

Control design of an electro-hydraulic actuator for variable-geometry suspension systems

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Abstract—This paper proposes the modeling and control design of an electro-hydraulic actuator for the variable-geometry suspension system. The purpose of this novel suspension type is to guarantee trajectory tracking through the variation of the camber angle. The motion of the wheel is realized by the electro-hydraulic actuator, whose elements are the cylinder and the electric valve. Thus, the trajectory tracking of the vehicle requires the positioning of the cylinder. First, the modeling of the system is presented using hydraulic and mechanical equations. Since the dynamics of the actuator contains several nonlinearities, a control-oriented linear model is formed. Second, the positioning problem of the cylinder is solved by using the Control Lyapunov Function technique, which is able to handle the discrete control input. Moreover, the control strategy requires pressure signals in the cylinder, which are estimated using a robust $\mathcal{H}_\infty$ observer. The efficiency of the proposed algorithm is demonstrated through simulation examples.

I. INTRODUCTION AND MOTIVATION

The variable-geometry suspension is a novel solution, by which the stability and the dynamics of the vehicle can be enhanced. This mechanism has several advantages such as small and light construction and low cost. Besides it has only few drawbacks. Generally, the interventions of the variable-geometry suspension are the variation of the camber angle and the modification of the position of the control arm. In the literature various applications of the variable-geometry suspension can be found. [1] proposes a rear-suspension active toe control to enhance driving stability. Another application of the variable-geometry suspension is the active tilt control, which supports the driver in balancing the vehicle, see [2]. An active camber suspension mechanism that can enhance the handling performances of the vehicle is presented by [3].

The paper is based on a novel conception of the variable-geometry suspension, which results in the variation of the steering angle, see Figure 1. In the conventional applications the purpose of the variable-geometry suspension is to modify the camber angle of the wheel, with which a lateral tire force between the tire and the road is generated [4], [5]. Since

the impact of the camber angle on the lateral wheel force is limited, the conventional system requires a large camber angle, which can be disadvantageous from the aspect of the tire. However, in this novel solution the camber angle γ also modifies the steering angle of the wheels, which may have significant effects to the lateral tire forces. If the camber angle is varied then the scrub radius r_δ is also influenced. The scrub radius is a virtual distance, which can be considered the radius of the wheel steering rotation. Since the front wheels are driven, the longitudinal forces on the wheels create a moment on the wheels and, thus, realize steering angles. Therefore, the appropriate coordination of wheel tilting and driving can result in the improved maneuvering of the vehicle.

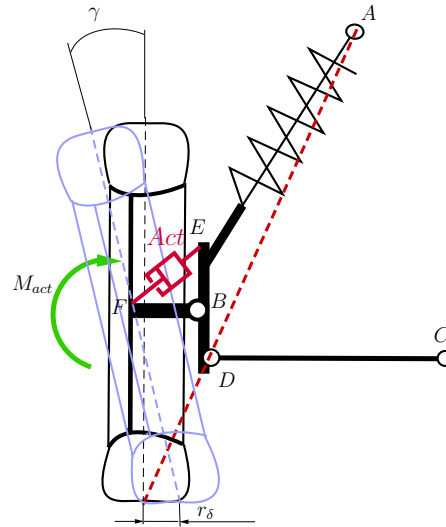


Fig. 1. Scheme of the suspension construction

Although there are some works on the control of the variable-geometry suspension and its integration with further vehicle dynamic control systems [6], [7], there are fewer results in the control of the suspension actuator. This paper contributes the control design of the electro-hydraulic actuator, using the Control Lyapunov Function technique and the robust $\mathcal{H}_\infty$ filtering. The proposed control algorithm provides solutions to several problems, such as the discrete control input signal and the estimation of the hydraulic pressure. In our earlier research a further method was presented, see [8]. However, this solution is efficient only for linear systems, whose control input is continuous. Moreover, [9] presents another Control Lyapunov Function based method, but it is

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The research was supported by the National Research, Development and Innovation Fund through the project "SEPPAC: Safety and Economic Platform for Partially Automated Commercial vehicles" (VKSZ 14-1-2015-0125). This paper was partially supported by the János Bolyai Research Scholarship of the Hungarian Academy of Sciences.

applied to active anti-roll bars, and the system has only two states. The efficiency of the control algorithm of the current paper is demonstrated through simulation scenarios using high-fidelity simulators.

