



(RESEARCH ARTICLE)



ELECTROHYDRAULIC SERVO MOTOR VERSUS ELECTRIC SERVO DC MOTOR IN THE PRESENCE OF MECHANICAL DISTURBANCES

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ABSTRACT

The manipulation of the rotating mechanical loads in an industrial context relies on the choice of actuator type. Several studies report on the comparison of power, size, and other technical specifications. This paper presents a comparative study of the angular velocity control of a mechanical load driven by an electro-hydraulic servo motor and the same load driven by an electric DC servo motor under identical mechanical disturbances. First, both motors are mathematically modelled to highlight their nonlinear dynamics. Second, a velocity PID controller is developed using the classical Ziegler-Nichols approach for each motor. Finally, the nonlinear servosystems are implemented and run in Matlab / Simulink with a mechanical disturbance profile. It is shown that the electric servo motor proves more robust than its electro-hydraulic equivalent, particularly in the presence of mechanical disturbance.

Keywords: Electrohydraulic Motor, Electrical Dc Motor, PID Control, Mechanical Disturbance, Servomechanism.

1 INTRODUCTION

In the industry, the handling of rotating mechanical loads remains a relevant topic. Aeronautics, materials handling, automotive, and agricultural machinery are just a few applications where rotating actuators are used. The most common power transmission systems are hydraulic, pneumatic, or electric. Their choice depends on the load type, duty cycle, environmental conditions, cost, safety and maintenance. Several studies have been conducted to help engineers better select the type of actuator. It has been shown that the power density of hydraulic motors is ten times greater than that of electric and pneumatic motors [1]. The power-to-size ratio, compact format and rapid response make the hydraulic systems essential, especially when dealing with heavy loads. However, it has been reported that electric machines are preferred to hydraulic machines for handling medium loads because of their energy efficiency[2]. Furthermore, the

authors [3] show that for the same power densities, the electric motor is faster. Nevertheless, the oversized dimensions of some electric motors for relatively simple applications discourage their use. Combining hydraulics and electricity has improved the energy efficiency and the speed of hydraulic systems, while increasing the power-to-size ratio of the electrical system. Thus, electrohydraulic actuators offer a more energy-efficient, more robust and faster solution when manipulating heavy loads [4] [5].

While several studies guide the choice of actuator type based on energy efficiency, size-to-power ratio, maintenance, and cost, there are few studies that perform a comparative analysis of the control management of these machines. The control of each actuator is developed and studied separately. The authors [6] [7] demonstrate the design and the implementation of control laws applied to an electric motor. Others examine the robustness of the closed loop in the presence of disturbance [8]. On the other hand, the authors [9] [10] investigate the design and implementation of control laws on an electro-hydraulic motor. Others also study the robustness of the control laws [11].

In this article, we conduct a comparative study on the angular velocity control of an Electric servoMotor (EM) and an ElectroHydraulic ServoMotor (EHSM) in the presence of disturbances and applied to the same mechanical load. The present comparative analysis is certainly not dimensional, but it is based on the same controller design methodology. And the both implementations are applied to the same reference signal and same mechanical load and disturbance. Thus, the objectives achieved in this study can be summarised in the following points:

- A parallel mathematical modelling of the two actuators;
- A parallel design of the angular velocity PID control law;
- A comparative analysis of the closed-loop performances on a similar load and mechanical disturbance.

The rest of the paper is organized as follows: Section II describes the mathematical modelling of the electrohydraulic servo motor. Section III shows the mathematical modelling of the electric servo motor. Section IV presents the numerical results of the two closed loops in the presence of the mechanical disturbance. Finally, the conclusion is drawn in section V.

