Analysis of the Gains Tuning Problem in a Backstepping Controller Applied to an Electrohydraulic Drive

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Abstract: This paper highlights the problem of tuning the gains of a non-adaptive backstepping controller in an electrohydraulic servo system. While the other non-adaptive controllers in the literature have precise gains tuning methods, the non-self-tuning backstepping controller has no rigorous gain tuning method. The proposed study aims to analyze the contribution of each backstepping controller gain in the closed-loop performance. Our final goal is to establish a rigorous gains-tuning method for the non-adaptive backstepping controller. The study starts with the development of three-stage gains backstepping controller using a non-conventional time derivative Lyapunov function. This particular Lyapunov function makes it possible to analyze the response of the system when all the controller gains are cancelled. Then, we analyze the effect of each gain by cancelling out the values of the others. The first simulation results show that the convergence of the tracking error to zero is not maintained when all gains are set to 0 despite the presence of a negative definite of the Lyapunov function time derivative. In this case, the equilibrium point is not the expected one as time goes to infinity. The second set of results indicates that adjusting the gain related to the feedback of the actual output only ensures the asymptotic convergence of the tracking error to zero as time goes to infinity. However, developing a heuristic tuning of the three controller gains like Ziegler Nichols tuning remains a challenge.

Key-Words: - tuning gains, backstepping controller, electrohydraulic servo system, heuristic method, chattering effect

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1 Introduction

Electro-Hydraulic Servo Systems (EHSS) are used to handle large mechanical loads with a fast, accurate, and robust response. In these systems, pressurized hydraulic oil is used to transmit power. Some industrial EHSS applications include aerospace actuation, [1], automobile actuation, [2], and machine tools, [3]. Most of these electrohydraulic actuators are implanted using PID control laws because of their well-known and flexible methodology. PID controller consists of tuning three gains. The literature identifies rigorous approaches for non-self-tuning controllers like Ziegler Nichols, [4]. Good results but only in a limited operating point range are achieved using the PID control strategy, [4]. The EHSS dynamics have strong nonlinearities, [5], making the linear control theory inadequate to guarantee satisfactory performances over a large panel of operating points. PID control may be combined with some methods like fuzzy logic, [6], sliding mode, [7], fractional order strategy, [8], and optimization tools, [9], to improve the performance. However, when linear control is used on nonlinear systems, it is difficult to ensure both improved performances and expanded working conditions, [10].

Of all the control laws encountered in the literature, the backstepping control has advantages, especially when faced with system nonlinearities. Unlike feedback linearization control, this approach allows choosing which nonlinearity to cancel, [11], thus improving the robustness of the closed-loop system. Moreover, the recursive construction of the Lyapunov function allows the flexibility of the architecture of the control law, [12]. Backstepping control consists of dismantling the system into first-order subsystems where a state variable is considered as a control signal, [13]. These virtual controls or desired system variable states, [14], are chosen to ensure the negative definition of the Lyapunov function time derivative. Experimental and numerical results show that the backstepping controller is more efficient than the PID controller, [15].

There are two drawbacks to using the backstepping approach. The first one is the explosion of complexity due to repeated calculations when the plant model has a high order, [15]. The second drawback, and the one discussed in this paper, is the lack of a rigorous gains-tuning approach. At each recursive step in the design of the backstepping controller, a gain to be adjusted appears. In this paper, we focus on non-self-adjusting backstepping controller gain strategies. In [16], the authors show that there is a trade-off between the chattering effect and the convergence of the tracking error while adjusting the gains of the backstepping controller. An optimal gain is difficult to find in the absence of a rigorous tuning method. The literature identifies rigorous methods for tuning the parameters of non-self-tuning controllers like Ziegler Nichols, [17], for PID controllers and pole placement for feedback linearizing controllers, [18]. However, to our knowledge, authors in the literature adjust the gains of the non-self-tuning backstepping controller via trial and error. Few authors try to analyze the effect of the gain in the closed loop performance. Authors, [19], show that the backstepping controller gains affect the robustness against parametric uncertainties. For each gain, they found a minimum value, an optimum value and a maximum value to guarantee convergence of the error. However, the contribution of each gain is not highlighted. In [20], authors show that the three gains of the backstepping controller affect the performance of the electrohydraulic brake system by varying one gain and setting the others to zero. They found that two of the three gains affect overshoot and steady state. However, their backstepping control law contains two input variables that operate alternately. The three gains appear in these input variables. The complex conditions of these inputs weaken the actual influence of the three gains.