The paper is organized as follows. Section II presents the modeling of the electro-hydraulic actuator for the variable-geometry suspension, containing the nonlinear description and the method of the control-oriented linearization. The control algorithm of the actuator is proposed in Section III. It incorporates the Control Lyapunov Function based method and the observer design. Section IV illustrates the efficiency of the control solution through simulation scenarios. Finally, Section V summarizes the contributions of the paper.

II. MODELING OF THE SUSPENSION ACTUATOR

In this section the modeling of the hydraulic actuator is presented. The purpose of the actuator is to realize the desired camber angle (γ). The control input of the system is the position of the valve $A(x_v)$ and the output of the system is the displacement of the piston. The selected actuator is linear and asymmetric, see Figure 2.

The actuation of the variable-geometry suspension is a crucial problem of vehicle control systems. Nowadays, there are two different constructions, such as pure electronic [10], [11] and electro-hydraulic [12], [13], [14] solutions. The advantage of the electronic solution is the simple construction, which requires only an electric power source. However, the electric motor is able to actuate only limited control torque, and it requires a large space in the vehicle. The other actuator type, the electro-hydraulic construction is capable of guaranteeing an increased control intervention, and the hydraulic cylinder can be built in small places at the suspension. Moreover, the static position of the wheel can be guaranteed without a continuous energy inflow into the system. As a disadvantage, it requires external hydraulic power sources and further elements to realize the pressure supply.

Since the actuator must realize a wide range of the camber angle, in this paper a double-acting hydraulic cylinder is used, in which the working fluid acts on both sides of the piston. The scheme is found in Figure 2.

The nonlinear model of the hydraulic actuator is derived from motion and hydrodynamic equations [15]. The hydrodynamic equations have two types, such as the equations of flows and pressures. The equations of flows (4), (6) depend on the pressure difference ΔP and the surface of the valve $A(x_v)$, while the equations of the pressures (3), (5) depend on the flows Q_i , the position (x_p) and the velocity ($\dot{x}_p$) of the piston. Since the selected actuator is a double-acting construction, it can be supplied by either side.

Thus, the nonlinear model is formulated for two cases, such as extending and retracting. In the case of extending:

$$P_i = P_s, \quad P_l = P_0, \quad (1)$$

while in the case of retracting:

$$P_i = P_0, \quad P_l = P_s, \quad (2)$$

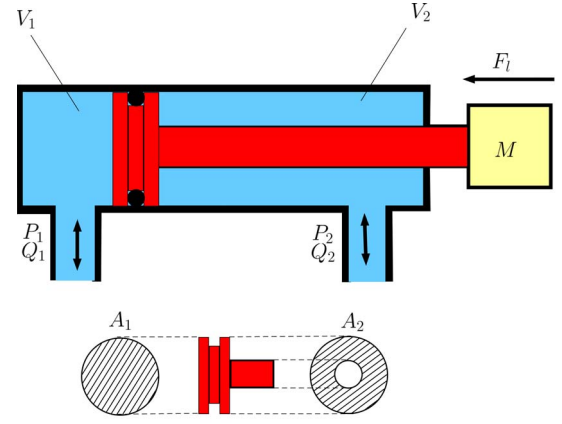


Fig. 2. Hydraulic actuator

where P_s is the supply pressure, P_0 is the environmental pressure.

The equations of the piston motion is formed as

$$\dot{P}_1 = \frac{B_f}{A_1 x_p + V_1} (-Q_1 - \alpha(P_1 - P_2) - A_1 \dot{x}_p) \quad (3)$$

$$Q_1 = C_f A(x_v) \sqrt{\frac{2|P_i - P_l|}{\rho}} \quad (4)$$

$$\dot{P}_2 = \frac{B_f}{A_2(l - x_p) + V_2} (Q_2 + \alpha(P_1 - P_2) + A_2 \dot{x}_p) \quad (5)$$

$$Q_2 = C_f A(x_v) \sqrt{\frac{2|P_2 - P_l|}{\rho}} \quad (6)$$

$$m\ddot{x}_p = F_l - b\dot{x}_p + A_1 P_1 - A_2 P_2 \quad (7)$$

where P_1 and P_2 are the pressures in the chambers, F_l is the external force on the cylinder, V_i is the initial volume, B_f is the Bulk modulus and α is the internal cross-port leakage coefficient. Moreover, l is the maximum displacement of the piston, C_f is the leakage coefficient and b is the viscous damping coefficient. The further parameters are the surface of the valve $A(x_v)$, the density of the fluid ρ , the mass of the piston m , and the position of the piston x_p .

