An Aircraft Actuator Driven by Digital Hydraulic Pumps

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Abstract: Hydraulic systems are known by their high power weight ratio, reliability and good dynamic response. However, they present low energy efficiency normally caused by the main throttle losses. One of the main components of the hydraulic systems are the pumps. These components are responsible to be the flow rate source to the system. For aircraft applications, the most common solutions use variable displacement pumps in order to be able to control the amount of fluid that is delivered to the system depending on the work conditions. For instances, in steady state flight, the pump is set in a low volumetric displacement, only supplying the system leakage and keeping the system pressurized. Nevertheless, in this situation, the pump works in a low range of efficiency, with high output pressure and low flow rate, where the volumetric and mechanic losses increase. In order to reduce the effects of the use variable displacement pumps with low volumetric displacement, in this paper, a configuration using three fixed displacement pumps with different sizes in a digital approaching, on/off valves and a multichamber actuator will be presented. With these pumps, it is possible to obtain 8 different flow rate values, and with the combination of the four chamber actuator areas, 28 velocity values in steady state. With this arrangement, when the actuator is not moving, the actuator's chambers are keep closed and the pumps flow rate are directed to the reservoir with low output pressure, where the volumetric losses are reduced. The preliminary results seem to be promising, where the proposed system consuming around 4 times less energy save when comparing to the conventional solution. With the improvement of the amount of energy consumed by the hydraulic systems, the aircraft can save fuel, which increases its flight autonomy.

Keywords: Digital Hydraulics, Digital Pumps, Hydraulics Systems.

INTRODUCTION

The consumption of nonrenewable fuel sources is one of the main concerns in the XXI century. As mentioned by Lotfalipour *et al.* (2010), the combustion of fossil fuels is the biggest contributor for the emissions of carbon dioxide in the atmosphere. Besides, Davis & Caldeira (2010) quote that the carbon dioxide emissions are the primary cause of the global heating. The transportation sector is one of the large contributor for the pollution of the atmosphere. One of the reasons is the increasing of the automobile population around the world. Tsita & Pilavachi (2017) mention that, with the increasing of the number of cars, consequently the consumption of gasoline will increase, however, their average velocity will reduce. In addition, Neves *et al.* (2017) presented that, in the economics field, the transportation sector is the most backward in the carbon dioxide reduction.

With the increase of the carbon dioxide in the atmosphere, an environment awareness installed in a global level. With that, the requirements for new efficient projects became more rigid. In Xylia & Silveira (2016) is quoted that Sweden established a goal to acquire an independent fossil fuels car fleet until 2030. Another important statistic data is that in 2014, the countries of the European Union consumed 33% of their energy in the transportation sector, 94% of which came from petroleum sources (NEVES *et al.*, 2017).

In the aeronautic field, Gössling (2010) quotes that, the aviation sector is responsible for about 1.6 to 2,2% of the carbon dioxide global emissions, that means about 13% of the total emission of this gas only from transportation sector. The main reason is that the modern aircraft propulsion technology is based in the use of internal combustion engines (SLIWINSKI et al., 2017) due mainly to fossil fuels presenting high energy density.

In aircraft, one of the most important system is the hydraulic system. It is responsible for the actuation of many different functions of an aircraft as the control of the primary and secondary control surfaces (MOIR & SEABRIDGE, 2008), landing gears, cargo doors, steering (WARD, 2017) and so one. The main advantage of the use of hydraulics system instead electric systems, for the actuation of control surfaces, is the low weight/power ration and high dynamics behavior (ALLEYNE & LUI, 2000). However, the hydraulic systems are known by their low efficiency, normally below 50% (DE NEGRI *et al.*, 2014). In this case, the efficiency improvement of the hydraulic system, can cause the reduction of the fuel consumption for an aircraft, even a small improvement can be beneficent. For instance, Roboam (2011) mentions that in 2008 the aerospace industry was responsible by the emission of 2% of the carbon dioxide emitted by

man and every kilogram of unburned fuel per flight saves 1700 tons fuel and 5400 tons of carbon dioxide are not released into the atmosphere per year considering all the aviation industry (ROBOAM, 2011).

In order to achieve the goal to increase the hydraulic system efficiency for an aircraft, the approach of the use digital hydraulics could be a very promising solution. In these solutions, the throttle losses that occurred in conventional hydraulics system can be avoid by the use of on/off valves instead the proportional valves. Besides, there are also solutions using digital pumps, motors, actuator and accumulators in place to traditional hydraulic components. For this paper, a solution using digital pumps, on/off valves and a multichamber actuator will be presented.

AIRCRAFT ACTUATION SYSTEM

In aeronautical systems, control surfaces are extremely important for the aircraft control, even under normal flight conditions or under emergency conditions. Theses surfaces are divided in two main groups, the primary and secondary surfaces (MOIR & SEABRIDGE, 2008). The primary control surfaces are responsible for the main control of the aircraft being elevators, ailerons and rudder (MOIR & SEABRIDGE, 2008). The secondary control surfaces are responsible for the control of the wings lift being flaps, slats and speed breakes (van den BOSSCHE, 2004). Nowadays, there are many different solutions of actuators to control these surfaces, where the most common are called, electric hydromechanic actuators, electro hydrostatic actuators and electro-mechanical actuators (MARÉ, 2017).

For Electro-Mechanical Actuators (EMA), according to Moir & Seabridge (2008), this kind of arrangement replaces the electro-hydraulic actuator with an electric motor coupled in a gearbox. In Electric-Hydraulic Actuators normally is used a hydraulic cylinder connected directly with a proportional servo valve, where the hydraulic power comes from a centralized hydraulic power unit, which supplies energy for all actuators. For the electro hydrostatic actuator (EHA), the most applied solutions use fixed displacement pumps driven by a variable speed electric motor connected in the hydraulic cylinder, or variable displacement pumps with constant speed also connected in the actuator, or variable displacement pump and variable speed electric motor (ALLE, 2012; MARÉ, 2017). Figure 1 shows the schematic examples of the main technologies used to control the control surfaces.