The main contributions of this article are listed below:

- an unconventional Lyapunov function that makes possible the analysis of the performance while the controller gains are set to 0;
- an actual analysis of each gain contribution in the closed loop performance using a simple non-self-tuning backstepping control law with one input variable;
- a discussion of the perspective of tuning methods.

2 System Modelling

Fig. 1 shows the electrohydraulic servo system considered in this study. It is the same system presented in our previous work, [21]. It consists of a hydraulic motor that drives a rotating load. The hydraulic unit includes the pump, tank, pressure relief valve, and accumulator. It provides hydraulic oil flow at constant pressure. The electrohydraulic servo-valve is the interface between the operative part and the control part. The electric signal u(t) acts on the servo-valve spool by varying the oil flow into the hydraulic motor. Because a mechanical load is attached to the motor, a pressure difference $P_L(t)$ is noted across the hydraulic motor lines. The objective of the control law is that the actual angular velocity tracks the desired angular velocity. The measure of the actual angular velocity is sent to the control law via a sensor. The state-space equation (1) is developed from three equations. The first equation describes the motion equation of the load. The second equation is the continuity equation through the motor lines. The third equation shows the dynamics of the servo valve.

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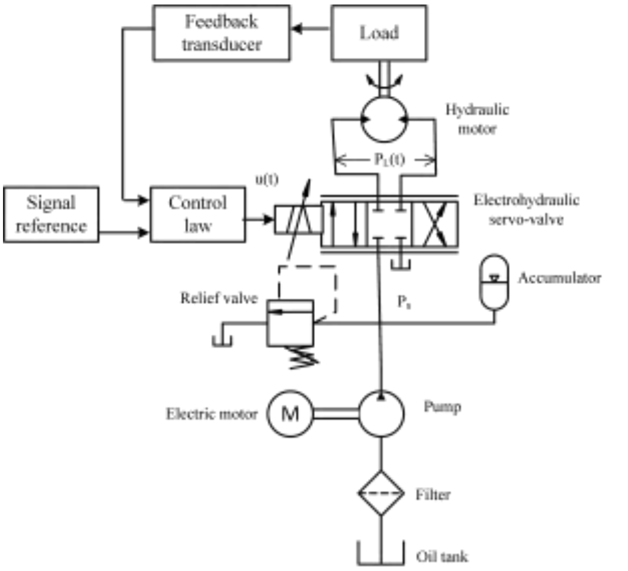

Fig. 1: Electro-hydraulic servo system

$$
\begin{aligned}
&\dot{x}_1(t)=\frac{d_m}{J}x_2(t)\text{-}\frac{B_m}{J}x_1(t)\\
&\dot{x}_2(t)=\frac{4\beta c_d}{V_m}\left(x_3(t)\frac{c_d}{\sqrt{\rho}}\sqrt{P_s\text{-}sigm(x_3(t))x_2(t)}\text{-}d_m x_1(t)\text{-}c_{sm}x_2(t)\right)\\
&\dot{x}_3(t)=\frac{K}{\tau}u(t)\text{-}\frac{1}{\tau}x_3(t)\\
&y(t)=x_1(t)
\end{aligned}
\tag{1}
$$

Where

$x_1(t)$ is the angular velocity $\dot{\theta}(t)$

$x_2(t)$ is the motor pressure difference due to the load

$x_3(t)$ is the servo-valve opening area due to the input signal

$u(t)$ is the control current input

J is the hydraulic motor's total inertia

d_m is the volumetric displacement of the motor

β is the fluid bulk modulus

V_m is the total oil volume of the hydraulic motor

c_d is the servo-valve discharge coefficient

ρ is the fluid mass density

c_{sm} is the leakage coefficient of the hydraulic motor

P_s is the supply pressure at the inlet of the servo valve

K is the servo-valve amplifier gain

τ is the servo-valve time constant

To satisfy the Lipschitz condition in this paper, we choose to approximate the sign function to the continuous function (2) proposed in the work of [22].