The relationships of the extending and retracting motions contain several nonlinearities. Since the aim of system modeling is to design the control of the actuator, some assumptions are considered as follows.

- The equations of pressure dynamics (3), (5) are linearized around

$$x_0 = x_p|_{\gamma=0}, \quad (8)$$

where x_0 is the position of the piston at zero camber angle. Using this assumption the first terms of the equations are constant, such as

$$\eta_1 = \frac{B_f}{A_1 x_0 + V_1}, \quad (9)$$

$$\eta_2 = \frac{B_f}{A_2(l - x_0) + V_2}. \quad (10)$$

- The Taylor series expansion is applied to linearize the equations of flows, such as (4), (6). The equations are linearized around A_0 valve position and $P_{i,0}$ pressures. In this case the Taylor series is formed as

$$Q_i \approx K_{A,i}A_i + K_{P,i}P_i, \quad (11)$$

where the coefficients are

$$K_{A,i} = \left. \frac{\partial Q_i}{\partial A} \right|_{A_0, P_{i,0}} = C_f \sqrt{\frac{2\Delta P_{i,0}}{\rho}}, \quad (12)$$

$$K_{P,i} = \left. \frac{\partial Q_i}{\partial P_i} \right|_{A_0, P_{i,0}} = \frac{C_f A_0}{\sqrt{2\Delta P_{i,0}\rho}}. \quad (13)$$

$\Delta P_{i,0}$ denotes the pressure difference, which has two cases: In the case of extending:

$$\Delta P_{1,0} = |P_s - P_{1,0}|, \quad \Delta P_{2,0} = |P_{2,0} - P_0| \quad (14)$$

In the case of retracting:

$$\Delta P_{1,0} = |P_{1,0} - P_0|, \quad \Delta P_{2,0} = |P_s - P_{2,0}| \quad (15)$$

Although $\Delta P_{1,0}$ and $\Delta P_{2,0}$ have different values in the two cases, these values are close to each other. Thus, the $\Delta P_{i,0}$ can be handled as a constant. Moreover, the external force F_l is ignored during the linearization.

The assumptions result in the following control-oriented linear form:

$$\begin{aligned} \dot{P}_1 = & \eta_1 K_{P,1} P_1 - \eta_1 \alpha P_1 + \eta_1 \alpha P_2 - \eta_1 A_1 \dot{x}_p - \\ & - \eta_1 K_{A,1} A(x_v) \end{aligned} \quad (16)$$

$$\begin{aligned} \dot{P}_2 = & \eta_2 \alpha P_1 + \eta_2 K_{P,2} P_2 - \eta_2 \alpha P_2 - \eta_2 A_2 \dot{x}_p + \\ & + \eta_2 K_{A,2} A(x_v) \end{aligned} \quad (17)$$

$$\ddot{x}_p = \frac{A_1}{m} P_1 - \frac{A_2}{m} P_2 - \frac{b}{m} \dot{x}_p \quad (18)$$

The linearized equations can be transformed into a state-space representation. The state vector of the system is $x_{hyd} = [P_1 \ P_2 \ \dot{x}_p \ x_p]^T$ and the control input of the system is $u_{hyd} = A(x_v)$. The state equation is formed as

$$\dot{x}_{hyd} = A_{hyd} x_{hyd} + B_{hyd} u_{hyd} \quad (19)$$

where

$$A_{hyd} = \begin{bmatrix} \eta_1(K_{P,1} - \alpha) & \eta_1 \alpha & -\eta_1 A_1 & 0 \\ \eta_2 \alpha & \eta_2(K_{P,2} - \alpha) & -\eta_2 A_2 & 0 \\ \frac{A_1}{m} & -\frac{A_2}{m} & -\frac{b}{m} & 0 \\ 0 & 0 & 1 & 0 \end{bmatrix}$$

$$B_{hyd} = \begin{bmatrix} -\eta_1 K_{A,1} \\ \eta_2 K_{A,2} \\ 0 \\ 0 \end{bmatrix}$$

Note that the value of the control input is related to the position of the valve. It means that $u_{hyd} = 0$ represents the closed state of the valve, when the flow of the fluid is prohibited. When $u_{hyd} \neq 0$ the valve is open.