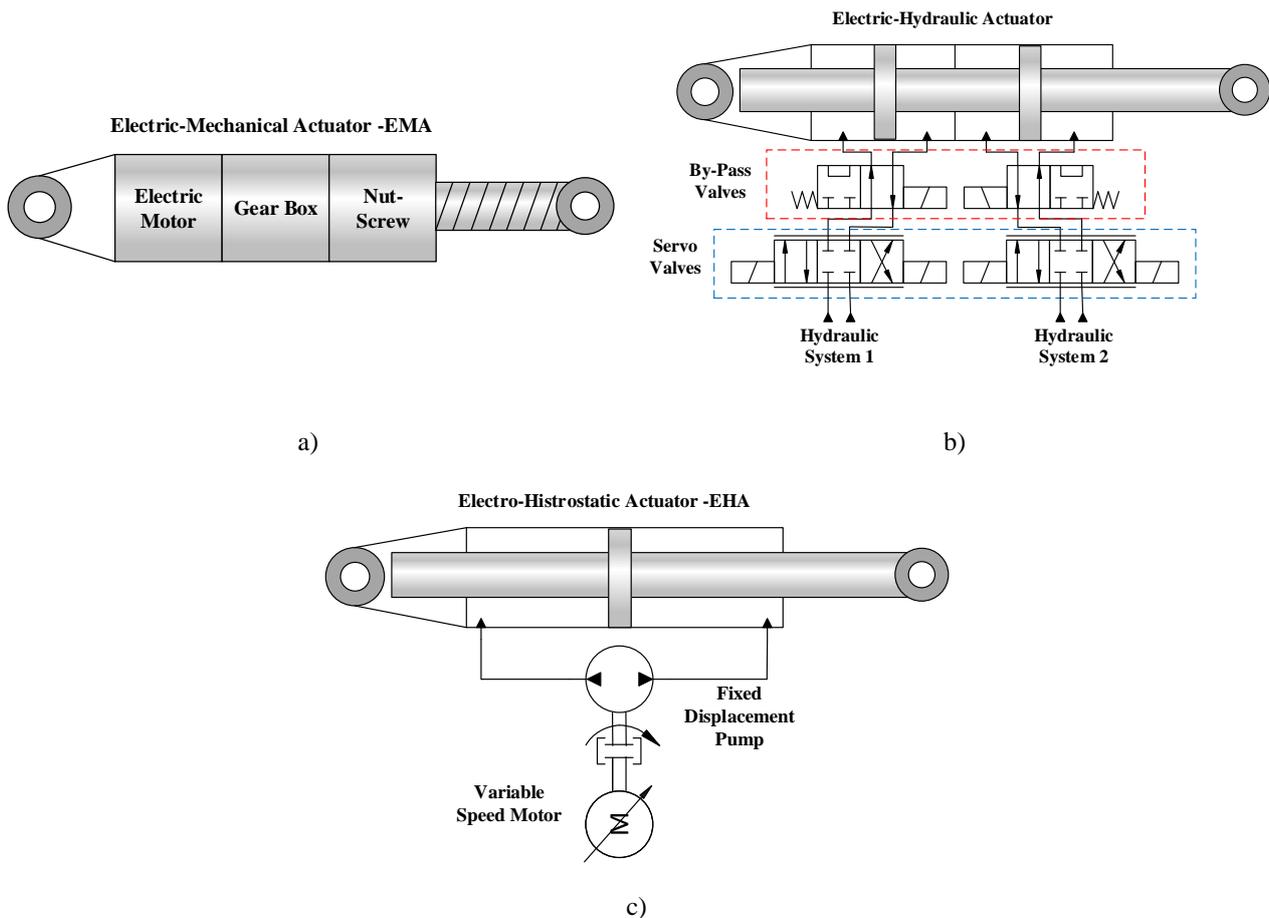a)   b)

c)

Figure 1 – Main Conceptions for Aircraft Control Surfaces: a) Electric-Mechanical Actuator; b) Electric-Hydraulic Actuator; c) Electro-Hydrostatic Actuator. Based in Maré (2017).

Maré (2017) quotes that the main difference between Electro-Hydrostatic Actuators (EHA) and EMA is that in EMAs the transmission between motor and load is basically pure mechanic. Wang et al. (2016) mention that use of the Electric Mechanical Actuator can bring some advantages as:

- Reduce flight control weight by 25%;
- Reduce maintenance by 42%;
- Reduce the mean time to repair by 50%;
- Increase the aircraft availability.

However, the same authors point out the main disadvantages such as:

- The Jamming of the actuator;
- Sizing of the actuator.

According to Wang et al. (2016) due the main disadvantages of the EMAs, this kind of actuators are only used in secondary flight control in the spoiler of the Boeing 787 and the engine thrust reversers in the A380.

For Electric-Hydraulic Actuators. In this kind of solution, the aircraft is equipped with two or more independent hydraulic systems that can supply all the main aircraft actuators. For instance, according to Moir & Seabridge (2008) the Airbus A320 and the Boeing 767 present three independent hydraulic systems. In the case of the A320, this aircraft presents five hydraulic pumps, where the first hydraulic system consists in one Engine Driven Pump – EDP. The second system consists in one Electric Motor Driven Pump EDP and one pump installed together with the Ram Air Turbine – RAT. The last one has one Engine Driven Pump and one Electric Motor Driven Pump (MOIR & SEABRIDGE, 2008). In the Boeing 767 there are eight hydraulic pumps. The first and the second hydraulic systems have Engine Driven Pump and one Demand Electric Motor Driven Pump. The last hydraulic system has two EDP, one Air Driven Pump and one coupled in the Ram Air Turbine (MOIR & SEABRIDGE, 2008). In order to increase the reliability and safe of the aircraft, in solutions that use Electric-Hydraulic Actuators, normally the actuator is installed in a tandem configuration, which means, two actuators connected together in the same body as can be seen in Figure 1b. In this configuration each actuator has one servo valve to control de position of the control surface and one by pass valve, which connects the actuator chambers in a case of fault of the servo valve or the hydraulic system.

In Electro-Hydrostatic Actuators, the hydraulic system works in a closed system using decentralized power units. According to Alle et al. (2016), the Electro-Hydrostatic Actuators present the advantages of reduce the amount of pipe lines and joint thought the aircraft eliminating leakage and problems with pressure drops, since the Electro Hydrostatic Actuator are decentralized units. Besides, the same authors quote that self-contained hydraulic servo actuators are ease to replace in case of a fault. Nowadays, the Electro-Hydrostatic Actuators, are very important components for the concept called More Electric Aircraft – MEA. In this approach according to Belan (2018), due the low efficiency of the hydraulic systems used in aircraft, there is the favoring development of more electric solutions in replace of mechanic, hydraulic and pneumatic systems. Belan (2018) also highlights that the MEA concept considered besides efficiency, aspects such as reliability, weight, power density, design and manufacturing complexity, thermal robustness, maintainability, cost and controllability (CAO, et al., 2012; NAAYAGI, 2013). Nowadays, the Airbus A380 and the Boeing B787 are the maximum expression of the MEA concept, as highlight Boglietti, et al. (2009). With the development of the MEA concept, the Electro-Hydrostatic Actuators became an important solution due the reduction of the centralized hydraulic power units. However, this solution still needs hydraulic power. Belan (2018) quotes that hydraulic power is still preferred solution to control de primary control surfaces due mainly safety and reliability (ROBOAM, 2011; CAO, *et al.*, 2012; MARÉ, 2017)

For cruise flight conditions, the solutions that use variable displacement pumps (Electric Hydromechanic, Electro Hydrostatic Actuators) to control the primary and secondary control surfaces, the hydraulic pump remains most of the time in a low volumetric displacement range, only suppling the system leakage. In this situation, the pump operates in a low efficiency condition where the effects of the low lubrication should be taken in consideration (PINTO et al., 2016).