$$sign(x(t)) \square \frac{x(t)}{\sqrt{x^2(t)+\alpha}}=sigm(x(t)) \tag{2}$$

3 Backstepping Controller Design

In this section, the angular velocity backstepping controller is derived. It is the same controller presented in [21]. Here, we focus on the controller gains locations and tuning. The desired state variables are denoted by $x_{id}(t)$.

Now, consider the first subsystem $\dot{x}_1(t)=\frac{d_m}{J}x_2(t)\text{-}\frac{B_m}{J}x_1(t)$ of the state space model (1). The first candidate Lyapunov function for this subsystem is

$$V_1(t)=\frac{1}{2}e_1^2(t) \tag{3}$$

Where $e_1(t)=x_1(t)-x_{1d}(t)$. The time derivative of this Lyapunov function gives

$$\dot{V}_1(t)=e_1(t)\left(\frac{d_m}{J}e_2(t)+\frac{d_m}{J}x_{2d}(t)-\frac{b_m}{J}e_1(t)-\frac{b_m}{J}x_{1d}(t)-\dot{x}_{1d}\right) \tag{4}$$

If we choose $x_{2d}(t)$ as the first virtual control such that

$$x_{2d}(t)=\frac{J}{d_m}\left(\frac{b_m}{J}x_{1d}(t)+\dot{x}_{1d}-k_1e_1(t)\right) \tag{5}$$

Where the first gain controller $k_1>0$, we obtain

$$\dot{V}_1(t)=-\left(\frac{b_m}{J}+k_1\right)e_1^2(t)+\frac{d_m}{J}e_2(t)e_1(t) \tag{6}$$

Now, we choose the second candidate Lyapunov function for the second subsystem of (1)

$$\dot{x}_2(t)=\frac{4\beta c_d}{V_m}\left(x_3(t)\frac{c_d}{\sqrt{\rho}}\sqrt{P_s\text{-}sigm(x_3(t))x_2(t)}\text{-}d_m x_1(t)\text{-}c_{sm}x_2(t)\right)$$

as

$$V_2(t)=\frac{1}{2}e_1^2(t)+\frac{1}{2}e_2^2(t) \tag{7}$$

Its time derivative gives

$$
\begin{aligned}
\dot{V}_2(t)=&-\left(\frac{b_m}{J}+k_1\right)e_1^2(t)\\
&+e_2(t)\left(\begin{aligned}&\frac{d_m}{J}e_1(t)+\frac{4\beta c_d}{v_m\sqrt{\rho}}e_3(t)\sqrt{P_s-sigm(x_3(t))x_2(t)}\\&+\frac{4\beta c_d}{v_m\sqrt{\rho}}x_{3d}(t)\sqrt{P_s-sigm(x_3(t))x_2(t)}-\frac{4\beta d_m}{v_m}x_1(t)-\frac{4\beta c_{sm}}{v_m}x_2(t)-\dot{x}_{2d}(t)\end{aligned}\right)
\end{aligned}
\tag{8}
$$

If we choose $x_{3d}(t)$ as the second virtual control such that

$$x_{3d}(t) = \frac{v_m\sqrt{\rho}}{4\beta c_d\sqrt{P_s - sigm(x_3(t))x_2(t)}}\begin{pmatrix} -\frac{d_m}{J}e_1(t) + \frac{4\beta d_m}{v_m}x_1(t) \\ +\frac{4\beta c_{sm}}{v_m}x_{2d}(t) + \dot{x}_{2d}(t) - k_2 e_2(t) \end{pmatrix} \tag{9}$$

