

Robust Control of an Electronic Throttle System Via Switched Chattering Control: Benchmark Experiments

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Abstract: A robust controller with chattering is proposed based on a simplified model of an electronic throttle system. The chattering term provides robustness against un-modeled nonlinearities, e.g., limp-home nonlinearity, parameter dispersion, and friction phenomena. As the simplified model of the throttle system can be seen to correspond to a horizontal one-degree-of-freedom robot manipulator, the proposed controller is based on a previously designed robust control for regulation of robot manipulators with friction. Moreover, this controller uses only position measurements as the throttle benchmark unit does not have velocity measurements. The conceived controller is then tested in the Benchmark throttle numerical platform offered by the host conference and their affiliates.

Keywords: Robust Control, Chattering Control, Model Reduction, Throttle System, Benchmark.

1. IN TRODUCTION

Electronic throttle control (ETC) is an automobile technology which severs the mechanical link between the accelerator pedal and the throttle. The electronic control unit (ECU) determines the required throttle position in order to satisfy the driver torque demand depending on accelerator pedal position, engine speed, vehicle speed, etc. The electric motor within the ETC is then driven to the required position via a closed-loop control algorithm within the ECU. As the engine management system of modern vehicles relies heavily on the performance of this servomechanism, the underlying control system must be efficient, robust and easily tunable.

ETC is an important topic for industrial automotive because of its importance in regulating air flows. High comfort and performance in automotive applications is based on beneficial performance of this control unit (see, for instance, [Nakano et al., 2006] and [Choi et al. 1996]). The ETC is an electromechanical system that controls the throttle valve in response to the gas pedal improving vehicle drivability, fuel consumption, and pollutant emissions [Pavkovic et al., 2006]. Fig. 1 shows a diagram of a throttle system. This system has a DC motor that manipulates the throttle valve, and a sensor to measure the throttle position. The return-spring brings back the throttle to the so-called limp-home (LH) position in the case of power supply failure [Pavkovic et al., 2006]. It has been established in [Deur et al., 2004] that the friction effect in the DC motor, and the nonlinearity of the return spring, significantly affect the performance of the electronic throttle device.

A reduced throttle system model is analyzed in [Nakano et al., 2006] by ignoring the inductance of the DC motor. This model captures the friction effect of the motor by employing the Coulomb friction framework. However, this model has four unknown parameters. Here, a simplified model is deri-

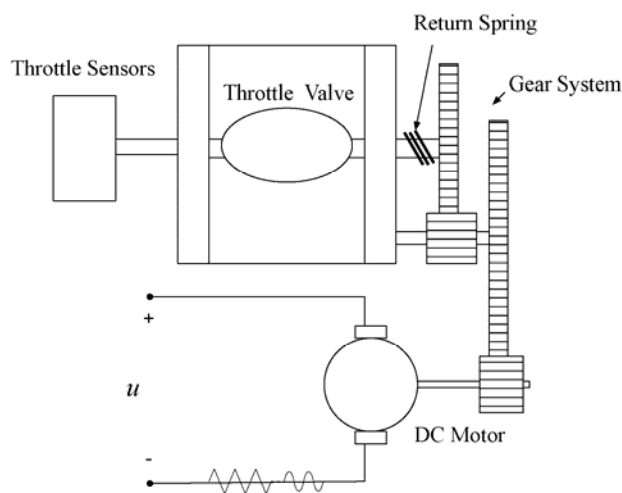


Fig. 1. Throttle electromechanical system.

ved from the one proposed by [Nakano et al., 2006] to obtain a more compact controller that only depends on a single parameter. The reduced system has been identified as a dynamic model corresponding to a horizontal one-degree-of-freedom robot manipulator, thus, the chattering controller reported in [Orlov et al., 2003] can be applied straightforwardly. This controller is robust against friction phenomena and it uses only position measurements. Numerical experiments applied to the throttle Benchmark show acceptable performance of our throttle controller. It is noteworthy that the Benchmark system captures nonlinearities such as limp-home, and parameter dispersion, within others.

This paper is structured as follows. Section 2 deals with the throttle modeling and the control design. The numerical Benchmark experiments are shown in Section 3. Finally, in Section 4, the conclusions are stated.