In order to avoid the pump operation in a low efficiency range, a solution using digital pumps, digital valves and a multichamber actuator is presented in this paper. Besides, the solution proposed is not a completely closed circuit, the proposed is called Digital Electro-Hydrostatic Actuator – DEHA, due the similarity of the operation when compared to EHA.

DIGITAL HYDRAULIC FOR AIRCRAFT

Hydraulic systems are known by their low energy efficiency. This low efficiency is caused due the massive use of resistive components that throttle the flow passages. This characteristic becomes the hydraulic systems with an efficiency normally below 50% (DE NEGRI et al., 2014). For the application in the aircraft industry the hydraulic systems are mainly used to control the primary (elevators, ailerons, rudders) and secondary (flaps, slats, speed brakes) control surfaces. Besides the control of the control surfaces, they are also responsible for e.g. landing gear, brakes, gears steering and cargo doors (WARD, 2017). In order to improve the hydraulic system efficiency, the digital hydraulics emerges as a new solution and some researchers about this new technology are being provide around the world (DE NEGRI et al., 2014; SCHEIDL et al., 2013; HEITZIG et al., 2011; LINJAMA et al., 2009). Nevertheless, the use of digital hydraulics in aircraft is still an embryonic technology being at the beginning of its development. In Belan et al. (2015), a configuration

using a multichamber actuator with 12 on/off valves and three different supply pressure levels for the force control of the aircraft control surfaces is evaluated. In this study, the authors achieved very good results in energy saving, where the proposed solution spends 80% less energy than the conventional system. Ward (2017) evaluated a solution similar to the one elaborated by Belan et al. (2015) applied in a dynamic aircraft model. In Dell'Amico et al. (2018), a hybrid digital-proportional hydraulic actuation system for aircraft flight control is studied. In this work is evaluated the use of the DHA Digital Hydraulic Actuator in some control surfaces and the conventional system with servo valves in the other surfaces.

PROPOSED DIGITAL HYDRAULIC SYSTEM

The proposed digital hydraulic system will be described according to the classification presented in Linsingen (2013) and used by Belan et al. (2014). This classification divides the digital hydraulic systems into four subunits, being Conditioning Unit, Primary Conversion Unit, Secondary Conversion Unit, and Limitation and Control Unit. The primary energy conversion unit consists of three fixed displacement hydraulic pumps which are coupled to the same shaft. The volumetric displacement of each pump (D_z) was selected to form a geometric sequence with a common ratio of 2, given by:

$$D_z = D2^{z-1}; \quad z = 1,2,3, \tag{1}$$

where z is the index related to each pump [1] and D is the smallest pump volumetric displacement [m³/rot]. At the outlet of each pump an on/off valve was added in order to direct the hydraulic fluid to the reservoir or to the system, according to its demand. In addition, in order to prevent not only pressure peaks, but also return flow to the pumps, check valves have also been installed and three relief valves for the system safety. However, when the system is in normal operating condition, the relief valves must always remain closed, because the excess flow of the system is directed to the reservoir by means of the on/off valves. Figure 2 shows the hydraulic circuit of the proposed system.
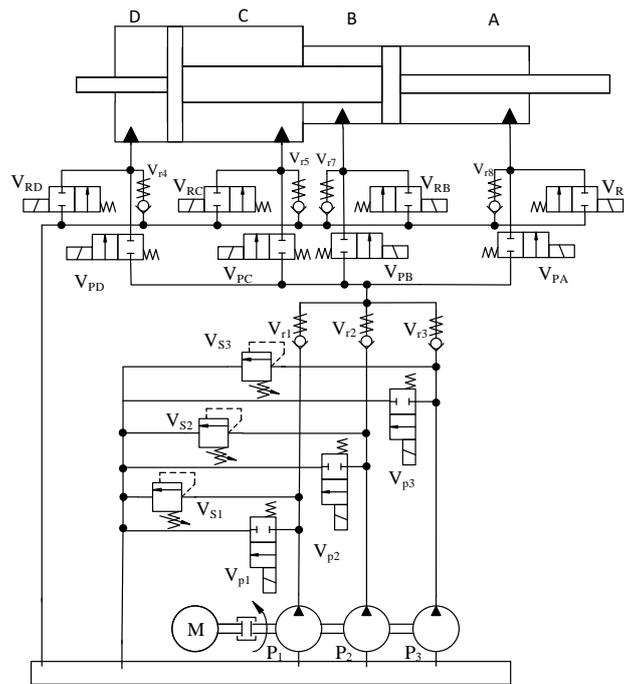

Figure 2 – Proposed digital hydraulic system – DEHA.

In the secondary conversion unit, a multichamber hydraulic cylinder was used with four chambers with distinct areas. In the limitation and control unit, eight on/off valves were used, two for each cylinder chamber and four check valves. These valves are responsible for selecting whether the flow from the digital pumps will be sent to the cylinder chamber or directly to the reservoir, depending on the action of the controller. For the conditioning unit, a pressurized hydraulic fluid reservoir was used. The use of a pressurized reservoir is necessary because of the flight conditions of the aircraft, where their altitude may vary. According to Moir & Seabridge (2008), this action is necessary so that the reservoir is filled completely by hydraulic fluid, which means, without the presence of air, thus improving the volumetric efficiency of the pumps. With the use of three digital pumps, which have independent outputs, it is possible to obtain seven values of discrete flows.

For the system design, it was assumed that the actuator should move the control surface with a maximum velocity of 45°/s, being this value a setting point where the system works in saturation. Considering the size of a real actuator, the

total displacement of the actuator was calculated as 0.075 m, as shows Figure 3. The values obtained where based in a real actuator located at the Division of Fluid and Mechatronic Systems – FLUMES of the Linköping University.
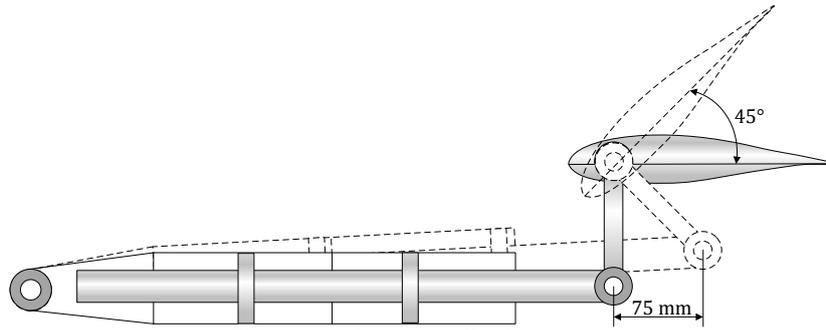

Figure 3 – Actuator displacement.

