Development of a hydraulic supply system with pressurized reservoir for a digital hydraulic actuator

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Abstract

The aviation sector actively pursues enhanced energy efficiency and performance, as these improvements trigger a cascading effect that iteratively reduces system weight and size, ultimately lowering fuel consumption, a priority for minimizing operational costs and addressing environmental concerns. In flight control surface actuation systems, where hydraulics is widely employed for its maturity and reliability, such advancements may translate to reductions in hydraulic pumps and conditioning components, fewer and smaller hydraulic lines (and the fluid mass they contain), smaller heat exchangers, reduced fuel reserves required for thermal management, and minimized leakage risks at connection points. In this context, digital hydraulics has been in development in recent years as a promising approach to significantly improve the energy efficiency of flight control systems while maintaining the reliability critical for the application. This paper presents a hydraulic supply system capable of efficiently delivering three different constant-pressure lines, which can support digital hydraulic actuators under development for aviation and other high-performance systems. Since the technology is intended for aviation, the presented architecture includes a pressurized reservoir obtained through a combination of actuators connected to the reservoir and pump outlet. The proposed supply matches pressure reducing and relief valves together with a pressure-compensated pump and three accumulators to generate and maintain the demanded pressure lines. Via simulations on the Hopsan software, the solution is tested alongside with the digital hydraulic actuator for step and sine position inputs. The pressure lines show little fluctuation and the actuator performs as expected, which indicates that the proposed supply is capable of providing the necessary conditions of operation. The simulated system as a whole (supply and actuator) achieves efficiencies of up to 59%, far greater than the obtained by traditional servo hydraulics, and cavitation is avoided in all times.

Keywords: digital hydraulics, supply system, energy efficiency, flight control systems, pressurized reservoir.

1 Introduction

In the recent decades, the growing concern over the environmental situation has affected how engineering decisions are taken. This shift pushes attention to solutions that present distinguished energy efficiency, not only across entire machines, but also within each of the systems that comprise them.

The transport sector relies on oil for more than 95% of its energy needs, which leads to a consumption of about 60% of the worldwide petroleum supply as fuel [1]. Aviation has grown faster than any other transport mode. However, due to the time taken for new technologies to penetrate the sector, efficiency may lag up to 20 years. [1]

According to ICAO [2], international aviation by itself contributes around 2% of the global CO_2 emissions and 12% of the total transport emissions. Nevertheless, the emissions from aviation are projected to increase by two- to fourfold by 2050. In this scenario, the sector could be responsible for 4.1 to 11.3% of the total carbon budget set by the Paris Agreement to limit global warming to 1.5°C [2]. Therefore, the industry must not only switch to sustainable aviation fuels, however also increase the efficiency of all onboard systems. [2]

Hydraulics is one of the systems heavily applied in aviation. This technology is most often responsible for braking systems, cargo doors, wheel steering, landing gear movement and flight control surfaces operation [3]. According to [4], hydraulic systems are constantly engaged in critical tasks, especially in the context of flight control surfaces, due to their proven reliability, high stiffness and precise control.

Hydraulic systems applied in aircraft are conventionally based on servo-hydraulic actuators (SHA). Such technology is mature and trusted, however it can be improved in energy efficiency. This energy efficiency is defined mostly by throttle losses and internal leakage in the components [4]. The authors in [5] simulated the hydraulics in flight control under commonly used maneuvers in commercial flights and found that the efficiency of a conventional hydraulic positioning system in such cases can be lower than 3%, depending on the mission.