2. MODELING AND CONTROL DESIGN

Consider the dynamic model of a throttle system [Nakano et al., 2006],

$$\dot{x}_1 = x_2, \quad (1)$$

$$\dot{x}_2 = a_1 x_1 + a_2 \text{sgn}(x_2) + a_3 x_2 + a_4 u, \quad (2)$$

where x_1, x_2 , are the state variables representing position and velocity of the throttle valve, respectively. The system parameters $a_i, i = 1,2,3,4$, depend on the gear ratio, motor inertia, frictional coefficient, spring constant, etc. These parameters are assumed unknown for control design, as their values can change with aging and they are also environmental dependent. Therefore, friction cannot be compensated by supposing a_2 known. The term $a_2 \text{sgn}(x_2)$ in (1)-(2) captures the Coulomb friction force, the term $a_3 x_2$ captures, among others, the viscous force, and the term $a_1 x_1$ represents the spring force. In order to give the reader an idea of the values of these parameters, Table 1 gives their estimated values for the specific throttle system presented in [Nakano et al., 2006].

Table 1: System Parameters.

Parameter	Value
a_1	-10.24
a_2	-0.295
a_3	-75.65
a_4	537.8

It is important to note that the values of the parameters for the particular ETC model of the benchmark are not needed as it is seen next. As detailed in Appendix I, the viscous and the spring force terms can be omitted to obtain a simplified model system that is only used for control design. A control law with many terms may lead to saturated controller values, as contribution of all terms may add up. Thus, by reducing the number of terms in the model system, the designed control law also has fewer terms and it can avoid saturation. Thus, for control design the following simplified model is obtained

$$\dot{x}_1 = x_2, \quad (3)$$

$$\dot{x}_2 = a_2 \text{sgn}(x_2) + a_4 u. \quad (4)$$

Equation (4) can be scaled by a factor of $1/a_4$ and defining $x := x_1/a_4$ system (3)-(4) has the following second-order-differential-equation representation:

$$\ddot{x} + \rho \text{sgn}(\dot{x}) = u, \quad (5)$$

where $\rho = -a_2/a_4$, which is assumed unknown, and x and \dot{x} are the position and velocity of the system, respectively. It is noteworthy that this second-order system corresponds to a horizontal one-degree-of-freedom robot manipulator. Thus, all the control theory developed in this field can be applied [Kelly et al., 2005]. Here, the controller proposed in [Orlov et al., 2003] is used as it is robust against friction phenomena and un-modeled dynamics. Next, the main result in [Orlov et al., 2003] is summarized (case of horizontal one-degree-of-freedom robot manipulator).

Theorem 1 ([Orlov et al., 2003]): *The following control law globally asymptotically stabilizes the system (5) around the desired constant position x_d ,*

$$u = -k_d \dot{\hat{x}} - k_p e - k_\alpha \text{sgn}(e), \quad (6)$$

$$\dot{\hat{x}} = -L \hat{x} + k_d e, \quad (7)$$

where $e := x - x_d$, and L, k_d, k_p , and k_α are given positive constants, and $k_\alpha > \rho$. Moreover, the equilibrium point $(\dot{x}, \hat{x}, e) = 0$, is globally asymptotically stable.

Remark 1. The control parameter k_α is tuned on-line using a trial-and-error technique. That is, as it must satisfy $k_\alpha > \rho$ and ρ is unknown ($\rho = -a_2/a_4$), the value of k_α is tuned until a good performance is obtained.

3. THROTTLE BENCHMARK EXPERIMENTS

The throttle Benchmark platform [Zito et al., 2009] is a Simulink model together with a series of test cases and control specifications. Participants in the Benchmark must design the control system. The Benchmark emulates an electronic throttle unit employed in diesel and gasoline engines. Nonlinearities, such as friction, limp-home and parameter dispersion, are captured in this Benchmark virtual experiment. According with this Benchmark platform, the test input to the model is a saturated PWM signal, and its output is the throttle position expressed in % (indicated as %pos in the following, with 100%pos corresponding to fully open throttle and 0%pos to fully closed throttle).

In this section, the controller stated in Theorem 1 is tested when $k_p = 25, k_\alpha = 5, k_d = 2$, and $L = 100$ (see Fig. 2).

The Benchmark model is accompanied by three different sets of reference signals:

- A1. Steps and combinations of steps.
- A2. Ramps
- A3. More complex signals (cycles).