The linear formula yielded represents the dynamics of the electro-hydraulic actuator, incorporating the position of the valve and the dynamics of the cylinder. In the following the described state-space representation is used in the design of the actuator control.

III. CONTROL DESIGN OF THE ELECTRO-HYDRAULIC ACTUATOR

In this section the control design of the hydraulic actuator is presented. Although the dynamics of the system is described through the state-space representation, there are some further challenges in the control design.

- The dynamics of the pressure is significantly faster than the dynamics of the piston motion. Therefore, the poles of A_{hyd} vary in a high range, which can lead to numerical problems in the control design.
- The control input u_{hyd} has discrete values $[A(x_{v,min}); 0; A(x_{v,max})]$, which results from the construction of the electro-hydraulic valve. Therefore, the controller must guarantee the stability and the performances of the system considering this constraint.
- In practice, the measurement of the pressure in the chambers is difficult, or it can be expensive in the industrial application.

In the rest of this section the control strategy based on the Control Lyapunov Function and the estimation of the hydraulic pressures P_1, P_2 are proposed.

A. Control strategy

The goal of the control design is to guarantee the required camber angle through the positioning of the cylinder. Since there is a direct mechanical contact between the cylinder and the wheel, the reference signal of the cylinder is computed using the geometry of the suspension. If the distances between point D and each connection point of the hydraulic actuator (DE and DF) have the same value L , then the reference position is formed as

$$x_{ref} = \frac{L \sin\left(\frac{\pi}{2} - \gamma_{ref}\right)}{\sin\left(\frac{\frac{\pi}{2} + \gamma_{ref}}{2}\right)} \quad (20)$$

where γ_{ref} is the reference camber angle, which results from the vehicle dynamical control, see [7]. Thus, the control problem is to guarantee the minimization of the performance

$$z = x_{ref} - x_p \quad |z| \rightarrow \min. \quad (21)$$

The control design of the actuator is based on the Control Lyapunov Function. The state-space representation of the system is given in the following form:

$$\dot{x} = f(x) + gu \quad (22)$$

where $f(x)$ is a vector, which incorporates smooth polynomial functions and $f(0) = 0$.

The global asymptotical stability of the system at the origin is guaranteed by the existence of the Control Lyapunov Function of the system defined as follows [16]:

A smooth, proper and positive-definite function $V : \mathbb{R}^n \rightarrow \mathbb{R}$ is a Control Lyapunov Function for system (22) if

$$\inf_{u \in \mathbb{R}} \left\{ \frac{\partial V}{\partial x} f(x) + \frac{\partial V}{\partial x} g \cdot u \right\} < 0 \quad (23)$$

for each $x \neq 0$.

If there exists a Control Lyapunov Function for the system (22), then the system can be stabilized [16]. In the followings the tracking problem (21) is handled as a stabilization of the system around the reference signal [9]. Thus, new states are defined, considering the reference signal (20). The tracking error is formed as

$$e = x_{ref} - x_p. \quad (24)$$

Since the variation of the reference signal x_{ref} is slower than the dynamics of x_p , the derivative of the error is approximated in the form

$$\dot{e} = -\dot{x}_p. \quad (25)$$

Using the defined tracking error (24), its derivative (25) and the elements of x_{hyd} , the next new state ξ is defined

$$\begin{aligned} \xi &= [P_1 \quad P_2 \quad \alpha e \quad \dot{e}] \\ &= [P_1 \quad P_2 \quad \alpha(x_{ref} - x_p) \quad -\dot{x}_p]^T \end{aligned} \quad (26)$$

and the derivative

$$\begin{aligned} \dot{\xi} &= [\dot{P}_1 \quad \dot{P}_2 \quad \alpha \dot{e} \quad \ddot{e}] \\ &= [\dot{P}_1 \quad \dot{P}_2 \quad -\alpha \dot{x}_p \quad -\ddot{x}_p]^T \end{aligned} \quad (27)$$