An alternative to improve the energy efficiency is the use of a Digital Hydraulic Actuator (DHA) presented in [6] [7]. In this conception, the servo valves are replaced by a set of on/off valves, which can select between different pressure lines for each chamber of a hydraulic cylinder. In this configuration, the system responds similarly to servo-hydraulics while reducing the energy dissipation on the valves, minimizing the throttling and internal leakage. The combination of the different pressures with the actuator areas allows a spectrum of forces with p^A discrete force values, where p is the number of pressures and A is the number of actuator chambers. In the specify configuration shown in [6], the four actuator chambers and three pressure lines results on 81 discrete force possibilities. Furthermore, the DHA offers enhanced reliability and potentially lower production costs, as it replaces a single complex component (servovalve) with multiple simpler ones operating in parallel. Although this increases the total number of valves, it improves fault tolerance and each unit is significantly simpler and cheaper to manufacture. For a comprehensive discussion of additional aspects and advantages of the DHA, readers are referred to [5], [6] and [7].

One of the challenges for the implementation of digital hydraulic actuators, such as the one presented in Figure 1, is the hydraulic supply system. It must provide multiple constant pressure values while maintaining energy efficiency. In [6], the author proposes a configuration where the high-pressure line is connected to one of the aircraft's centralized hydraulic supplies, the medium-pressure line to another, and uses the return to reservoir as the low-pressure line. However, this approach as an impact on reliability, as the supply lines are no longer segregated and, therefore, a failure in one of the systems could propagate to all. Furthermore, the return line could introduce issues regarding cavitation and fluid availability at the pumps if used as a pressure line.

In [7], the authors suggest that the supply system for DHA could be composed with digital pumps, as they improve efficiency. Such technology is presented in [8] and [9] with different approaches. For both cases, the control systems involved are velocity controlled and, therefore, have a strong reliance in flow rate regulation. Since the DHA is force controlled, as approached in [6], the solution presented would not be ideal.

This paper presents the development of a hydraulic supply power unit for the DHA presented in [6], while ensuring segregation and without compromising energy efficiency. Maintaining the pressures constant is critical, as large variations affect the force spectrum. In addition, the system must be capable of pressurizing the reservoir line in order to avoid cavitation in the pump, as the proposed application is aircraft. In this case, the actuation system requires three different pressures and, therefore, the supply was developed considering lines for high-pressure, medium-pressure and low-pressure; and a reservoir line.

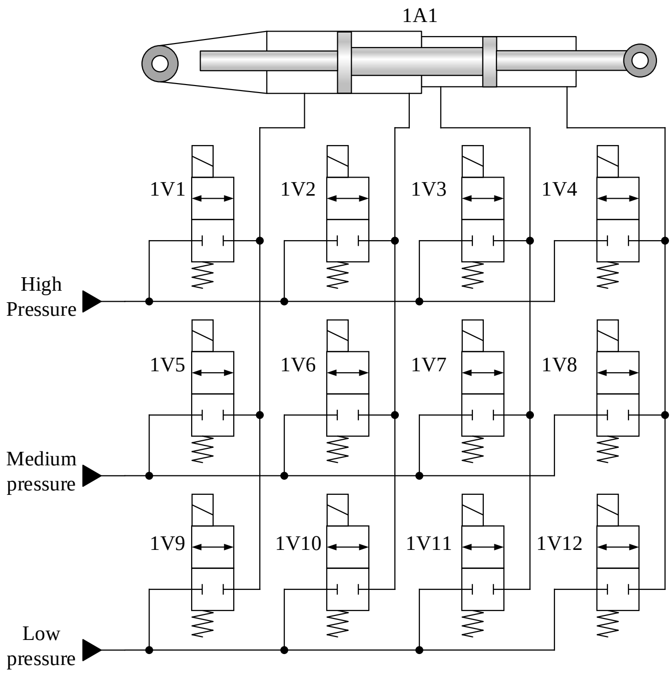

Figure 1: DHA in details with its pressure lines (Based in [6]).

This paper is divided into five sections. Section 2 presents the supply system layout and the function of each selected component. Section 3 describes the efficiency approach and equations. Section 4 shows the simulation model, the system parameter definition and the results obtained. Finally, Section 5 brings conclusions and, briefly, further advances using such technology.