The designed controller should satisfy a set of specifications with the given throttle model for the various sets of reference signals. These specifications for control design are given as follows:

- with signals A1
 - static error should not be greater than $0.125\%_{\text{pos}}$ (quantization error);
 - settling time (at $\pm 5\%$) should not be greater than 200ms for amplitudes over $50\%_{\text{pos}}$ and than 100ms for smaller amplitudes;
 - maximum overshoot should not be greater than $0.5\%_{\text{pos}}$ for amplitudes over $50\%_{\text{pos}}$ and than $0.125\%_{\text{pos}}$ for smaller amplitudes;
- with signals A2, tracking error should not be greater than 2.5% of ramp slope (expressed in $\%_{\text{pos}}/\text{s}$);
- integral square error (ISE) must be minimized for all signals (A1, A2, A3).

Firstly, the controller is tested with reference signals of type A1. Performance evaluation under step reference signals is shown in Figures 3 to 7. The dotted lines are the desired

position, and solid lines are the experimental results. From Fig. 4, we can appreciate a static error slightly above of 0.125%Pos and a settling time around of 200ms. The overshoot is below of 0.5%Pos. So, the specifications demanded by the throttle problem are almost completely satisfied. However, for the stairs reference signal, as can be seen in Fig. 6, the overshoot is sometimes above of 0.5%Pos. We believe this is because of the torsion effect of the throttle base. That is, as the throttle unit has a gear system and possibly a belt effect in the valve base, torsion effect (or joint elasticity) can be observed (Spong and Vidyasagar, 1989). Dynamic modeling of mechanical systems with elastic joint increases the

number of state variables (Spong and Vidyasagar, 1989). Indeed, according with this control theory, to obtain a robust control able to cope with the torsion effect, additional information is required: the shaft position measurement of the throttle motor. The Benchmark platform does not supply this information, so it hinders the design of a controller with torsion effect compensation. Finally, it is noteworthy from Fig. 7 that the saturation applied to the controller affects its performance. We have seen that when a larger range of saturation is allowed the controller performs better. These results are not included as the Benchmark does not allow changing this saturation range.

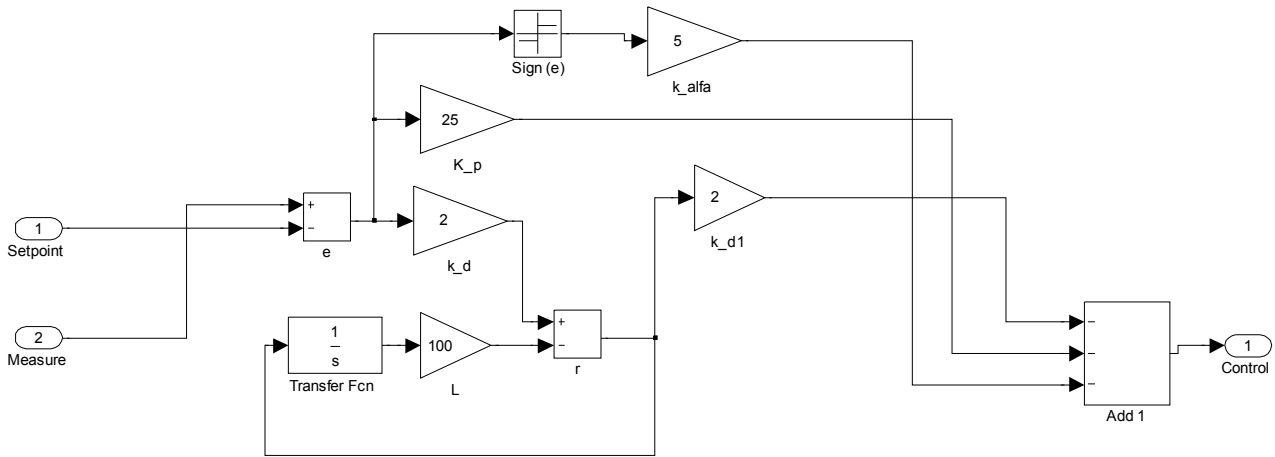


Fig. 2. Simulink realization of the control law.

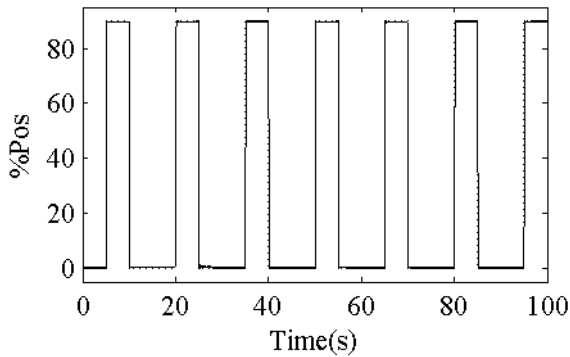


Fig. 3. Results for a step signal.