2 ELECTROHYDRAULIC SERVO MOTOR

Figure 1 shows a schematic of the EHSM under consideration. The system has two parts: the operational part and the control part. In the operational part, the positive displacement pump delivers hydraulic oil from the tank to the electrohydraulic servovalve. The relief valve limits the maximum pressure in the system. Depending on the input control signal $i(t)$, the electrohydraulic servovalve opens a passage allowing oil to supply the hydraulic motor. When the load-induced pressure is reached, the mechanical load is moved with an angular velocity $\dot{\theta}(t)$. Then, the oil returns to the tank via the other passage of the servo valve. In the control part, the output signal, corresponding to the motor speed, is captured by the feedback transducer and transmitted to the control computer for comparison with the reference signal. According to the control law specifications, an electrical control signal is calculated and sent to the servovalve to open the oil passage.

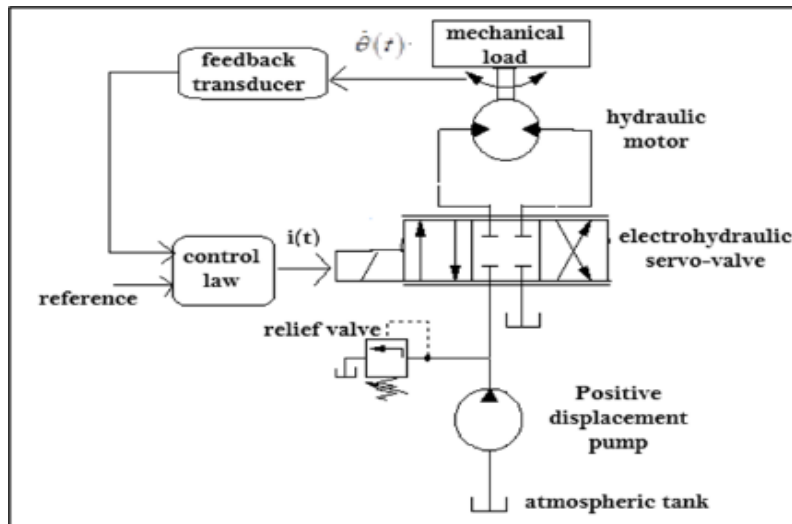


Figure 1: Schematic diagram of the EHSM

The EHSM can be broken down into three subsystems, resulting in three modeling equations. The first equation describes the dynamics of the mechanical load. The second equation describes the dynamics of the hydraulic fluid across the motor. The third equation describes the dynamics of the servovalve, which provides the interface between the control and operating sections. Then, we obtain the following third nonlinear state space model with $x_1(t)$, $x_2(t)$, $x_3(t)$ and $u(t)$ being the angular velocity of the motor, pressure difference across the motor lines, the servovalve opening area and the input signal control respectively. The other parameters are tabulated in Table I.

$$\begin{aligned}\dot{x}_1(t) &= \frac{1}{J} (D_m x_2(t) - B x_1(t)) \\ \dot{x}_2(t) &= \frac{4\beta c_d}{V_m \sqrt{\rho}} x_3(t) \sqrt{P_s - \text{sign}(x_3(t)) x_2(t)} \\ &\quad - \frac{4\beta D_m}{V_m} x_1(t) - \frac{4\beta c_{sm}}{V_m} x_2(t) \\ \dot{x}_3(t) &= \frac{1}{\tau} (K u(t) - x_3(t))\end{aligned}\quad (1)$$

We observe that the system is strongly nonlinear with a continuous square-root function and a discontinuous sign function. In the absence of a physical electrohydraulic system for this experiment, we used a numerical simulation module implemented in Matlab as shown in Figure 2.

