R, Terzo M 2014 *Design and development of an automotive magnetorheological semi-active differential*, Mechatronics, No. 24, pp 426-435.

[15] Pacejka H B, 2002, Tyre and Vehicle Dynamics. Oxford, UK: Elsevier, Butterworth-Heinemann.

[16] Ahani H, Familian M and Ashtari R, 2020. *Optimum Design of a Dynamic Positioning Controller for an Offshore Vessel*, Journal of Soft Computing and Decision Support Systems, 7(1), pp.13-18.

[17] Izadi V, Shahri P K and Ahani H, 2020 *A compressed-sensing-based compressor for ECG* Biomedical engineering letters, pp.1-9.

[18] Surakanti S R, Khoshnevis S A, Ahani H and Izadi V, 2019 *Efficient Recovery of Structrual Health Monitoring Signal based on Kronecker Compressive Sensing*, International Journal of Applied Engineering Research, 14(23), pp.4256-4261.

ISSN: 2067-3809