The choice of multichamber hydraulic cylinder areas was based on that used in Belan et al. (2016). The Table 1 shows the actuator areas.

Table 1 – Actuator areas.

Chamber	Areas
A	$A_A = 1.344 \times 10^{-3}$ m^2
B	$A_B = 0.707 \times 10^{-3}$ m^2
C	$A_C = 1.102 \times 10^{-3}$ m^2
D	$A_D = 1.544 \times 10^{-3}$ m^2

Considering the worst case of the actuator forward movement, the lowest velocity possible, using the two forward chamber areas D and B and assuming a setting time of 1s, the maximum flow rate required for the system considering the saturation is 10,13 L/min. From manufacturer's data catalogs and using a rotational frequency of 1800 rpm (30 Hz), the volumetric displacement chosen for each pump is shown in Table 2.

Table 2 – Pumps volumetric displacements.

Pumps	Volumetric Displacement
Pump 1 (P$_1$)	$D_1 = D = 1 \times 10^{-6}$ m^3/rot
Pump 2 (P$_2$)	$D_2 = 2 \times 10^{-6}$ m^3/rot
Pump 3 (P$_3$)	$D_3 = 4 \times 10^{-6}$ m^3/rot

As mentioned before, with the use of three digital pumps with independent outputs it is possible to obtain seven discrete flows. Considering the values in Table 2 and the rotational frequency of 1800 rpm, Table 3 shows the possible values of discrete flows rates in ascending order.

Table 3 - Available pumps flow rate.

Pumps	Flow Rate
Pump 1 (P$_1$)	1.8 L/min
Pump 2 (P$_2$)	3.6 L/min
Pump 1 (P$_1$) and Pump 2 (P$_2$)	5.4 L/min
Pump 3 (P$_3$)	7.2 L/min
Pump 1 (P$_1$) and Pump 3 (P$_3$)	9.0 L/min
Pump 2 (P$_2$) and Pump 3 (P$_3$)	10.8 L/min
Pump 1 (P$_1$), Pump 2 (P$_2$) and Pump 3 (P$_3$)	12.6 L/min

With the use of the seven discrete flow values in combination with the four chambers multichamber hydraulic actuator, it is possible to obtain 28 discrete velocity values, 14 forward and 14 backward. These values are possible due to the fact that each flow rate value is switched only for one cylinder chamber a time. That means that the remaining chambers remain connected to the reservoir. Thus, the discrete velocities can be calculated as:

$$v_d = \frac{q_{vb}}{A_j}, \tag{2}$$

where q_{vb} is the flow rate referring to the combination of pumps used and A_j the area referring to the cylinder chambers. Figure 4 shows the values of the discrete velocities.
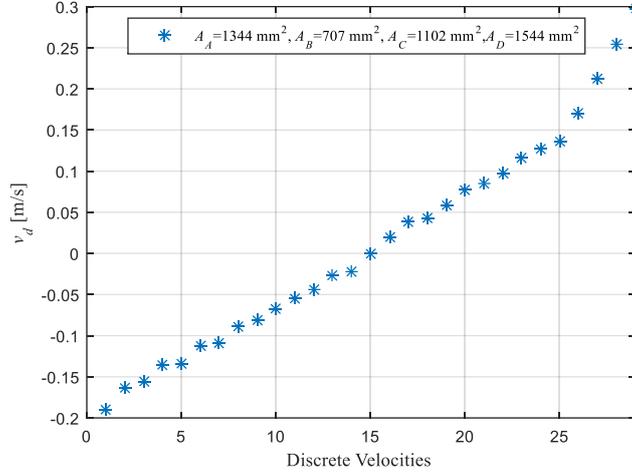

Figure 4 - Distribution of the discrete velocities values.

From the results shown in Figure 4, it is noted that the maximum forward actuator velocity of the cylinder is greater than the maximum backward velocity. However, there is a linearity in some regions of the discrete velocity values.

SYSTEM CONTROL

For the control of the system, its implementation was done in MATLAB/Simulink. For an example of the control system, Figure 5 shows a schematic diagram of the control system.
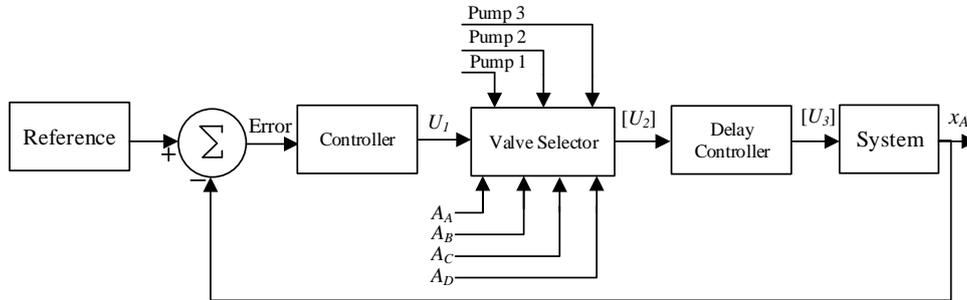

Figure 5 – System control block diagram.

The reference signal x_r, is compared to the actual position signal of the actuator x_A. The error signal is sent to the PI controller, generating the control signal U_1. This signal represents the velocity that the actuator must reach to achieve the reference position x_r. The signal U_1 is then sent to the valve selector, which with the cylinder areas and the pumps combination, selects the valves that must be actuated or not, generating the signal $[U_2]$. The signal $[U_2]$ in turn is sent to the delay controller. After the delay controller, the control signal is then sent to the system valves. Figure 6 shows the valve selector matrix. Each column of the matrix, shown in Figure 6, is responsible for the valves that must be opened and closed to obtain the velocity shown by row two. The first row represents the index of the iteration that the velocity (second row) was calculated. In order to reduce the switching numbers of the valves, a switching cost function was implement. This function is based in the works of Huova et al. (2017) and Belan et al. (2018) and it is given by:

$$c_s = |v_{d(i)} - v_{ref}| + C_f(\Delta v_{dA} + \Delta v_{dB} + \Delta v_{dC} + \Delta v_{dD}), \tag{3}$$