Where the second controller $k_2 > 0$, we obtain

$$\begin{aligned} \dot{V}_2(t) = &-\left(\frac{b_m}{J} + k_1\right)e_1^2(t) - \left(\frac{4\beta c_{sm}}{v_m} + k_2\right)e_2^2(t) \\ &+ \frac{4\beta c_d}{v_m\sqrt{\rho}}e_2(t)e_3(t)\sqrt{P_s - sign(x_3(t))x_2(t)} \end{aligned} \tag{10}$$

Finally, consider the third subsystem $\dot{x}_3(t) = \frac{K}{\tau}u(t) - \frac{1}{\tau}x_3(t)$. Choose the final candidate Lyapunov function for this subsystem as

$$V_3(t) = \frac{1}{2}e_1^2(t) + \frac{1}{2}e_2^2(t) + \frac{1}{2}e_2^3(t) \tag{11}$$

The time derivative of this Lyapunov function gives

$$\begin{aligned} \dot{V}_3(t) = &-\left(\frac{b_m}{J} + k_1\right)e_1^2(t) - \left(\frac{4\beta c_{sm}}{v_m} + k_2\right)e_2^2(t) \\ &+ e_3(t)\begin{pmatrix} \frac{4\beta c_d}{v_m\sqrt{\rho}}e_2(t)\sqrt{P_s - sign(x_3(t))x_2(t)} + \frac{K}{\tau}u(t) \\ -\frac{1}{\tau}e_3(t) + \frac{1}{\tau}x_{3d}(t) - \dot{x}_{3d}(t) \end{pmatrix} \end{aligned} \tag{12}$$

If we choose the control signal $u(t)$ such that

$$u(t) = \frac{\tau}{K}\left(\frac{1}{\tau}x_{3d}(t) + \dot{x}_{3d}(t) - \frac{4\beta c_d}{v_m\sqrt{\rho}}e_2(t)\sqrt{P_s - sigm(x_3(t))x_2(t)} - k_3 e_3(t)\right) \tag{13}$$

Where the third controller gain $k_3 > 0$, we obtain

$$\dot{V}_3(t) = -\left(\frac{b_m}{J} + k_1\right)e_1^2(t) - \left(\frac{4\beta c_{sm}}{v_m} + k_2\right)e_2^2(t) - \left(\frac{1}{\tau} + k_3\right)e_3^2(t) \tag{14}$$

Fig. 2 shows the implementation of the backstepping controller in Matlab /Simulink with the highlight of the three steps.

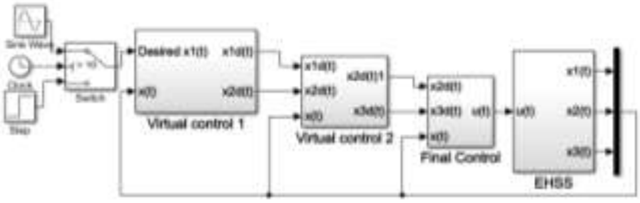

Fig. 2: Closed-loop controlled system block diagram in Matlab /Simulink environment

3.1 Tuning Issue Analysis

One can note that the gains of our backstepping controller are chosen such that the time derivative of the final Lyapunov function gives (14). In most works, [13], [19], [20], these gains are located such that the other terms $\frac{b_m}{J}$, $\frac{4\beta c_{sm}}{v_m}$ and $\frac{1}{\tau}$ do not appear in the time derivative of the Lyapunov function. Because (14) leads to inequality (15), the Lasalle principle states that the tracking errors $e_1(t), e_2(t)$ and $e_3(t)$ go to zero as time goes to infinity.

$$\dot{V}_3(t) \le -k_1 e_1^2(t) - k_2 e_2^2(t) - k_3 e_3^2(t) \le 0 \tag{15}$$

Unlike the feedback linearization controller or PID controller, the backstepping controller does not show a weighted sum of the angular velocity error and its time derivatives. The different tracking errors $e_i(t)$ are not related in a direct sense. The angular velocity tracking error occurs in the first virtual control. The pressure difference tracking error occurs in the second virtual control and so on. The gains of the different tracking errors, k_1, k_2 and k_3 must be adjusted in a given order shown by the work of [16], where certain dynamics of the state variables are neglected. In [20], the authors show that varying the controller gains one by one may affect the robustness of the closed-loop response under parametric uncertainties. Their backstepping controller has two variable inputs. Our control law has one variable input. Additionally, no disturbance is present and the focus is on the effect of each gain in the closed-loop performance. We vary the value of each gain while the others are set to zero. Our objective is to see if a heuristic method like the one of Ziegler Nichols can be developed.

