Digital Electro Hydrostatic Actuator with Variable Speed Digital Hydraulic Pump: A Design Overview

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Abstract.

The electric drive technology has achieved significant maturity and established a new global trend focused on the development of more electric and efficient systems. In the field of hydraulic systems, works on digital hydraulics have shown that it is an attractive solution to improve the energy efficiency. In this context, this paper presents a new approach design of a digital hydraulic actuator comprising a symmetrical cylinder controlled by a digital hydraulic pump driven by a variable rotational speed source in order to maximize the hydraulic energy savings and the controllability of the system. The simulated results show that the non-dissipative control strategy implemented by this system topology can reduce significantly the energy losses. Moreover, the use of a variable rotational speed source makes it possible to obtain a continuous flow rate for each pump unit combination, which increases the digital hydraulic pump flow rate range. A servo-hydraulic actuator was used as a baseline to compare the behaviour and the energy savings. The analysis shows that the baseline system has consumed about ten times more energy than the proposed digital hydraulic solution.

Keywords. Digital Hydraulics, Digital Hydraulic Pump, Variable-Speed Digital Electro Hydrostatic Actuator.

1. INTRODUCTION

In recent decades, new environmental regulations have led researchers to develop energy-optimized systems to improve energy savings. Latest studies have presented the digital hydraulic as a promising alternative to improve the energy efficiency of hydraulic systems. In the digital hydraulic concept, the servo valve is replaced by a set of on/off valves in order to avoid the energy dissipation of the throttling control, commonly used on typical hydraulic position control systems. Additionally, it is possible to implement different control strategies to control the opening and closing movements of the on/off valves, which brings some flexibility to the system [1 – 4].

The technological advances in electric drives have started a new global trend based on the use of electricity as a major power source. This tendency presents some benefits such as high efficiency, weight reduction, and reduced maintenance costs [5]. In the aviation industry, the More Electric Aircraft – MEA concept aims to replace the typical servo valve controlled hydraulic actuators (SHA) and its centralized power unit by the use of decentralized and power-by-wire systems [6].

Considering the global electrification tendency and recent research in digital hydraulics, this study proposes a new hydraulic actuator design, which comprises a symmetrical multi-chamber cylinder, a digital hydraulic pump controlled by a variable rotational speed motor, and a set of free leakage on/off seat valves. The objective of this paper is to describe the main aspects of the preliminary design of the proposed solution by evaluating its dynamic model, which was sized according to typical flight control surface requirements. A servo hydraulic actuator model is used as a baseline to compare the dynamic response and the hydraulic energy consumed.

2. DIGITAL HYDRAULICS

Digital Hydraulics emerged as a solution to improve the energy efficiency of hydraulic systems, reducing the throttling losses and internal leakages presented in servo valves, using simple components like on/off valves and flexible control strategies to better manage the hydraulic energy [1]. Digital hydraulics is divided into two main branches, called fast switched commutation and parallel connection [1]. In [2], a review of digital hydraulics was presented, where some characteristics of different digital hydraulic systems were discussed. It is shown that the parallel connection is becoming the commonly used approach since fast switching hydraulic still requires specific components due to their high switching frequency.

In [3], the digital parallel connection approach is used to combine the flow rate with the cylinder areas to provide velocity control. In [7], the parallel connection is used to combine different pressure levels with the cylinder areas to obtain force control. In both cases, a multi-chamber cylinder with different areas is used to increase the number of output combinations.

In [7] a strategy for defining the cylinder areas and the constant pressure sources in a secondary control digital actuator is presented. The authors point out that is important to predict linear distributions with evenly spaced output values when designing digital hydraulic systems, since this simplifies the control system and promotes smooth actuator movements with an adequate dynamic response.

However, due to design constraints, the number of discrete outputs of the digital hydraulic system is limited, which means that is not possible to meet intermediary values between two consecutive discrete output levels. If an intermediary value is demanded, the control strategy must be able to manage this lack of required output by choosing the best available output e.g. by using a cost function [3]. In this case, the cost function implies that the output can be higher or lower than requested.