where

$$\Delta v_{dA} = |q_{vbA(i)} - q_{vbA}|/A_A, \tag{4}$$

$$\Delta v_{dB} = |q_{vbB(i)} - q_{vbB}|/A_B, \tag{5}$$

$$\Delta v_{dC} = \left| q_{vbC(i)} - q_{vbC} \right| / A_C, \tag{6}$$

$$\Delta v_{dD} = \left| q_{vbD(i)} - q_{vbA} \right| / A_D, \tag{7}$$

where, v_{ref} is the reference velocity, $v_{d(i)}$ the velocity calculated in each interaction, C_f [1] the cost factor, $q_{vbA(i)}$, $q_{vbB(i)}$, $q_{vbC(i)}$, $q_{vbD(i)}$ the flow rate necessary for each interaction and q_{vbA}, q_{vbB}, q_{vbC}, q_{vbD} the actual flow rate in the actuator chambers. For the valve selector, three time intervals were implemented based on those used in Belan et al. (2018). dT_s is the time interval that the valve selector algorithm should operate according to simulation sample time (0.1 ms). dT_v defines the time that the combination valve matrix should be recalculated and the dT_{delay} defines the minimum time interval for a valve combination change. The rows four, five and six represent the on/off valves of pumps one, two and three, respectively. The subsequent lines represent the valves of the chambers D, C, B, A, the pumps flow rate q_{vb} and actuator chamber area associate to the request velocity, respectively.

Interaction	i	26	22	– – – ·	27
Velocity [m/s]	v_d	-0,1906	-0,1633	– – – ·	0.2970
Switching Cost	c_s	0,0094	0,0367	– – – ·	0.4970
Pumps / Pump 1		0	1	– – – ·	0
Pump 2		0	0	– – – ·	0
Pump 3		0	0	– – – ·	0
Chamber D / V_{PD}		0	0	– – – ·	0
V_{RD}		1	1	– – – ·	1
Chamber C / V_{PC}		1	1	– – – ·	0
V_{RC}		0	0	– – – ·	1
Chamber B / V_{PB}		0	0	– – – ·	1
V_{RB}		1	1	– – – ·	0
Chamber A / V_{PA}		0	0	– – – ·	0
V_{RA}		1	1	– – – ·	1
Flow Rate [m³/s]	q_{vb}	0.00021	0.00018	– – – ·	0.00021
Area j [m²]	A_j	0.00112	0.00112	– – – ·	0.000707

16 x 29

Figure 6 - Valve selector matrix.

Figure 7 - Delay controller (Adapted from Montavani et al., 2018).

The delay controller, as previously mentioned, applies a delay to the opening signal of each on/off valve. As the response time of commercially valves are different, due to factors such as manufacturing process and wear, the application of the delay controller is necessary to avoid short circuit between high and low pressure lines. The Figure 7 shows the behavior of the delay controller.

The d_{off} is the time delay that the valve spool takes to initiate its moment when the control signal is turned off and the d_{on} is the time delay that the valve takes to react to the opening signal. The t_{dc} is the time of the valve spool dynamics that can be different for opening (t_{dc_on}) and closing (t_{dc_off}) and the t_{dp} is the time to the pressure dynamics (MONTAVANI et al., 2018). The Delay time that should be applied in the valves can be expressed by.
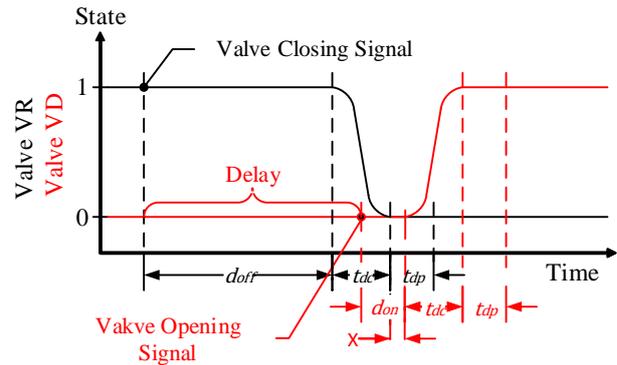
$$t_{delay} = d_{off} + t_{dc\_off} - d_{on}, \tag{8}$$

and the minimum time between two different change of valve combination can be expressed by.

$$dT_{delay} = t_{delay} + t_{dc\_on} + t_{dp}. \tag{9}$$

SYSTEM SIMULATION

The system modeling development previously was performed through the Hopsan 2.9.0 dynamic system simulation software. Figure 8 shows the layout elaborated in the Hopsan software, where each set of valves belonging to each chamber of the multichamber actuator and also the three digital pumps are highlighted.
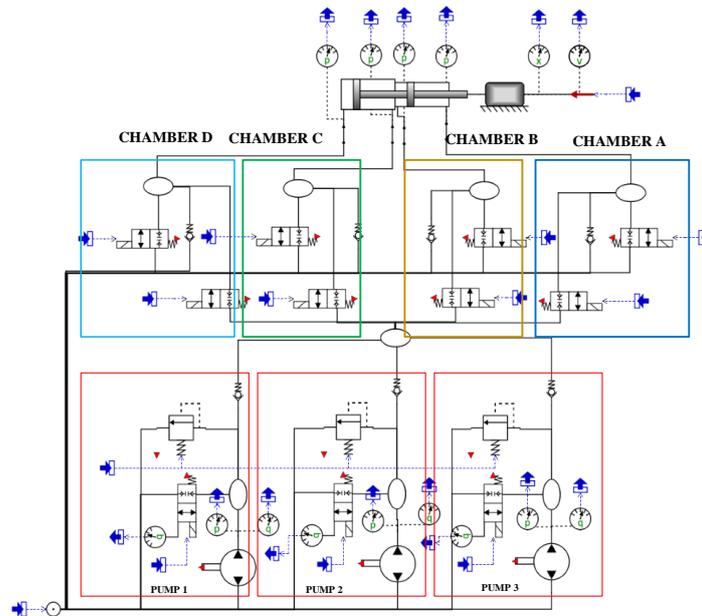
Figure 8 – Hopsan system model.

For the modeling of the system, the reservoir pressure was assumed constant with a value of 7.5 bar (0.75 MPa). This consideration is necessary to avoid cavitation in the suction of the digital pumps and actuator chambers, which are in an expanding movement. Figure 9 shows the parameters for the on/off valves and the actuator.

Symbol	Variable	Unit	Value
	ω_{n_on}	rad/s	400
	ω_{n_off}	rad/s	200
	d_{on}	s	0.003
	d_{off}	s	0.030
	ξ	1	0.9
	c_{p_on}	1	0.64
	c_{p_off}	1	0.65
	ρ	kg/m^3	850
	d	m	0.0023
	x_{vmax}	m	0.001
	f	1	1

Symbol	Variable	Unit	Value
	A_D	m^2	1.544×10^{-3}
	A_C	m^2	1.102×10^{-3}
	A_B	m^2	0.707×10^{-3}
	A_A	m^2	1.344×10^{-3}
	s_l	m	0.2
	β	Pa	1.3×10^9
	$V_{O_{A,B,C,D}}$	m^3	7×10^{-5}, 6.8×10^{-5}, 6.9×10^{-5}, 6.8×10^{-5}
	B	Ns/m	0.1
	k_{vLeak}	m^3/(sPa$^{\frac{1}{2}}$)	1×10^{-16}

Figure 9 – On/Off valves and actuator parameters.