4 Results

In this section, the results of the numerical simulation are presented. The performances of the proposed backstepping controller are obtained in the

Matlab/ Simulink environment using the closed-loop block diagram of Fig. 2. The sampling time is 0.01 seconds and the simulation is performed in 20 seconds. Table 1 lists the numerical value used for the simulation.

Table 1. Numerical values used for the simulation

Symbol	Description	Value and units
	EHSS	
α	Sigmoid function constant	10^2
τ	Servo valve time constant	0.01 s
K	Servo valve amplifier gain	$8.10^{-7} m^2/mA$
V_m	Total oil volume of the hydraulic motor	$3\ 10^{-4} m^3$
β	Fluid bulk modulus	$8\ 10^8$ Pa
c_d	Flow discharge coefficient	0.61
P_s	Supply pressure	$9\ 10^6$
c_{sm}	Leakage coefficient	$9\ 10^{-13}\ m^5/$
d_m	Volumetric displacement of the motor	$3\ 10^{-6} m^3/rad$
ρ	Fluid mass density	$900\ Kg/m^3$
J	Total inertia of the motor and	$0.05\ N.m.s^2$
B	Viscous damping coefficient	0.2 N.m.s

Fig. 3 shows the reference signal describing the desired trajectory of the angular velocity used for the simulation. In the first ten seconds, the desired angular velocity is a step of amplitude 1 rad/s. In the last ten seconds, the desired angular velocity has a sinusoidal waveform with an amplitude of 1 rad/s and a frequency of 2 rad/s.

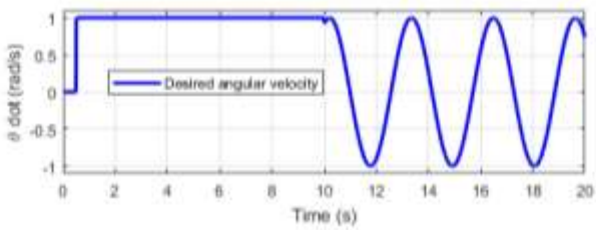

Fig. 3: Reference signal used for the simulation

The first sets of simulations present the performances of the backstepping controller when the gains k_1, k_2 and k_3 equal 0. Fig 4 shows that angular velocity tracking error does not converge to zero as time goes to infinity. To appreciate the infinity response behaviour, the time of simulation is extended to 80 seconds. The tracking error does not go to infinity as time goes to infinity. It is seen that its value is limited to approximately 90 rad/s. According to (14), the time derivative of the deducing Lyapunov function is negative define. Hence, the equilibrium state $e_i(t)=0$ is asymptotically stable. In Fig 5, at the start of the simulation, the zoomed view shows that the tracking error is negligible before 4 seconds. The tracking error converges to 0 before 4s. However, without a disturbance in the closed loop system at 3,5 seconds, the system response starts to diverge from the desired output. We can conclude that the equilibrium state is not asymptotically stable and diverges to a limit cycle or other equilibrium state.