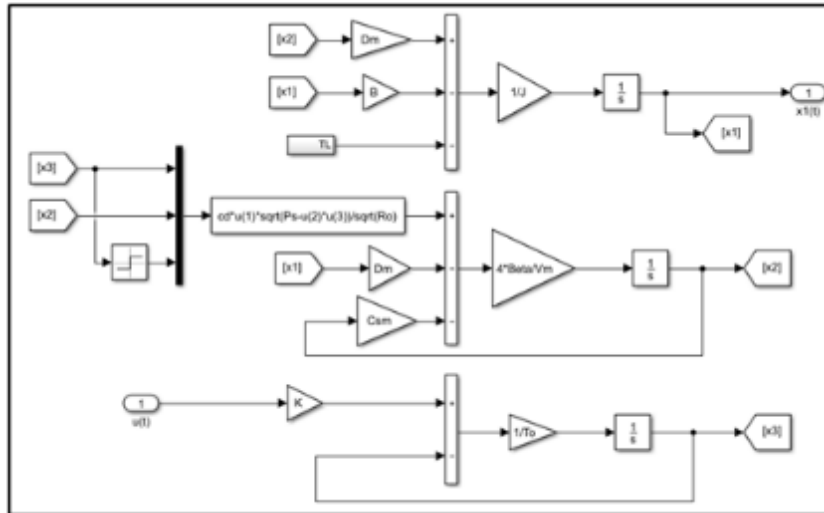


Figure 2: Implementation of the EHSM in Matlab / Simulink

3 ELECTRICAL SERVO DC MOTOR

Figure 3 presents the schematic of the EM under consideration. This is a DC motor with a separate magnetic field. The system has two parts: the operational part and the control part. In the operational part, the magnetic field is created in the field circuit, which constitutes the stator. When a voltage $V_a(t)$ is applied to the armature circuit surrounding the rotor, the rotor begins to rotate under the influence of electromagnetic forces at an angular velocity $\dot{\theta}(t)$. In the control part, the output signal, corresponding to the motor speed, is captured by the feedback transducer and transmitted to the control computer for comparison with the reference signal. According to the control law specifications, an electrical control signal is calculated and sent to the armature voltage input to create a current in the armature.

The EM can be broken down into three subsystems, resulting in three modeling equations. The first equation describes the dynamics of the mechanical load. The second equation describes the dynamics of the electrical current in the armature. The third equation describes the dynamics of the current in the magnetic field. Then, we obtain the following third nonlinear state space model with $x_1(t)$, $x_2(t)$, $x_3(t)$ and $u(t)$ being the angular velocity of the motor, the current in the armature, the current in the magnetic field and the input signal control, respectively. The other parameters are tabulated in Table I.

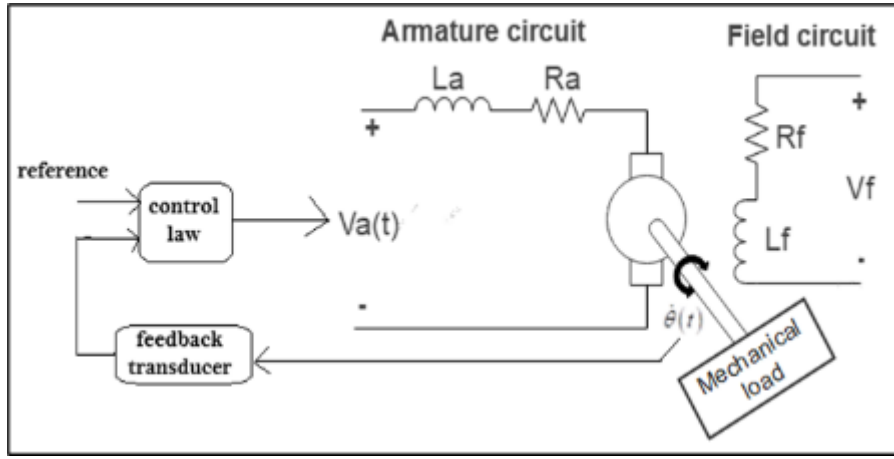


Figure 3: Schematic diagram of the EM

$$\begin{aligned}
 \dot{x}_1(t) &= \frac{1}{J} (c_2 x_3(t) x_2(t) - B x_1(t)) \\
 \dot{x}_2(t) &= \frac{1}{L_a} (u(t) - R_a x_2(t) - c_1 x_1(t) x_2(t)) \\
 \dot{x}_3(t) &= \frac{1}{L_f} (V_f - R_f x_3(t))
 \end{aligned}
 \tag{2}$$