In addition to the digital system model, a model using the conventional control technique with proportional servo valve and pump with variable displacement was also developed. This model was developed in order to compare its efficiency with the proposed digital system. For its elaboration, the same multichamber digital hydraulic actuator was used, being submitted to the same loading conditions. The proportional valve used is four-way with three positions, with its center closed. Proportional valves, used in aircraft, are usually sliding spool servo proportional. This kind of valves have an internal leakage, which comes from the internal clearances between the sleeve and the spool. The proportional servo valve chosen was MOOG, series 760, that provides a flow rate of 18.9 L/min (5 gpm) at a pressure drop of 69 bar (6.9 MPa), and 6 ms time response. According to catalog data, this valve has a leakage of 1.9 L/min in the center position and a pilot leakage of 0.98 L/min at 207 bar (20.7 MPa), approximately. However, experimental results demonstrated that the real internal leakage is different from the catalog data. In order to simulate the real valve leakage in the model, Cruz (2018) developed a valve model considering its internal geometry. Figure 10 shows the conventional proportional system used to compere the efficiency with the digital system and Figure 11 demonstrates the valve leakage model compared with the experimental results.
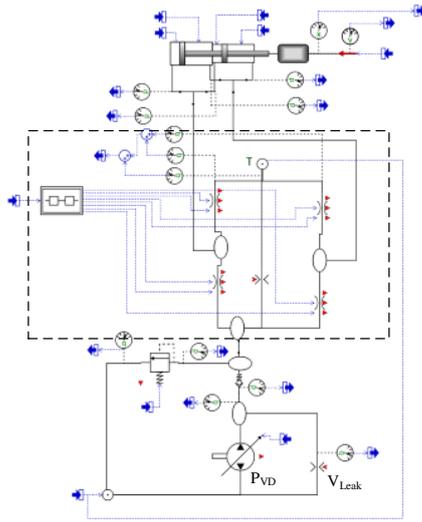
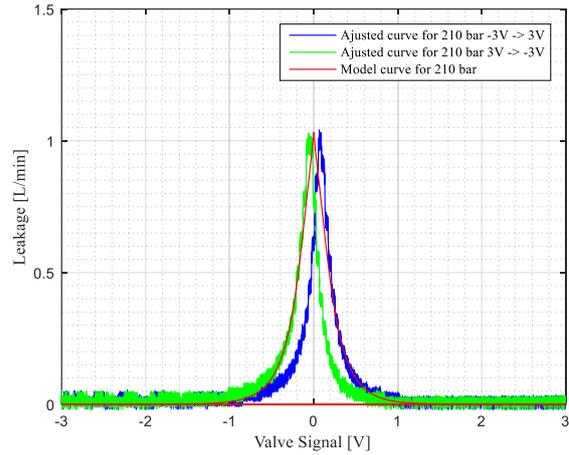**Figure 10 - Conventional proportional system with servo valve (Based in the model proposed by Cruz, 2018).**

Figure 11 - Servo valve internal leakage (Based in the model proposed by Cruz, 2018).

According to the results obtained from Figure 11, the valve has an internal leakage, without considering the pilot leakage, around 1 L/min with a pressure drop of 210 bar. Cruz (2018) quotes that the pilot leakage is around 0.8 L/min. Thus, the total leakage is 1.8 L/min. This value represents the total flow rate that the pumps needs to supply in order to keep the system pressurized. The volumetric efficiency of the variable displacement pumps is dependent on output pressure and volumetric displacement, as shown in Bravo (2017). For this work, the results presented in Bravo (2017) were used as a base for the calculus of the pump leakage. The pump leakage was modeled as a restriction valve (V_{Leak}) connected directly to the reservoir. The valve leak flow coefficient was obtained with the pump operating in the worst condition, which means, with low displacement. For aircraft applications, this is the case the pumps remain most of the time due the steady state flights. According to the results presented in Bravo (2017) and using a pump displacement equals to 20%, the pump volumetric efficiency can be expressed by

$$\eta = -9.10^{-9}p_{vp} + 0.9287, \tag{10}$$

where p_{vp} is the output pump pressure. Considering the system work pressure as 25 MPa (250 bar), the pump volumetric efficiency is equals to around 68%. To supply the system leakage and the internal pump leakage (considering the pump efficiency), the minimum pump output flow rate is 2.37 L/min. Thus, the V_{Leak} flow coefficient is 4.58x10^{-13} m^3/(sPa). The volumetric displacement of the pump was modeled with two linear functions, according to the actuator position error. When the actuator is not moving, with zero error, the volumetric displacement is selected to a sufficient level only to supply the internal leakage of the system, which in the case studied, is equal to 2.37 L/min, representing the sum of pilot leakage plus leakage due to valve clearances and the losses in the hydraulic pump. When the position error is maximum positive or negative, the pump has its maximum volumetric displacement. Equation (11) shows the implementation of the variation of the volumetric displacement of the pump as a function of the error (e)

$$\begin{cases} if\ 0\ \leq\ e \leq 0.075\ \rightarrow D = 10.821e + 0.1884 \\ if-0.075\ \leq\ e < 0\ \rightarrow D = -10.821e + 0.1884. \end{cases} \tag{11}$$

SIMULATION RESULTS

In this section, the preliminary results obtained will be presented. For the simulations, the valve delay time t_{delay} was set to 58.4 ms, dT_{delay} is 74,4 ms, dT_v is 6x10^5, C_f is 100, the relief valve opening pressure in 27.5 MPa (275 bar) and the reservoir pressure is 0.75 MPa (7.5 bar). The proportional and integral gains for the PI controller are 1 and 0.01 for the digital system and 10 and 0.1 conventional system, respectively. In addition, the controller output signal of the digital is multiplied for a gain of 3.9603 to convert the signal in velocity and for 13.333 in the conventional system in order to open completely the valve for a position error of 0.075 m. The load force in the actuator is 10000 N against the forward actuator movement and the system equivalent mass is 47 kg. The friction force was modelled according model present in the Hopsan software called *Translational Mass with Coulomb Friction*, being the viscous friction coefficient equals to 40000 Ns/m, the static friction force 1450 N and the kinetic friction force 900 N (BELAN, 2018). Figure 12a shows the position response for the digital and conventional system and Figure 12b shows the velocity control signal for the digital system. Figure 13a presents the output pressure of the pumps and Figure 13b shows the pressure in the actuator's chambers. Figure 14 shows the signal for the on/off valves.