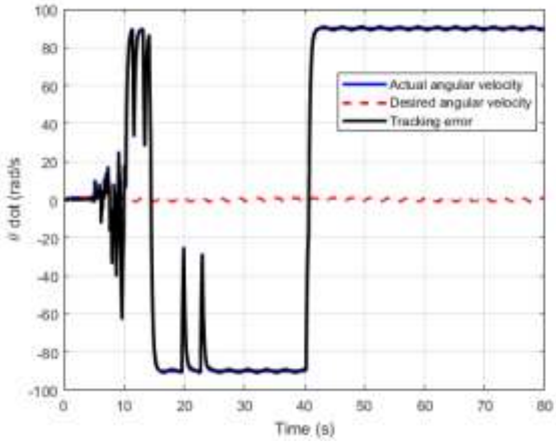

Fig. 4: System response when $k_1=k_2=k_3=0$

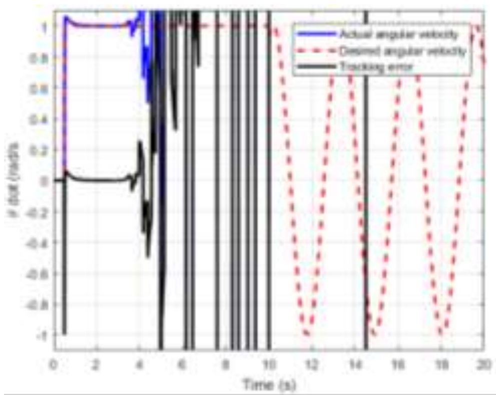

Fig. 5: Zoomed view of system response when $k_1=k_2=k_3=0$

4.1 k2 and k3 Gains Tuning Results

The next set of figures shows the backstepping controller performances where the first gain k_1 is 0. The gains k_2 and k_3 are varied to see the effect of these gains on the backstepping controller performance. Fig. 6 and Fig. 7 show that the response obtained with $k_1=0$ gives the same result obtained in the previous section. The angular velocity tracking error converges to 0 at the start of the simulation. As time goes to infinity, the tracking error diverges from zero but remains bounded around ± 90 rad/s. The variation of k_2 and k_3 gains does not affect the backstepping controller performances.

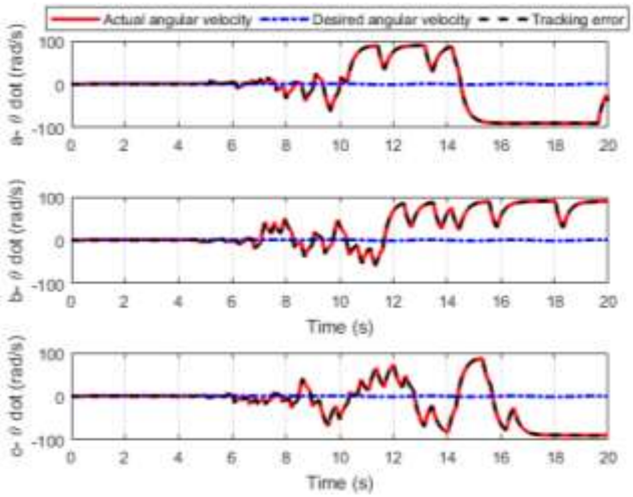

Fig. 6: System response when $k_1 = k_3 = 0$ a) $k_2 = 0$ b) $k_2 = 100$ and c) $k_2 = 1000$

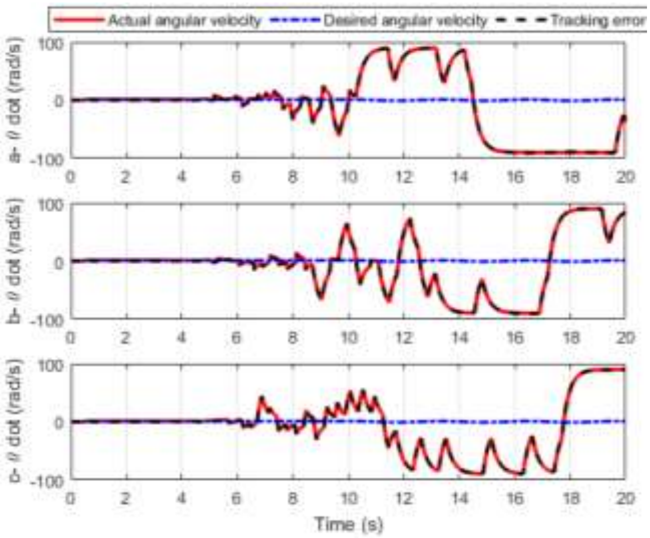

Fig. 7: System response when $k_1 = k_2 = 0$ a) $k_3 = 0$ b) $k_3 = 100$ and c) $k_3 = 1000$