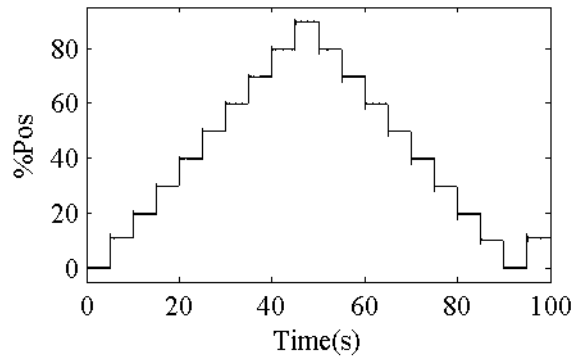


Fig. 5. Results for a step signal (stairs signal).

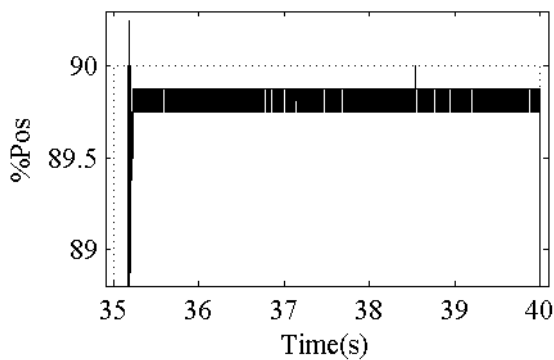


Fig. 4. Zoom to the picture in Fig. 3 when the biggest error is obtained.

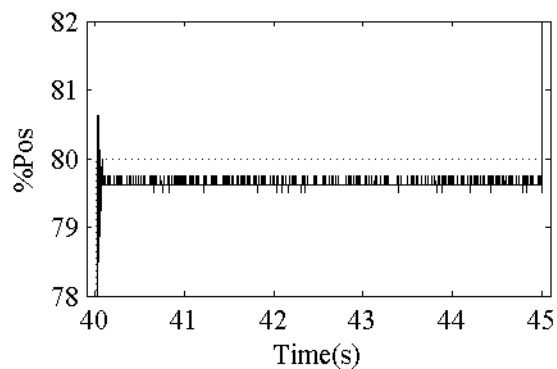


Fig. 6. Zoom to the picture in Fig. 5 when the biggest error is obtained.

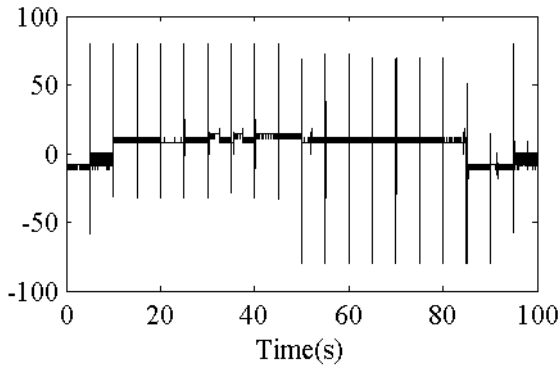


Fig. 7. Control performance for the step signal in Fig. 5.

Secondly, the controller is tested with reference signals of type A2. Performance evaluation, under a ramp reference signal, is shown in Figures 8 to 11. Fig. 8 shows that the reference trajectory and the measured one are similar. In Fig. 10 the tracking error between both trajectories is shown. The ramp slope of the triangular reference signal is 10 (%Pos/sec), and the maximum absolute error is approximately 0.8 %Pos (or 8% of the ramp slope). This value is above the specified one of 2.5%.

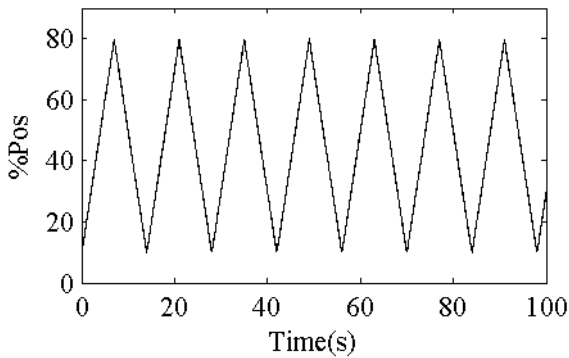


Fig. 8. Results for a ramp reference signal.