2 System layout

2.1 Pressure supply system

In order to achieve a hydraulic source with multiple pressure supply lines the following principles were considered. The high-pressure line is directly connected to the pump outlet. A pressure-compensated variable displacement pump is used, in which the outlet pressure is applied to regulate the pump volumetric displacement. When perfectly adjusted, the flow rate provided will match the system needs. The power consumption of the pump is, therefore, minimized whenever possible, which increases the systems energy efficiency [10] [11].

For the medium and low-pressure lines an arrangement of pressure valves and accumulator is assumed: A pressure reducing valve regulates the minimum pressure by diverting fluid from a higher-pressure line, while a relief valve controls the maximum pressure by returning fluid to the reservoir line. This concept results on a pressure hysteresis, allowing small pressure fluctuations, without compromising the force spectrum. The hydraulic diagram of the complete supply system can be visualized in Figure 2.

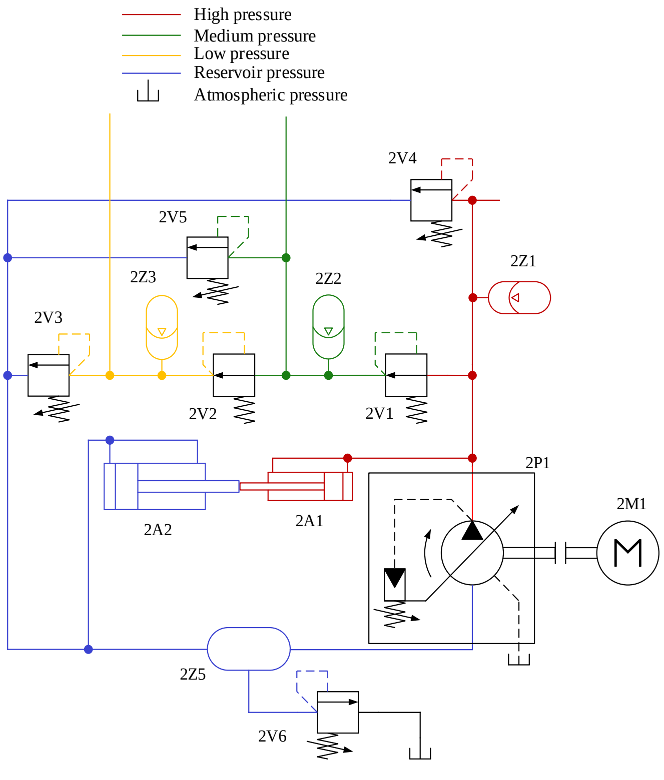

Figure 2: Supply system with pressurized reservoir.

In aviation, the return line is part of a more complex hydraulic system, may exceed hundreds of meters and handles the influence of multiple actuators. Given that, it may be poorly suited to operate as a pressured supply line, since it is susceptible to pressure fluctuations and cavitation. Further research is needed to analyze this use possibility and, if possible, provide the required adaptations.

Accumulators are included in all three pressure lines to attenuate pressure oscillations and store hydraulic energy within each line specified pressure range. In conventional systems, fluid from the actuator retracting chamber is directed to the reservoir making its energy unavailable for further use. This configuration prioritizes energy dissipation rather than storage. Conversely, the proposed approach maximizes energy use by directing any incoming flow (whether from the actuator retracting chamber or the pump) to the highest-pressure accumulator capable of accommodating it within its pressure range before discharges to the reservoir.

Also, if needed, additional pressure lines can be incorporated by applying the same valve scheme present in the medium and low-pressure lines and defining a new range.

2.2 Pressurized reservoir

In aircraft, since the vertical orientation of the systems change, the reservoir line must always be pressurized in order to avoid cavitation in the pump inlet and to ensure fluid supply disregarding position.

It is known that the reservoir must absorb fluid volume fluctuations within the system. For instance, when actuator movement reduces the amount of fluid in the pressure lines, the excess must be stored in the reservoir until the system requires it again. A single accumulator will not meet this function effectively, as its pressure varies greatly with volume changes [12].