3. Digital Hydraulic Actuator Proposed

In this study, the use of the variable rotational speed source coupled to a digital hydraulic pump to supply a symmetric cylinder is proposed, denominated as Variable Speed Digital Electro Hydrostatic Actuator (VSDEHA) (Fig. 3.1). In this configuration, the variable rotational speed digital hydraulic pump is not limited to discrete output flow, the pump is able to provide continuous values of flow rate for each pump unit combination, which increases the number of output flow available to the hydraulic system.

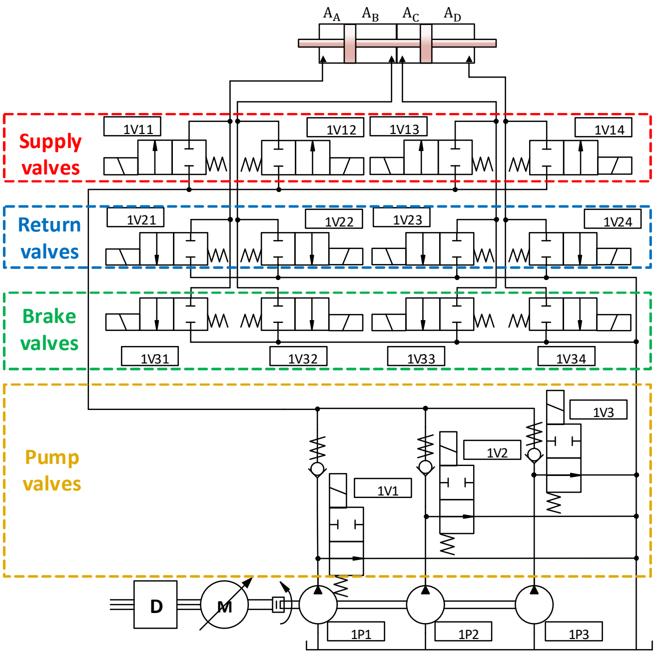

Figure 3.1. Variable Speed Digital Electro Hydrostatic Actuator.

The proposed digital hydraulic pump is composed of three pump units with different volumetric displacement sizes distributed and related to each other by a numerical sequence of power of two ($2^n, n = 1, 2, 3, \ldots$), which increases the number of different flow rates available for each rotational speed **Error! Reference source not found.**. The pumps are controlled by the on/off valves 1V1, 1V2, and 1V3, which direct the flow rate to the system or to the reservoir, independently. The pump combination is defined by the number of pumps connected to the system. In this case, there are seven combinations available for the system. When using a variable rotational speed source there is no need to use a multi-chamber cylinder to obtain different velocities. Nevertheless, the multi-chamber cylinder strategy can be adopted to manage the size constraints and the work pressure levels. The cylinder movement is controlled by on/off valves, where the valves 1V11, 1V12, 1V13, and 1V14 connect the pump with each cylinder chamber A_A, A_B, A_C and A_D, respectively. The valves 1V21, 1V22, 1V23, and 1V24 connect the cylinder chambers with the return line according to the cylinder movement direction.

The third set of valves, called brake valve (BV), is used to connect the cylinder chambers to the reservoir line (1V31, 1V32, 1V33, 1V34). During the development of the solution it was noticed that, in the presence of assistive external loads, the cylinder velocity was reaching values over the maximum designed, compromising the controllability of the system. The brake valves are designed to restrict the flow rate and to limit the maximum cylinder velocity when an assistive load condition is detected. In this case, under normal conditions, the return and brake valves are used to connect the cylinder chambers to the return line. However, when the assistive load is detected, only the brake valves are used.

4. SYSTEMS DESIGN MODELLING

The method proposed by [8] was used to design the system for position control. The system parameters are the cylinder stroke of 0.1 m, a mass of 200 kg, viscous friction of 20 kNm/s, and maximum working pressure of 20 MPa. As system requirements, a step response of 0.05 m, a settling time of 1 s, and a maximum force of 76 kN were used based on **Error! Reference source not found.**. In this context, the results obtained according to the design method were: the maximum cylinder velocity of 0.11 m/s, the maximum flow rate of 6.7×10^{-4} m³/s (40.2 L/min), and the total cylinder area of 0.006 m², which was divided to obtain a four-chamber cylinder with equal areas of 0.003 m².