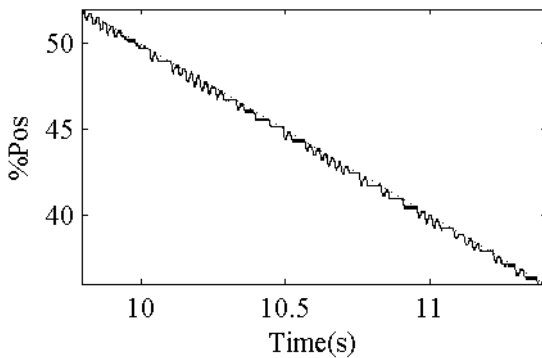


Fig. 9. A zoom to the picture in Fig. 8.

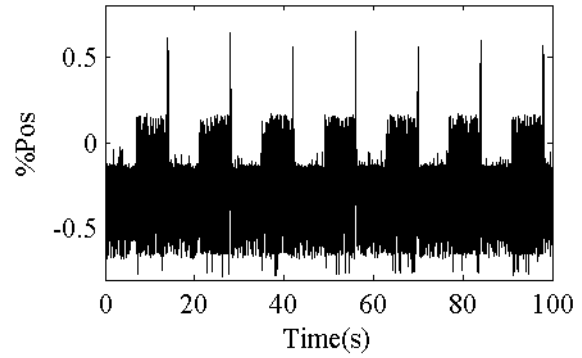


Fig. 10. Tracking error for the ramp reference signal.

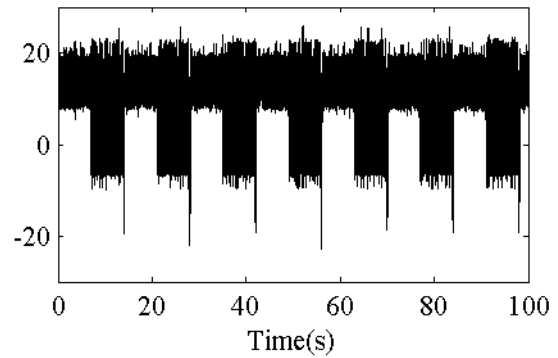


Fig. 11. Control performance for the ramp reference signal.

Thirdly, the controller is tested with reference signals of type A3. Performance evaluation is shown in Fig. 8. The integral square error for this signal is shown in Fig. 9

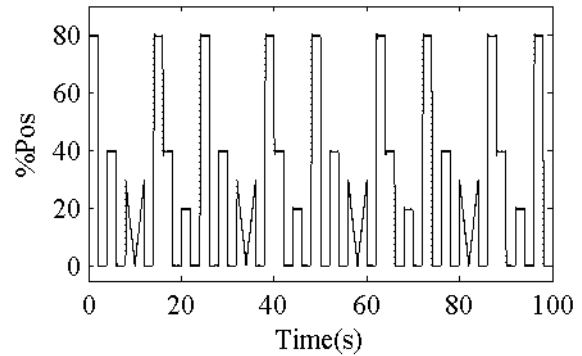


Fig. 12. Results for a more complex signal.

Finally, it is noteworthy that the presented simulations demonstrate the robustness of the control since, to the author's knowledge, the Benchmark captures external perturbations as, for example, disturbances on the air inlet valve (as can be seen in the open-loop testing). It would also be interesting to test the robustness with a modified Limp Home position, but this is not easily modifiable in the Benchmark.

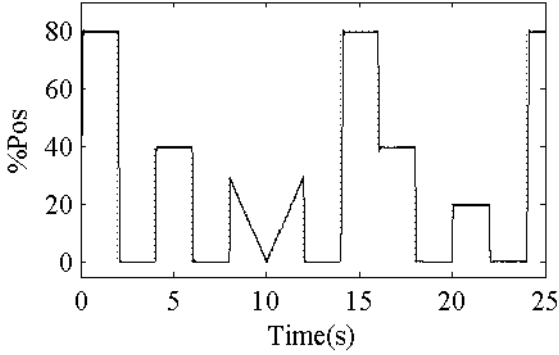


Fig. 13. A zoom to the picture in Fig. 12.