4.2 k1 Gain Tuning Results

In this section, the response of the proposed backstepping controller is analyzed by varying the k_1 gain while the other gains are set to 0. Fig 8 shows that the k_1 gain strongly affects the behaviour of the closed-loop response. The convergence of the angular velocity varies while changing the tuning of the k_1 gain. The closed-loop response displays the same behaviour encountered in the previous section when $k_1 = 0$. The response behaviour drastically changes when $k_1 \neq 0$. In Fig.8 a and Fig. 8 b, we note that the response remains limited at 90 rad/s. Meanwhile, in Fig 8 c, when $k_1 = 1000$, high-frequency sustained oscillations with amplitude less than 40 rad/s are visible in the response. In Fig 9, the k_1 gain is varied with a larger sweep to observe further changes. However, beyond the value of 1000, the response shows the same profile.

Fig. 10 shows that the tracking error converges to 0 when the value of the k_1 gain is close to 27. In Fig 10. b, the tracking error is negligible when $k_1 = 27.2$. In Fig 10. a, the tracking error shows large overshoots at 10 s when the reference signal changes its form. The zoomed view of the response, when $k_1 = 27$ according to Fig. 11, shows that the tracking error is small on either side of the change in the profile of the reference. In Fig 10. c, the chattering effect or high frequency sustained oscillations with small amplitude occurs in the backstepping controller response when $k_1 = 27.5$. According to [16], there is a trade-off between the robustness and the chattering effect in the closed-loop response while adjusting the k_1 gain. The robustness, in the simulations where k_1 is about 27, is described by the change in the reference signal. As one can see, large overshoots occur in Fig.10 a while no overshoot is visible in Fig. 10 c.

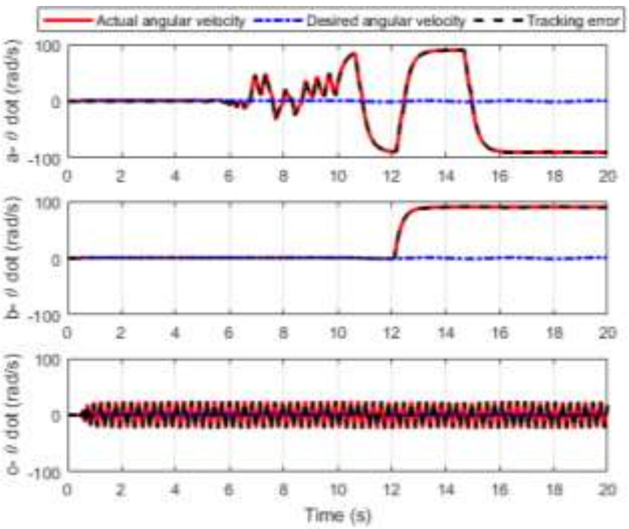

Fig. 8: System response when $k_3 = k_2 = 0$ a) $k_1 = 0$ b) $k_1 = 100$ and c) $k_1 = 1000$

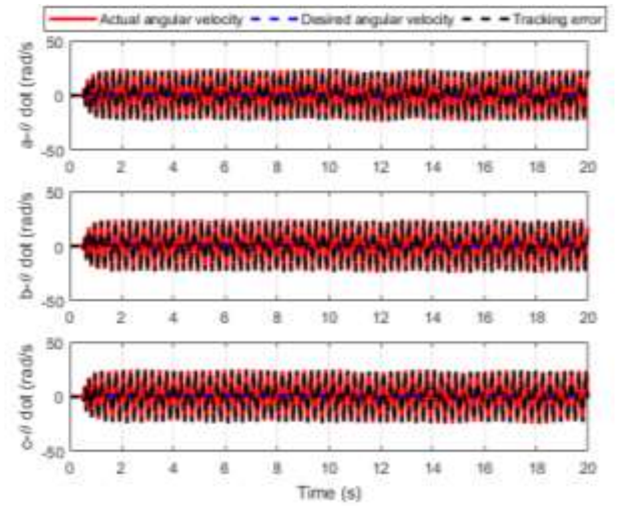

Fig. 9: System response when $k_3 = k_2 = 0$ a) $k_1 = 1000$ b) $k_1 = 2000$ and c) $k_1 = 3000$