To supply the maximum flow rate, the volumetric displacement of the pump units was distributed in a power of two sequence with the pump sizes of 6.36×10^{-7}, 1.27×10^{-6}, and 2.54×10^{-6} m³/rad (4, 8, 16 cm³/rev). The maximum rotational speed is 178.02 rad/s (1700 rpm), which is based on a generic four poles induction machine. At this rotational speed, the digital pump theoretically supplies a maximum flow rate of 7.9×10^{-4} m³/s (47.6 l/min). The equation of motion was used to model the symmetrical cylinder as

$$p_A A_A - p_B A_B + p_C A_C - p_D A_D - F_{fri} - F_{load} = M\ddot{x}, \tag{4.1}$$

where p_A, p_B, p_C, and p_D are the pressures in each cylinder chamber [Pa], A_A, A_B, A_C, and A_D are the cylinder chamber areas [m²], F_{fri} is the friction force [N], F_{load} is the load force [N], M is the total mass moved [kg], and $\ddot{x}$ is the acceleration [m/s²].

The dynamic friction force was modeled using the LuGre friction model. The parameters were adapted from [9] and are presented in Table I, where σ_0, σ_1, and σ_2 are the average stiffness coefficient of bristles [N/m], the average damping coefficient of bristles [kg/s], and the viscous coefficient [kg/s], respectively; f_c, f_s, and $v_s$are the Coulomb friction force [N], the stiction friction force [N], and the Stribeck velocity [m/s], respectively. The cylinder chambers were modeled using the Continuity Equation applied to each cylinder chamber, that is

$$q_{v_{in}} - q_{v_{out}} = q_{v_{leak}} + Av + \frac{V}{\beta}\frac{dp}{dt}, \tag{4.2}$$

where $q_{v_{in}}$, $q_{v_{out}}$, and $q_{v_{leak}}$ are the inlet, outlet and leakage flow in each chamber [m³/s], respectively. A is the cylinder area [m²], v is the cylinder velocity [m/s], V is the chamber volume [m³], β is the effective Bulk modulus [Pa], and p is the pressure in the chamber [Pa].

The on/off valves were modeled using the orifice flow equation, being

$$q_v = c_d f \pi d x_{max}\sqrt{\frac{2\Delta p}{\rho}}, \tag{4.3}$$

where c_d is the discharge coefficient, f is the fraction of the perimeter of the spool, d is the spool diameter [m], x_{max} [m] is the maximum spool displacement, Δp is the valve pressure differential [Pa], and ρ is the fluid specific mass [kg/m³].

TABLE I. LuGre friction model parameters adapted from [9].

Parameters	Value
σ_0 [N/m]	5×10^7
σ_1 [kg/s]	0.1
σ_2 [kg/s]	5×10^4
f_c [N]	1510
f_s [N]	1600
v_s [m/s]	0.02

The valve dynamics for the opening and closing movements were modeled by a second-order transfer function [10]. The dynamic parameters of the valve were obtained from [9] and are described in Table II, where ω_{on} and ω_{off} are the valve bandwidths [rad/s] to open and close, respectively; the time delays on and off are the times to charge and discharge the valve solenoids, and ξ is the damping coefficient. The sizes of the valves were adjusted to achieve a better balance between the energy efficiency and controllability.

TABLE III. Valve parameters adapted from [9].