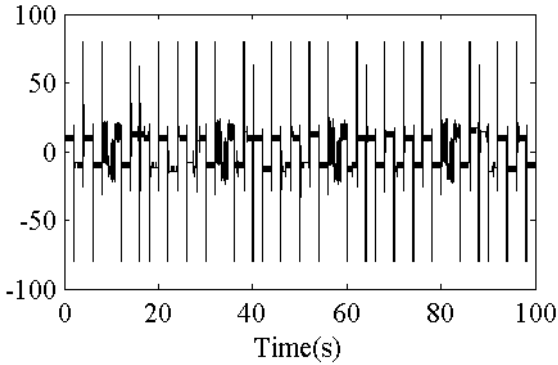


Fig. 14. Control performance for the signal in Fig. 12.

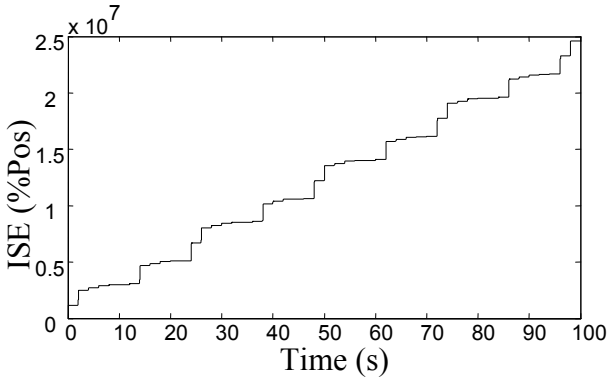


Fig. 15. Integral square error.

4. CONCLUSIONS

It has been proposed a throttle control based on a simplified throttle model and using a well known result in robotic theory. According to Benchmark experiments, this controller is robust against un-modeled nonlinearities and friction phenomena. The granted controller almost fulfils the specifications demanded by the throttle control Benchmark. However, in the throttle Benchmark there is a torsion delay (basically due to the joint flexibility of the throttle base) that has not been considered and which is affecting the controller performance. To obtain a robust control able to cope with the torsion effect, additional information is required: the shaft position measurement of the throttle motor. The Benchmark platform does not supply this information, making difficult to

design a controller with torsion effect compensation. In a nutshell, although without the shaft position information it is possible to avoid overshoot with some controllers, if the shaft position was known our controller could be improved and maybe it would met some of the specifications that, in its present form, is not fulfilling. Finally, it is also noteworthy that the saturation effect limits the performance of the proposed controller. If it was possible to change the saturation to a wider range the controller would, obviously, perform better.

APPENDIX I

Firstly, let's rewrite the system (1)-(2) in matrix form as

$$\dot{X} = AX + U + f(x), \quad (8)$$

where $X^T = (x_1, x_2)$, $A = \begin{pmatrix} 0 & 0 \\ a_1 & a_3 \end{pmatrix}$, $U^T = (0, a_4 u)$, and $f(x)^T = (x_2, a_2 \text{sgn}(x_2))$. Note that a_1 is the coefficient of the spring force and that a_3 is the coefficient of the viscous force and thus they are both negative.

Secondly, let's consider the auxiliary system

$$\dot{X} = AX. \quad (9)$$

The first equation of this system states that $\dot{x}_1 = 0$, and then $x_1(t) = x_1(0) = \text{const}$. Thus, the second equation of system (9) reads,

$$\dot{x}_2 = a_1 x_1(0) + a_3 x_2$$

and using the change of variables $\bar{x}_2 = x_2 + a_1 x_1(0)/a_3$, it can be rewritten as

$$\dot{\bar{x}}_2 = a_3 \bar{x}_2,$$

which is asymptotically stable. Therefore, the system (9) is stable in the sense of Lyapunov (this result can be also concluded by applying theorem 4.5 in [Khalil, 1996]) and it is BIBO-stable as for every bounded initial condition the trajectories of the system remain bounded. Then, by the converse Lyapunov theorem 4.7 in [Slotine et al., 1991] there exists an autonomous Lyapunov function, V , such that its time derivative along any state trajectory of system (9) is negative semi-definite, i.e.,

$$\dot{V} = \frac{\partial V}{\partial X} AX \leq 0.$$

Finally, let's consider the time derivative of V along any state trajectory of the initial system (8), then

$$\dot{V} = \frac{\partial V}{\partial X} (AX + U + f(x)) \leq \frac{\partial V}{\partial X} (U + f(x)).$$

Hence, for stability analysis (and for control design) the term AX can be omitted.

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