where α is a tuning parameter, concerning to the speed of the system. The candidate Control Lyapunov Function is defined in the form

$$V = \frac{1}{2} \xi^T \xi. \quad (28)$$

The derivation of (28) results in the function

$$\begin{aligned} \dot{V} &= \xi^T \dot{\xi} \\ &= [P_1 \quad P_2 \quad \alpha(x_{ref} - x_p) \quad -\dot{x}_p] [\dot{P}_1 \quad \dot{P}_2 \quad -\alpha \dot{x}_p \quad -\ddot{x}_p]^T \end{aligned} \quad (29)$$

It can be seen that (29) contains the pressure dynamics and the motion dynamics of the cylinder simultaneously. However, the pressure dynamics is significantly faster than the motion dynamics, which can lead to numerical problems in the control strategy. It means that the impact of pressure dynamics on the stability is less significant than the motion. Therefore, the pressure is eliminated from V and $\dot{V}$.

In this case, the following scenarios are considered

- if $u_{hyd} = A(x_{v,max})$ then $P_1 = P_0$ and $P_2 = P_s$,
- if $u_{hyd} = A(x_{v,min})$ then $P_1 = P_s$ and $P_2 = P_0$,
- if $u_{hyd} = 0$ then P_1 and P_2 are not influenced by the control.

The approximation of (29) is formed as

$$\dot{V} \approx (-\ddot{x}_p - \alpha \dot{x})(-\dot{x}_p + \alpha(x_{ref} - x_p)) \quad (30)$$

Using (7) in (30), the stability criterion of the system is formulated through (23) such as

$$K_1 \dot{x}_p + K_2 \dot{x}_p^2 + K_3 \alpha(x_{ref} - x_p) + K_4 \alpha(x_{ref} - x_p) \dot{x}_p < 0, \quad (31)$$

where the coefficients are

$$K_1 = \frac{P_1 A_1}{m} - \frac{P_2 A_2}{m}, \quad (32)$$

$$K_2 = -\frac{1}{m} + \alpha \quad (33)$$

$$K_3 = -\frac{P_1 A_1}{m} + \frac{P_2 A_1}{m}, \quad (34)$$

$$K_4 = \frac{1}{m} - \alpha. \quad (35)$$

In the control algorithm the reference signal is tracked if (31) is guaranteed. It means that it is necessary to select the pressures P_1 and P_2 according to (31).

It results in the following control algorithm.

- 1) Since P_1 and P_2 are connected to each other through u_{hyd} , the stability of the system is checked at all of the valve positions $[A(x_{v,min}); 0; A(x_{v,max})]$. It requires the computation of (31).
- 2) Then, it is necessary to select the value $A(x_v)$, where (31) has the minimum value.
- 3) Furthermore, if the relative error $\frac{x_{ref} - x_p}{x_{ref}} \cdot 100 < 1\%$ and the condition of (31) is guaranteed for the closed position of the valve, then the valve is closed. In this way the energy consumption of the control system is reduced.

B. Observer design for pressure estimation

The presented control algorithm guarantees the precise tracking of the reference signal x_{ref} . The computation of (31) requires information about the states x_p , $\dot{x}_p$, which are considered to be available. The values P_1 and P_2 are substituted based on the proposed rule. At the stability checking of the closed position the current P_1 and P_2 values must be considered. However, the measurement of P_1 and P_2 requires expensive tools, which means that the estimation of the signals is required.

The estimation of the pressures is based on the state-space representation (19). In the following the robust $\mathcal{H}_\infty$ method is used to estimate the states P_1 , P_2 from the measured signals x_p , $\dot{x}_p$. The state-space representation of the system together with the performance and the measurement equations is written as

$$\dot{x}_{hyd} = A_{hyd} x_{hyd} + B_{hyd} u_{hyd} \quad (36)$$

$$z = C_1 x_{hyd} \quad (37)$$

$$y = C_2 x_{hyd} + D_{21} d \quad (38)$$

where d is the disturbance on the measurement, e.g. sensor noise.