Parameters	Valves			
	Supply	Return	Brake	Pump
ω_{on} and ω_{off} [rad/s]	2000	2000	2000	1000/1500
ξ [-]	0.7	0.7	0.7	0.7
Time delay on [s]	0.01	0.005	0.001	0.01
Time delay off [s]	0.01	0.005	0.001	0.01
x_{max} [m]	0.003	0.0017	0.001	0.001
c_d [-]	0.67	0.67	0.67	0.67
d [m]	0.0014	0.0006	0.00033	0.0046
f [-]	1	1	1	1
ρ [kg/m³]	890	890	890	890

In the digital hydraulic pump, the internal leakages were modeled by a laminar orifice connecting the pump outlet with its inlet. The output flow rate for each pump is determined by

$$q_{vp} = D\omega - k_{vin}\Delta p, \tag{4.4}$$

where D is the pump volumetric displacement [m^3/rad], ω the pump rotational frequency [rad/s], and Δp is the differential pressure [Pa]. A variable leakage coefficient (k_{vin}) [m^3/(sPa)] is calculated in real-time using a lookup table from the manufacturer's datasheet, which calculates the difference between the theoretical and the datasheet flow rate.

The manufacturer datasheet used was the Bosch Rexroth – External gear pump standard performance AZPW [11]. The external gear pump is used due to the ease of mounting several pumps on the same shaft and relative cost-benefit.

In this work, the dynamic behavior of the rotational speed source was considered as a second-order transfer function, where the dynamic parameters based on the typical response of electric motors presented in [12] were used, being the natural frequency of 10.47 rad/s and the damping coefficient of 1. The component models and lines of the hydraulic circuit were implemented using the software Hopsan 2.14.2 [13]. The supply and return lines have a total volume of 0.0015 m^3.

4.1. Controller

The system control was developed in the MatLab/Simulink. Fig. 4.1 shows the system block diagram, where the position of the actuator is controlled by a PI controller. The output of the controller is converted to a required flow information into the pumps and valves selector block, which have the flow, torque and overall efficiency maps of the pumps calculated from the manufacturer's datasheet. The selector block chooses the valves, the pump combination, and the rotational frequency necessary to supply the flow demanded to achieve the position. Since the same flow rate is possible to be achieved by different combinations of active pumps and rotational frequency, the selection is based on the overall efficiency of the pump combination [4].

The valve delay control block is used to synchronize the opening and closing of the valves [10]. In the selector block, a time interval of 0.05 seconds is used by the selector to calculate and send a new command to the system. This time is to ensure that the valves complete their movement before receiving a new command. The rotational frequency command is sent to the rotational speed source. The valve synchronized command and the rotational frequency are the inputs to the digital hydraulic system block, which comprises the hydraulic system implemented in Hopsan.

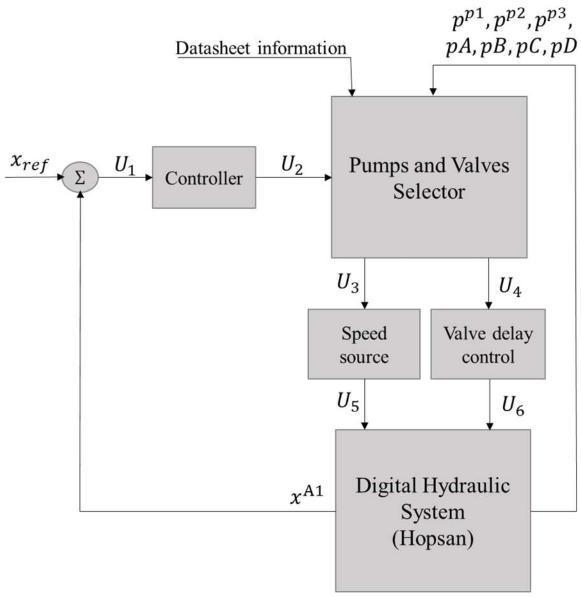

Figure 4.1. System block diagram.

4.2. Servo Hydraulic Actuator

The Servo Hydraulic Actuator (SHA) or Hydraulic Servo Actuator (HSA) is a typical name used on aircraft area to describe systems that have servo valve controlled actuators [14, 15]. The SHA model described in [9] was used and adapted to the same design parameters of the VSDEHA, in order to be used as base line to compare with the proposed solution. The parameters adopted for the servo valve were: flow coefficient k_v of $1.2 \times 10^{-7} \mathrm{m}^3/\mathrm{sPa}^{1/2}$, bandwidth ω_n of $1256.6\ \mathrm{rad/s}$ and dapping factor of 0.9. The servo valve is supplied by a constant pressure source of 20 MPa.