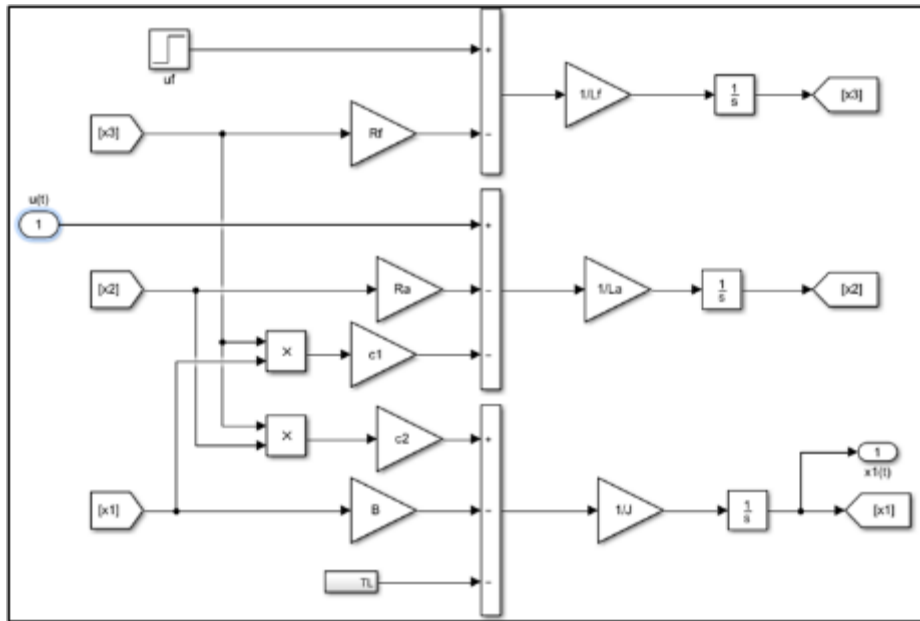


Figure 4: Implementation of the EM in Matlab / Simulink

Now, we observe that the system is strongly nonlinear with two continuous product functions acting on two equations. The Matlab simulation module replacing the physical electric motor for this experiment is shown in Figure 4.

4 NUMERICAL RESULTS AND DISCUSSION

This section is devoted to the results of numerical simulations of the two closed-loop controlled motors as shown in Figure 5 and Fig.6 in the Matlab environment. The tests performed on each motor lasted 30 seconds. The control laws used here are the traditional PID controller synthesized from the Ziegler-Nichols approach. All the numerical values used in this study are listed in the Table I.

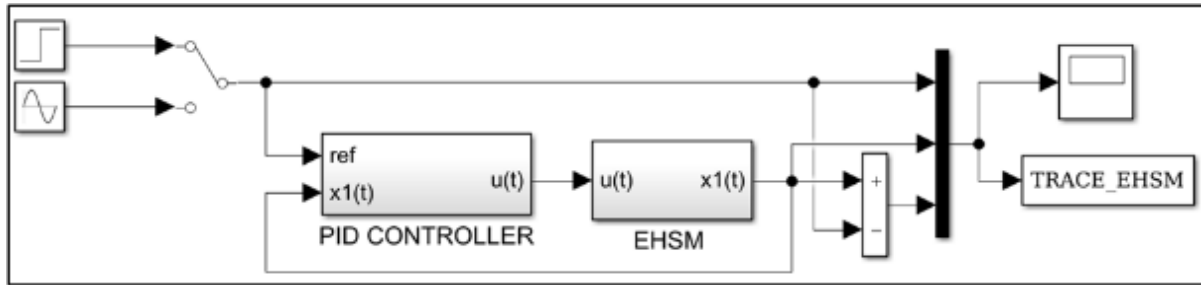


Figure 5: Implementation of the closed loop PID control of the EHSM in Matlab / Simulink