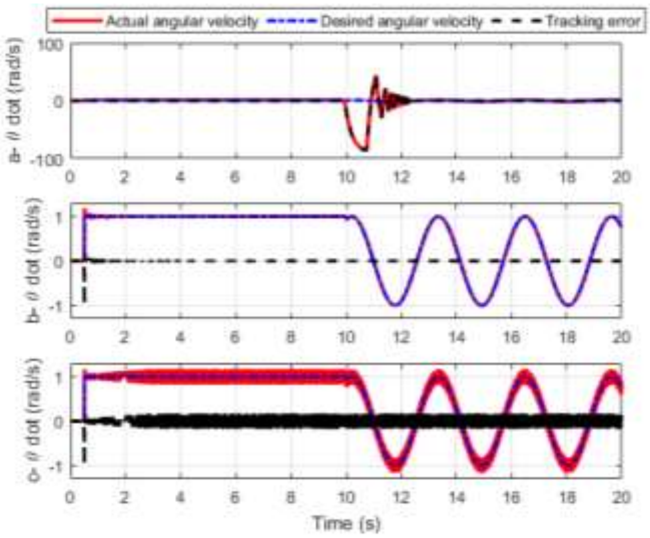

Fig. 10: System response when $k_3 = k_2 = 0$ a) $k_1 = 27$ b) $k_1 = 27.2$ and c) $k_1 = 27.5$

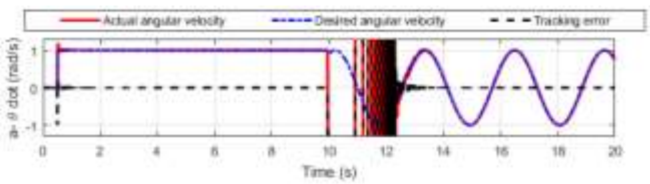

Fig. 11: Zoomed view of system response when $k_3 = k_2 = 0$ and $k_1 = 27$

5 Discussion and Gains Tuning Method Perspective

In this section, we discuss two issues encountered in the results section.

The first problem concerns the change in the response profile by varying the k_1 gain while no change is noted with the other gains. In (5), the k_1 gain is the coefficient of the feedback term appearing in the second virtual control. When k_1 is 0, the first virtual control does not give the desired second state variable. Since the second virtual control depends on the first one via the backstepping effect, the desired third state variable is also biased. The k_1 value provides feedback on the actual angular velocity in the backstepping controller. This feedback link guarantees the convergence of the tracking error to zero. However, without this link, the tracking error does not maintain the convergence as time goes to infinity.

The second problem concerns the limit values indicated in certain simulations. This problem may be related to the first one because the biased virtual control leads to biased tracking errors. Indeed, our tracking error is calculated using the reference signal. However, without the feedback link, the equilibrium point in the Lyapunov function is not the desired one. The limit value noted in some simulations may be the biased equilibrium point.

6 Conclusion

In this paper, we address the problem of adjusting the gains of the non-self-tuning backstepping controller. The objective is to analyze the effect of each controller gain in the closed-loop response to propose a standardized approach to tune these gains. The results show that only one of the three gains affects the convergence of the tracking error to zero as time tends to infinity. This gain provides the feedback link of the actual angular velocity in the backstepping controller. The calculated virtual controls are not those expected when this gain is not well adjusted. Future works will investigate the analytic value of this critical gain. The effect of the other gains once the critical gain is adjusted, will be studied to propose a rigorous gain tuning method.

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Contribution of Individual Authors to the Creation of a Scientific Article (Ghostwriting Policy)

-Honorine Angue Mintsa and Rolland Michel Assoumou Nzué carried out the design and the Matlab Simulink implementation of the backstepping controller.
-Gérémino Ella Eny was responsible for the literature review and the analysis of the results shown in Section 4.
-Nzamba Senouveau executed the numerical results by varying the different gains in the controller.

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Conflict of Interest

The authors have no conflict of interest to declare.