4.3. Energy Evaluation

For the energy analysis, the hydraulic power provided by the sources was considered as input and the hydraulic power consumed by the cylinder as output. The energy input of the SHA and VSDEHA are described, respectively, by

$$E_{in}^{SHA} = \int (q_v^{sv_1} p^{sv_1} + q_v^{sv_2} p^{sv_2})\, dt \tag{4.5}$$

$$E_{in}^{VSDEHA} = \int (q_v^{p_1} \Delta p^{p_1} + q_v^{p_2} \Delta p^{p_2} + q_v^{p_3} \Delta p^{p_3}) dt \tag{4.6}$$

where q_v^{sv} is the flow rate $[\mathrm{m}^3/\mathrm{s}]$ supplied by the pressure source, p^{sv} is the supplied pressure [Pa] at servo valve, q_v^p is the flow rate supplied for each pump unit $[\mathrm{m}^3/\mathrm{s}]$, and Δp^p is the pressure differential in each pump unit [Pa].

The hydraulic energy consumed by the cylinder in both systems is described by

$$E_{out}^{A_1} = \int (p_A A_A - p_B A_B + p_C A_C - p_D A_D) v^{A_1}\, dt, \tag{4.7}$$

where v^{A_1} is the cylinder velocity [m/s].

5. SIMULATION RESULTS

In order to evaluate the system behavior and the energy consumption, a workload profile was proposed using a spring load with a stiffness coefficient of 156 kN/m, which produces a load of 76 kN with a cylinder displacement of 0.05 m. This load profile is used to simulate the aerodynamic load into an aircraft control surface, which increases with the displacement as noticed in [9].

Figure 5.1 shows the reference profile and the simulation responses. The VSDEHA using brake valve was able to follow the reference within the predetermined settling time as the SHA, although subtle differences in behavior were observed in the return movement with assistive load. The first difference occurs between the time of 15 and 20 seconds on the negative step reference, where the SHA shows a better response.

The second difference was observed in the ramp reference between 35 and 36 seconds, where the VSDEHA follows the reference moving in a stair trajectory, effect of the successive switching of the valves to track the reference, with overshooting at some points. In both cases, SHA model has the advantage to manage the movement with assistive load, since it is possible to proportionally graduate the valve orifices, while the VSDEHA, with the proposed brake valve, has a fixed orifice size to control the maximum cylinder velocity.

The control strategy considers a minimum time interval for consecutive opening and closing of the valve. When a change is required within this time interval the system response may be delayed, the overshoot can be considered the result of this delay to close the valves.

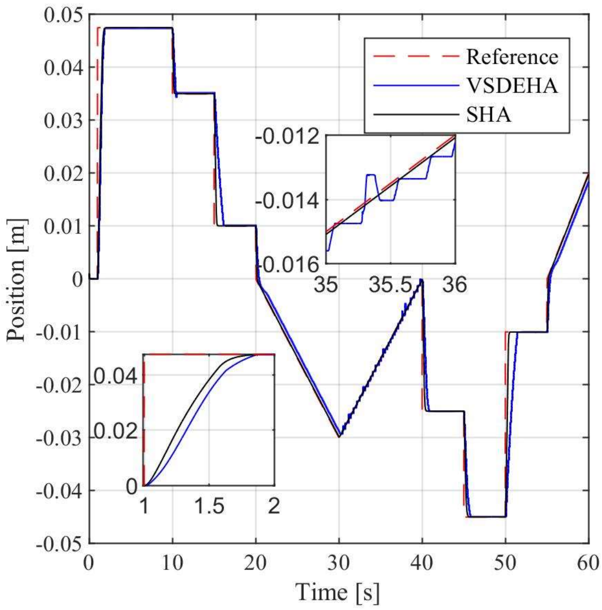

Figure 5.1. Position profile and system response.