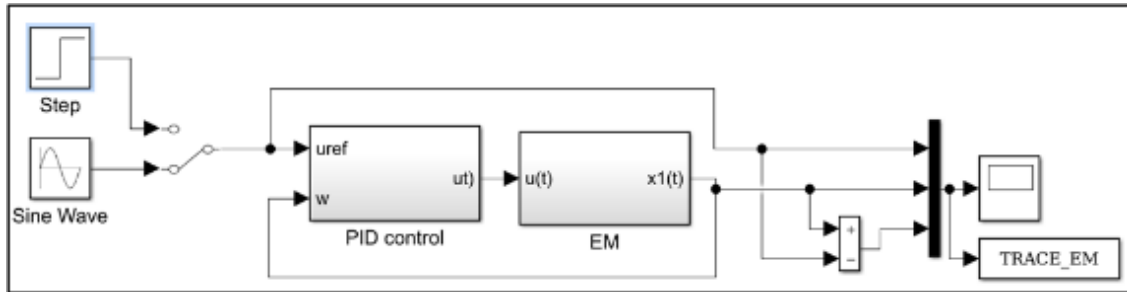


Figure 6: Implementation of the closed loop PID control of the EM in Matlab / Simulink

Table 1: Parameters value

Quantity	Symbol	Value and unit
Electrohydraulic servo motor		
Fluid mass density	ρ	900 kg/m ³
Volumetric displacement of the motor	D_m	3x10 ⁻⁶ m ² .rad ⁻¹
Oil bulk modulus	β	8x10 ⁸ Pa
Flow discharge coefficient	c_d	0,61
Total oil volume in the hydraulic motor	V_m	3x10 ⁻⁴ m ³
Servovalve constant	τ	0,01 s
Leakage coefficient	c_{sm}	1x10 ⁻¹³ m ⁵ /(N.s)
Supply Pressure	P_s	9x10 ⁶ Pa
Servovalve gain	K	10 ⁻⁵ m ² /mA
Electric servo motor		
Resistance of the armature circuit	Ra	1,5 Ω
Inductance of the armature circuit	La	0,012 H
Resistance of the field circuit	Rf	70 Ω
Inductance of the field circuit	Lf	80 H
Back emf constant	c_1	1,8 V.s/rad
Torque constant	c_2	0,3 Nm/A
Mechanical load		

Total inertia of the mechanical load	J	1 N.m.s ²
Viscous damping coefficient	B	10 N.m.s
PID Control parameters for the EHSM		
Proportional gain	k _p	0,12
Integral gain	k _i	1,6
Dérivative gain	k _d	0,0022
PID Control parameters for the EM		
Proportional gain	k _p	3 000
Integral gain	k _i	40 000
Dérivative gain	k _d	56,25

Figure 7 illustrates the profile of the mechanical disturbances applied to the two motors. The disturbance is 0 Nm from 0 second to 5 seconds. Between 5 seconds and 15 seconds, it is described by a sinusoidal wave with an amplitude of 10 Nm and a frequency of 1 rad/s. Then from 15 seconds until the end of the simulation, the disturbance returns to 0 Nm.