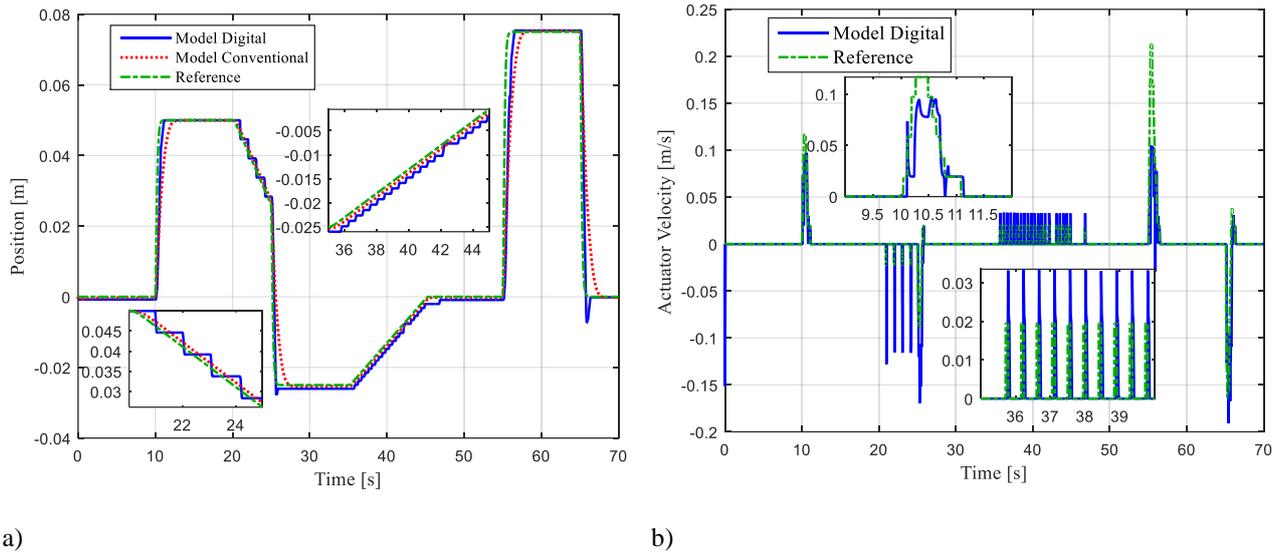**Figure 12 – System response: a) Position; b) Velocity.**
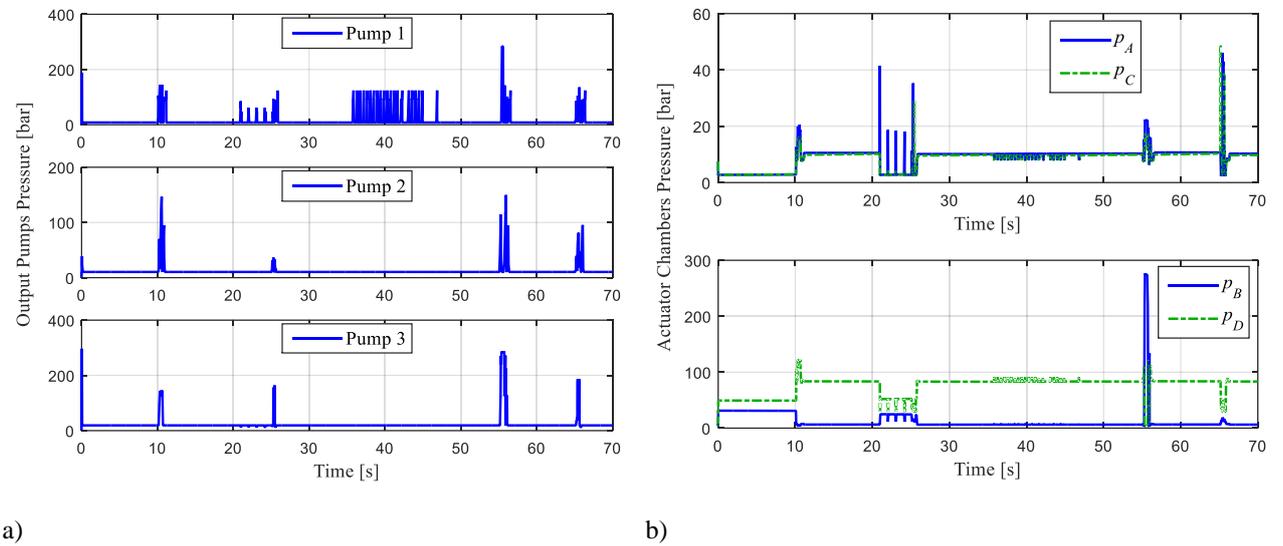

Figure 13 – System response: a) Output pumps pressure; b) Actuator chamber pressure.
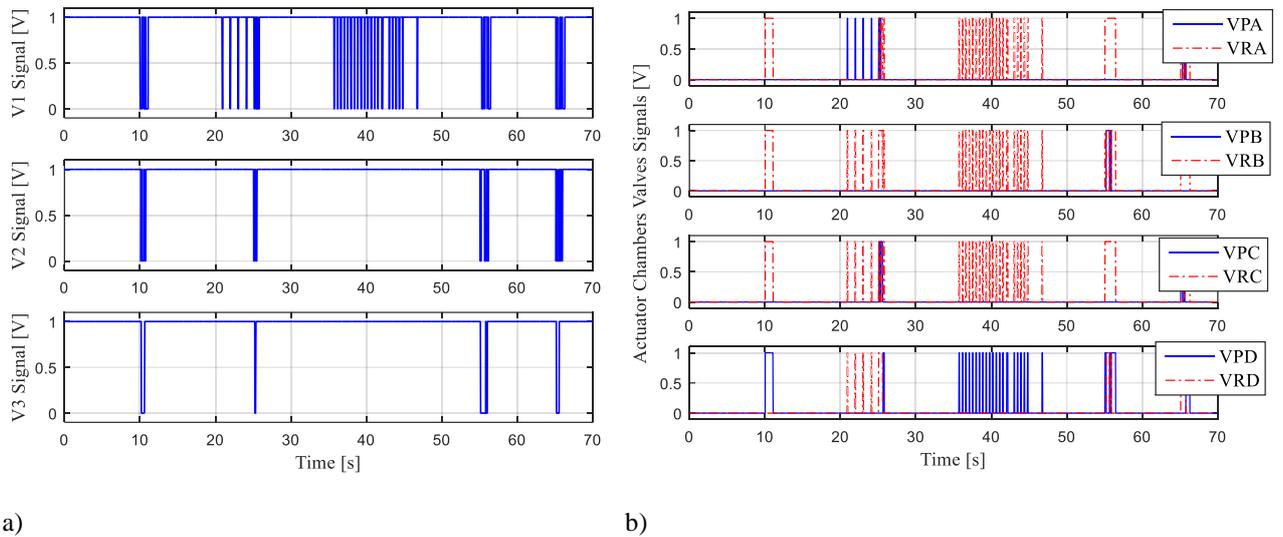

Figure 14 – System response: a) Pumps valves signal; b) Actuator valves signal.