Figure 5.2 shows the cylinder velocity behavior for both systems. The cylinder velocity is kept inside the range of maximum velocity (0.11 m/s). The largest difference between the models is noticed in the time interval of 30 and 40 seconds, where the external load effect quickly increases the cylinder velocity making the valve be activated to close and to open it consecutively, tracking the position reference.

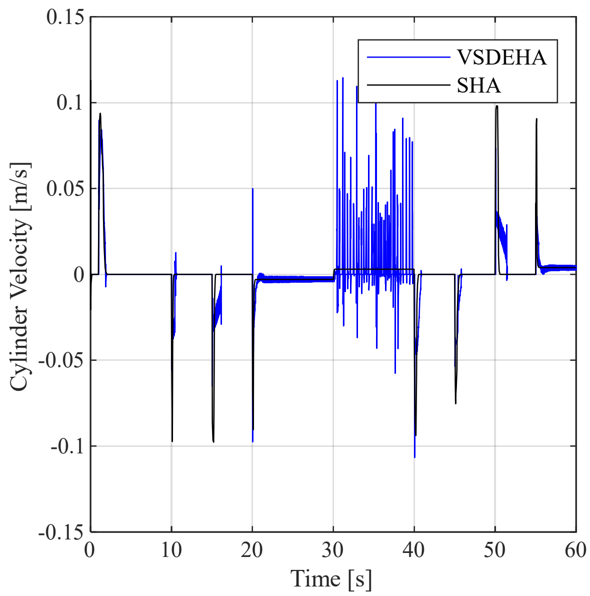

Figure 5.2. Cylinder velocity using SHA and VSDEHA.

Figure 5.3 shows the electric motor speed command and its dynamic response. In the time 1s, for the cylinder to reach the reference position, the pump requires a high rotational speed, which decreases according to the cylinder position error is reduced. At the same time, the pump combination is changed following the strategy to choose the best combination for the required flow rate. In the stand-by condition, the electric motor is kept running at 500 rpm to reduce the system time response and to keep the pump lubrication.

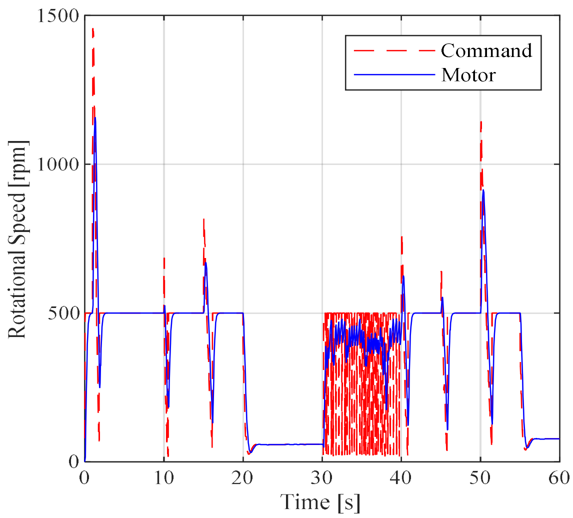

Figure 5.3. Electric motor speed behaviour.

From the electric motor perspective, the peaks presented are a point of attention for the power consumption and the dynamics necessary to perform the rapid variation of rotation speed. Moreover, the efficiency of the electric motor at low rotational frequency and high load have to be observed. In this work, the aspects related to the electric motor are not addressed.

The hydraulic energy of the system was monitored during the work profile and the results are shown in Fig. 5.4**Error! Reference source not found.**, where is presented the input and output energy accumulated.