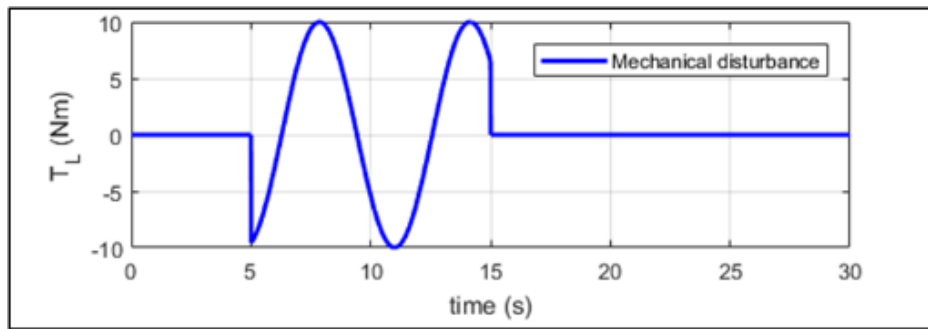


Figure 7: Perturbation profile during the simulation

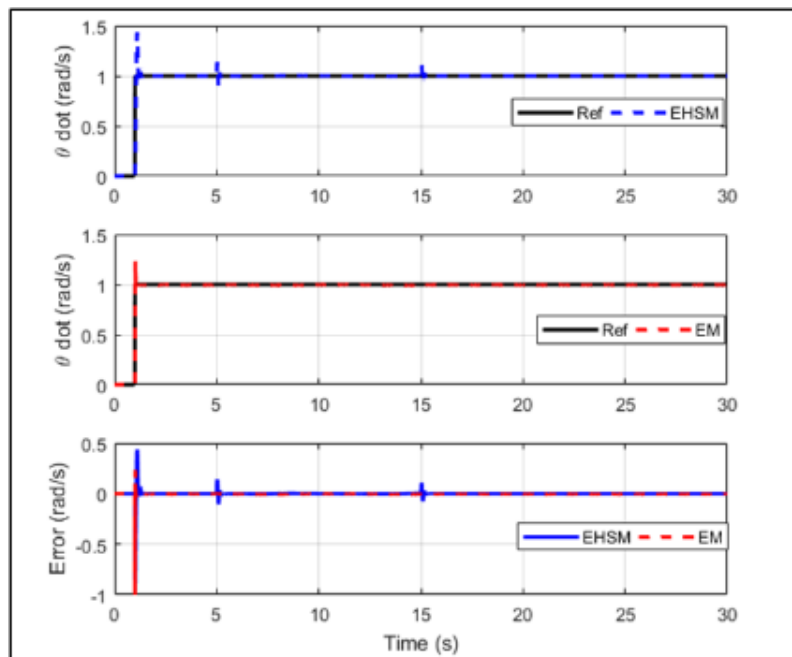


Figure 8: Closed-loop response when using the EHSM, the EM with a constant reference of amplitude 1 rad/s

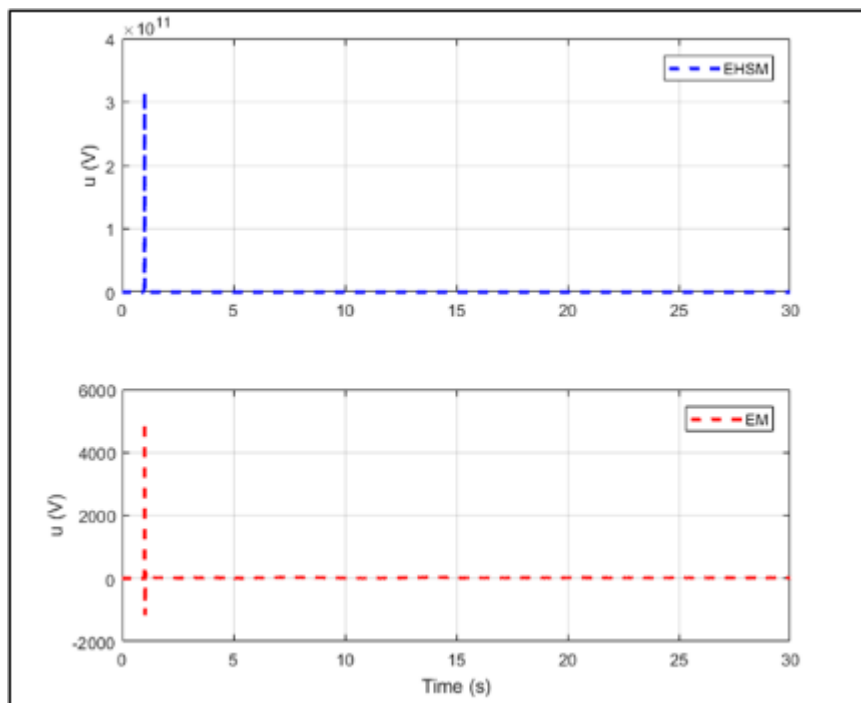


Figure 9: Control signal when using the EHSM, the EM with a constant reference of amplitude 1 rad/s