If the output estimation plant is [17]

$$P_{OE} = \begin{bmatrix} A_{hyd} & 0 & B_{hyd} \\ C_1 & 0 & 0 \\ C_2 & D_{21} & 0 \end{bmatrix} \quad (39)$$

where $A_{hyd} - B_{hyd}C_1$ is stable, then $\|\mathcal{F}_l(P_{OE}, K_{OE})\|_\infty < \Gamma$ if

$$K_{OE} = \begin{bmatrix} A_{hyd} + LC_1 - B_{hyd}C_1 & L \\ C_1 & 0 \end{bmatrix} \quad (40)$$

where the observer $L = -YC_2^T$ and Y satisfies the algebraic Riccati equation

$$A_{hyd}Y + Y A_{hyd}^T - Y \left(C_2^T C_2 - \frac{C_1^T C_1}{\Gamma} \right) Y = 0. \quad (41)$$

Note that the stability of the system is guaranteed since the original system is stable. Figure 3 shows an example for the accuracy of the pressure estimation. Figure 3(a) illustrates 1 second of a process, and in Figure 3(b) the signals are zoomed for a short time range. The mean of the estimation error is 0.77 MPa and its variance is 0.17, see Figure 3(c). It can be seen that the designed observer guarantees an appropriate estimation of the pressure signal. The estimated pressure signal is applied in the algorithm of the hydraulic control.

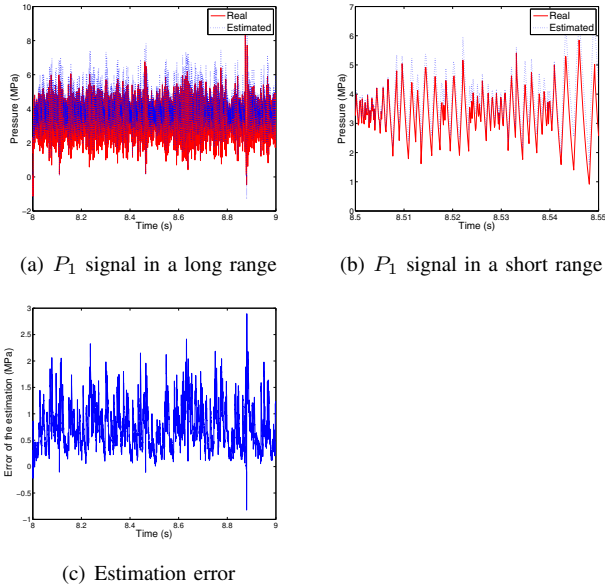


Fig. 3. Estimation of the pressure signal

IV. SIMULATION RESULTS

In this section the efficiency of the proposed control algorithm is demonstrated through high-fidelity softwares, such as CarSim, Matlab/Simulink and Matlab/SimMechanics. The

control algorithm is designed in Matlab/Simulink and SimMechanics, and in the implementation step the controller is connected to the vehicle in the CarSim software.

The simulation example demonstrates a cruising on a short section of the Waterford Hill race course, which is located in Michigan, shown 4(a). During the demonstration the vehicle

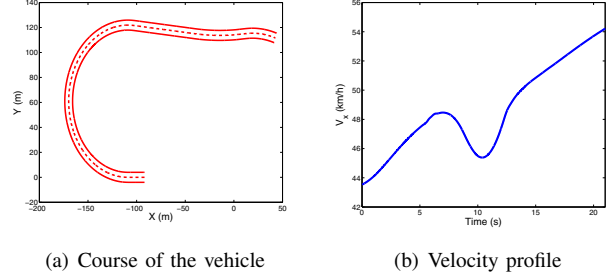


Fig. 4. Track of the vehicle

must perform a cornering maneuver, in which the steering of the front wheels is realized by the variable-geometry suspension system. Thus, in the example the electro-hydraulic actuator must be controlled through the valve to guarantee the cornering of the vehicle. As Figure 4(b) shows the velocity of the vehicle varies during the simulation.

The yaw-rate tracking of the vehicle is illustrated in Figure 5(a). It can be seen that the difference between the reference and the real signal is small, which means that the proposed control system is able to guarantee the trajectory tracking of the vehicle. It means that the designed electro-hydraulic actuator controller is able to guarantee the precise positioning of the wheel. The steering angles of the wheels are shown in Figure 5(b). Since in the lateral control design the steering intervention of the left and right wheels are the same, in the following the simulation results are demonstrated only on the actuator of the left wheel.