The results presented in Figure 12 show that the digital system can follow the reference signal with a good accuracy as well as the proportional system. When the actuator is in the forward movement (Figure 12a at 35s to 45s) and in backward movement (Figure 12a at 20s to 25s) with constant velocity, the digital system is able to follow the reference. However, the digital system doesn't have enough resolution. In these situations, only the smaller pump is used with the bigger actuator areas that produce the smaller velocities.

The pressures in the actuator chambers and the output of the pumps exhibit oscillatory behavior. This fact is caused by the fast switching of the on/off valves between low and high pressure lines. In some moments, the pressure in the chambers have a value close to zero, indicating that could occur cavitation in the system. This effect happens when the digital pumps are suppling fluid to a chamber that is increasing the volume, the other chamber that also has an increasing volume remains connect to the reservoir. However, the pressurized reservoir is not able to supplies enough fluid to keep the pressure value.

The energy supplied from the digital system (Equation (12)) was calculated as the integral in time of the hydraulic power supplied to the system, being q_{vP1}, q_{vP2} and q_{vP3} the flow rate for the digital pumps 1, 2 and 3, respectively and p_{P1}, p_{P2} and p_{P3} the output pressure for the pumps 1, 2, and 3, respectively. For the conventional system, the energy supplied (Equation (13)) was calculated in a similar way to the digital system, integrating in time, where, in this case, the power supplied is the sum of the system flow hate (q_{vPs}) and the pump leakage (q_{vpLeak}) multiplied by output pump pressure (p_{vp}).

$$E_{SD} = \int_0^t q_{vP1}p_{P1} + q_{vP2}p_{P2} + q_{vP3}p_{P3}\,dt, \qquad (12) \qquad\qquad E_{SP} = \int_0^t (q_{vPs} + q_{vPLeak})p_{vp}\,dt. \qquad (13)$$

The Figure 15a shows the results for the conventional and the digital systems. As can be seen the digital system consumes less energy than the proportional system. This is due to the digital system, when the actuator is without any movement, the flow rate of the digital pumps are directly to the reservoir, avoiding that the supply flow to be directed to the relief valve. In the conventional system, for the actuator to remain its position, the servo proportional valve is kept closed and the pump has its volumetric displacement reduced, suppling only the system leakage, as can be seen in Figure 15b, but this process happens at high pressure level. For this reason, energy dissipation becomes higher due to restrictive characteristics of the proportional servo valve and the pump, preventing the passage of hydraulic fluid.
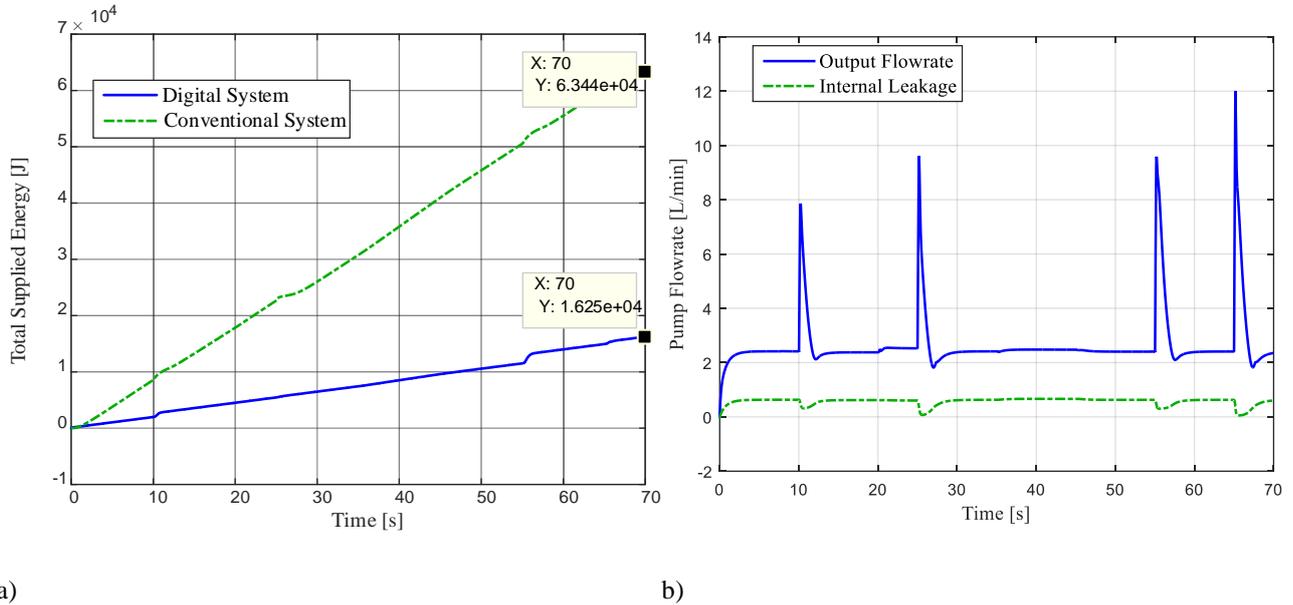

a)   b)

Figure 15 – System response: a) Total supplied energy; b) Variable displacement pump flowrate.

CONCLUSIONS

In this paper, a system using three digital pumps with a fixed volumetric displacement and a multichamber actuator was presented. Also a conventional system using a variable displacement pump was shown in order to compare its performance with the digital system. The results demonstrate that the digital system is able to control the position with good response and accuracy, even for a ramp input signal. However, some pressure oscillations and peaks occur. This behavior is caused by the fast switching of the digital valves between the high and low pressure lines. The main point of the digital hydraulics systems is to avoid the throttle losses that occur in conventional proportional servo hydraulic systems. As shown in Figure 15a, the digital system consumes around four times less energy than proportion system. For aircraft applications, the digital system can reduce the energy consumption of the aircraft because the hydraulic system is less required. In addition, other components of the hydraulic system can be reduced as the hydraulic fluid cooling system.

However, further studies should be conducted to verify the feasibility of installing this type of conception in a real aircraft, such as the construction of a test bench.

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