Figures 8 and 9 show the numerical results obtained for each motor when the reference signal has a constant amplitude of 1 rad/s. It is clear that the performance of the electric servomotor is superior to that of the electrohydraulic servomotor. In the Figure 8 showing the comparison of the two errors, we see that the error in the response of the electro-hydraulic servo motor reaches 50% at the beginning of the simulation, while the error observed in the electric motor barely reaches 15%. It is also noted that at the beginning and end of the mechanical disturbance, i.e. at 5 seconds and 15 seconds, the error of the electro-hydraulic servosystem reaches 10%, while no error is generated by the disturbances in the electrical servomotor. Figure 9 shows the control signal of the two motors. We observe that no additional effort is visible, whether the disturbances are triggered or not. At least, this effort is much less than that obtained at the beginning of the simulation during the step stage of the reference.

The next set of figures, i.e. Figures 10 and 11, presents the numerical results obtained with a sinusoidal reference signal of amplitude 1 rad/s and frequency of 1 rad/s. Here again, it is noted that angular velocity control of the electric servomotor is more effective than that of an electrohydraulic servomotor in the presence of the mechanical disturbances. The amplitude of the tracking error is almost identical to that observed when the reference signal is constant in the presence of disturbance. In the Figure 10 showing the error comparison, the electro-hydraulic servomotor exhibits a 15% error when the disturbance is triggered, and then a 10% error when it is stopped. On the other side, the electric servomotor displays a near-zero error. Figure 11 shows that the angular velocity control of an electrohydraulic servomotor provides more effort than that of an electric servomotor in the presence of disturbances.

Three reasons could explain the difference in performance observed between the two servomotors. Indeed, this is an angular velocity control of the rotational mechanical load in the presence of the same mechanical disturbances and using the same controller design procedure. The first reason is the absence of dimensional analysis before comparing the two servomotors.

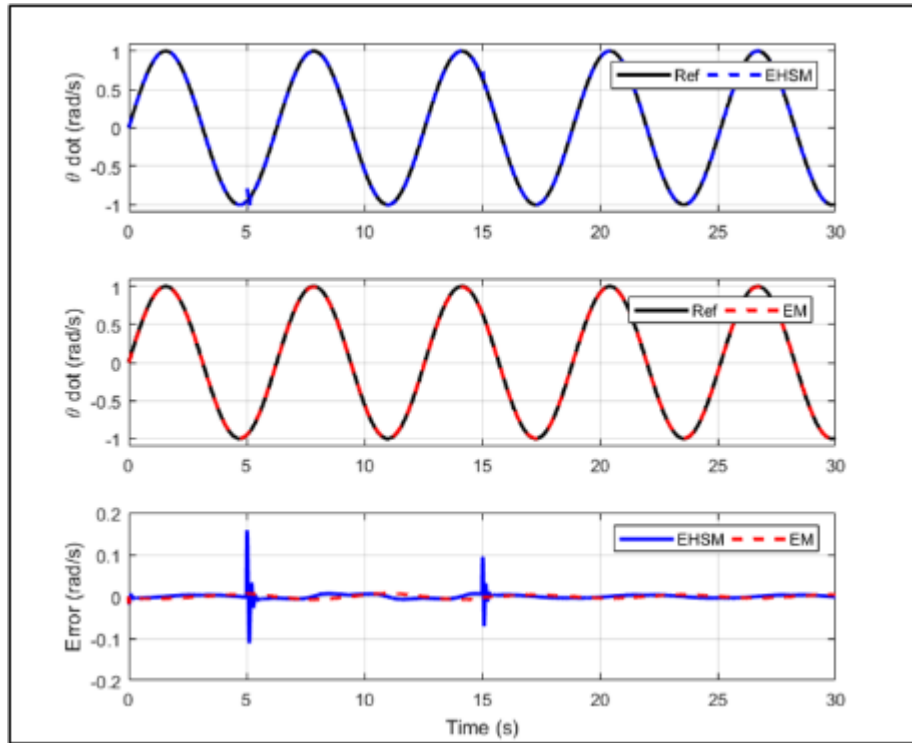


Figure 10: Closed-loop response when using the EHSM, the EM with a sinusoidal reference of amplitude 1 rad/s and frequency of 1 rad/s