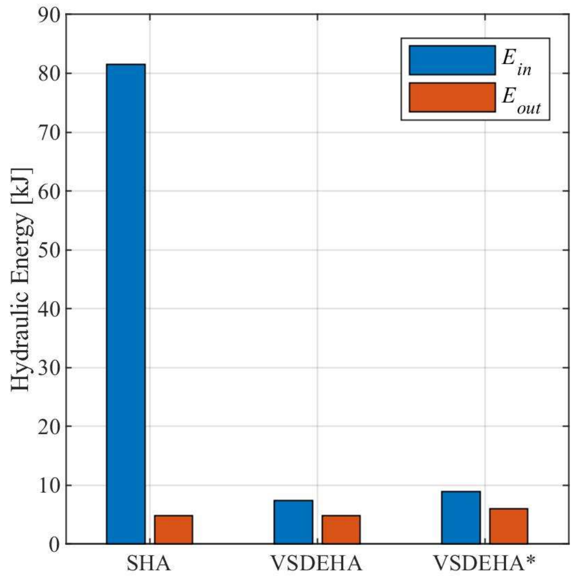

Figure 5.4. Energy input and output. *Without brake valve.

The total input energy of the SHA model was about 80 kJ and its output energy was about 5 kJ, which means that 75 kJ were lost, mainly due to the throttling control of the servo valve. When the system is on standby, the servo valve internal leakage needs to be supplied by the power source. The VSDEHA follows the power-on-demand concept and presented a total input energy of 7 kJ and an output energy of 5 kJ, approximately, which means that the energy dissipated during the work is related to hydraulic losses in the hydraulic circuit. When the cylinder is moving against the external load, there is no throttling control. In standby mode, the digital pump is connected to the reservoir at low pressure differential, avoiding high energy dissipation and the rotational speed is reduced.

A comparative analysis of the system with and without the use of the brake valves is presented in Fig.5.5. The brake valves are used to dissipate part of the external energy in order to limit the maximum cylinder speed and mitigate the overshoots. Without the brake valves, the cylinder exceeds the designed maximum speed, and compromise the controllability of the system, as highlighted in the time interval 9 to 11 and 35 to 36 seconds, where the valves are not able to close fast enough to follow the position reference and the overshoots are more frequent.

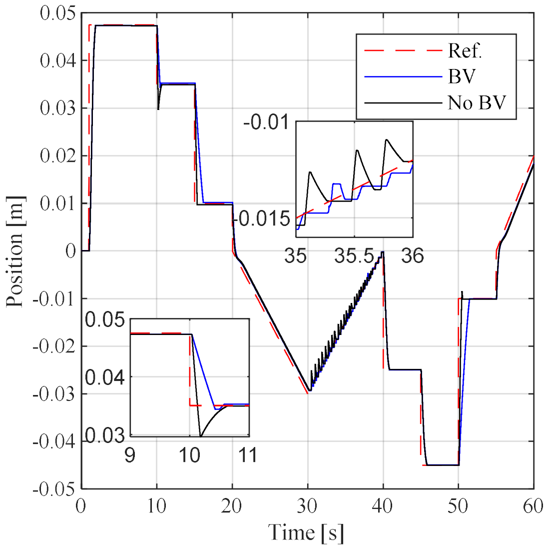

Figure 5.5. Position analysis with and without valve brake.

The total energy input of the system with and without brake valve was 7.3 kJ and 8.9 kJ, respectively (Figure 5.4). This means that, without the brake valves, the system spends 1.6 kJ more energy to follow the position reference.

6. CONCLUSIONS

This study presented a design of a digital hydraulic actuator comprising a symmetrical cylinder controlled by a digital hydraulic pump driven by a variable rotational speed source. The use of a digital hydraulic pump with variable rotational speed is justified due to the possibility to choose the best operational point for each pump combination to maximize the pump energy efficiency and to supply power on demand to the system.

The model is the first approach to evaluate the needs of the proposed topology and some parameters were based on previous studies developed by the authors on the aircraft area. The model presented a high energy-saving potential and a good compromise with the dynamic response of the system.

In digital hydraulics, one of the objectives is to avoid the use of throttling control. Although, it was demonstrated that the use of the brake valves, to dissipate the extra external energy, has improved the controllability without increasing the energy consumption of the system. However, the concept still requires future work evaluations to better manage the external energy and to improve the system controllability.

7. ACKNOWLEDGMENT

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