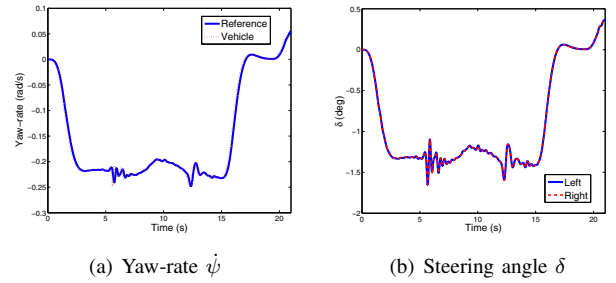


Fig. 5. Simulation results on the vehicle level

The pressures of the actuator in the left and right chambers of the left wheel actuator are illustrated in Figures 6(a),(b). As it has been mentioned in Section III the pressures vary rapidly during the simulation due to their fast dynamics. It can be seen that the pressure in the chambers often reaches the supply pressures $P_s = 12MPa$. The tracking of the desired cylinder position is illustrated in Figures 6(c)(d). Although the wheel has disturbances from the road, the

presented control system is able to guarantee the required position. The maximum of the displacement is below 0.2mm , which is an acceptable value for the electro-hydraulic actuator. Finally, Figure 6(e) shows a section of the control input $A(x)$. It can be seen that all of the positions (closed, left, right) appear, which guarantee appropriate performances and stability.

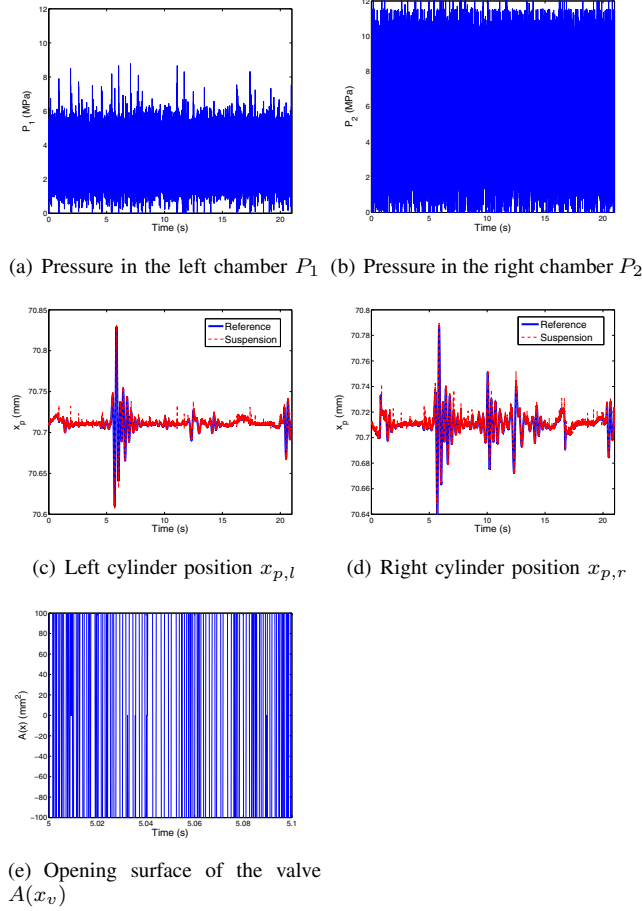


Fig. 6. Simulation results on the actuator level

V. CONCLUSIONS

The paper has proposed the modeling and control design of an electro-hydraulic actuator of the variable-geometry suspension system. The purposes of the suspension system are the improvement of lateral dynamics, which is achieved by the appropriate modification of the camber angle through the hydraulic cylinder positioning. A control-oriented model of the actuator has been formed based on the hydraulic and mechanical equations. The positioning problem has been solved by applying a Control Lyapunov Function-based algorithm, in which the discrete states of the control valve can be considered. Moreover, the control method contains a robust $\mathcal{H}_\infty$ observer to estimate the pressure signals in the cylinder.

The efficiency of the modeling and the control has been demonstrated through simulation examples using high-

fidelity simulation tools. The simulation results illustrate that the Control Lyapunov Function based control system is able to guarantee the trajectory tracking of the vehicle. The variable-geometry suspension with an electro-hydraulic actuator yields appropriate performances.

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