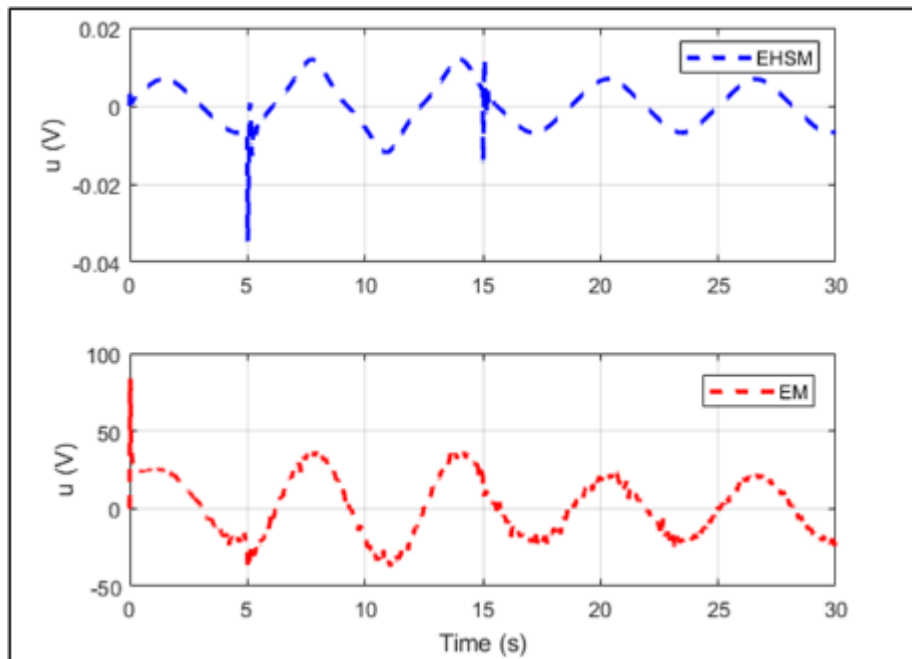


Figure 11: Control signal when using the EHSM, the EM with a constant reference of amplitude 1 rad/s

Several studies have noted that failure to respect scalability and universality can lead to errors in judgment when choosing or comparing actuators [12] [13]. The second reason could be the difference in dynamics between the two servosystems. The authors [14] [15] demonstrate that there is a relationship between the robustness and the system order. By comparing the mathematical models rendered in their controllable canonical forms, i.e. in the direct input-output relationship, we find that the order of the canonical controllable form of the electrohydraulic servomotor is 3 while that of the electric servomotor is 2. Then, the third reason could be the choice of the control law. The PID control law is derived from linear systems theory. It is synthesized for these two servomotors with an operating point of 1 rad/s.

Both servomotors exhibit different nonlinearities. The presence of disturbances, even similar ones, leads to independent deviation in the two servomotor dynamics, outside the validity limits of the linear corrector.

5 CONCLUSION

This study focused on comparing the performance of an angular velocity control of a mechanical load using an electrohydraulic servomotor and an electric servomotor. It has been demonstrated that the angular velocity control using the electrical solution is more robust than the electrohydraulic solution in the presence of mechanical disturbances. Furthermore, an additional comparative analysis by normalizing the two actuators for comparable power, bandwidth and physical scale and the use of nonlinear control laws are needed to justify and confirm this finding.

STATEMENTS ON COMPLIANCE WITH ETHICAL STANDARDS

Disclosure of conflict of interest

The authors declare that there is no conflict of interests.

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