The solution designed for this requirement is analogous to a bootstrap reservoir, using two actuators with different resultant areas mechanically coupled by their rods. The high-pressure line connects to the chambers of the smaller area actuator. The chambers of the second actuator are connected to, and thus pressurize, the reservoir line. The pressure ratio between the hydraulic lines is inversely proportional to the resultant area ratio of the two actuators. The arrangement can be observed in Figure 3.

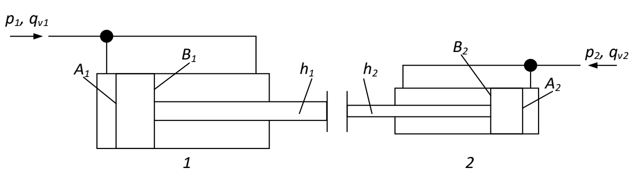

Figure 3: Detailed model of the actuator assembly in the pressurized reservoir.

In the implemented system, an area ten times smaller on the high-pressure side resulted in a pressure ten times smaller on the reservoir side.

Since the actuators are mechanically connected, there is a pressure relation between lines that can be represented as eq. (1), where p_1 is the reservoir pressure, p_2 is the high pressure and k is the area ratio between the larger area and the smaller area:

$$p_1 = \frac{p_2}{k} \ , \tag{1}$$

where

$$k = \frac{A_1 - B_1}{A_2 - B_2} \,. \tag{2}$$

As previously mentioned, one of the key functions of the system is to compensate volume variations, which is achieved through the movement of the piston rods. When fluid enters the return line, the rods consequently shift to the right, accommodating the variation. Conversely, when more fluid is in the pressure lines, the rods move to the left. This dynamic adjustment ensures proper reservoir volume while using the pump's outlet pressure to maintain reservoir pressure.

Additionally, the atmospheric pressure reservoirs shown in Figure 2 are intended for the pump drain and an emergency over-pressurization of the pressurized reservoir. The amount of fluid expected through these directions is, therefore, minimal. Such reservoir could be comprised of a small canister, to be emptied only in scheduled maintenance or following an emergency event.

3 Efficiency

3.1 Energy supplied by the pump

To calculate the system efficiency, the energy delivered to the system by the pump is first defined. The equation for the hydraulic power supplied by the pump is presented in eq. (3):

$$P_h = q_v.\Delta p \,, \tag{3}$$

where q_v [m³/s] is the pump flow rate and Δp [Pa] the pressure differential between the outlet and the inlet. The relationship between energy E [J] and power P [W] is given as shown in eq. (4):

$$P = \frac{dE}{dt} \,. \tag{4}$$

By substituting eq. (3) into eq. (4) the hydraulic energy E_h [J] supplied by the pump is obtained:

$$E_h = \int_{t_i}^{t_f} q_v.\Delta p \, dt \,. \tag{5}$$

3.2 Energy delivered by the actuator

The hydraulic power delivered by the actuator $P_{h,A}$ [W] is given by eq. (6):

$$P_{h,A} = F_h.v, \tag{6}$$

where F_h [N] is the hydraulic force applied by the actuator and v [m/s] is the velocity. By substituting eq (6) into eq. (4), eq, (7) is obtained, which expresses the hydraulic energy consumed by the actuator $E_{h,A}$ [J]:

$$E_{h,A} = \int_{t_i}^{t_f} F_h.v \, dt \,. \tag{7}$$

3.3 System efficiency

The efficiency η of a hydraulic system is given by the ratio of the energy supplied by the pump to the energy delivered by the actuator, as shown in eq. (8):

$$\eta = \frac{E_{h,A}}{E_h} \,. \tag{8}$$

4 Simulation

To ensure the proposed system works as expected and evaluate its dynamics and energy aspects, a simulation model of the scheme, shown in Figure 4, was implemented using the Hopsan® software, an open-source multi-domain system simulation tool developed at the division of Fluid and Mechatronic Systems at Linköping University. Also, this practice allowed for a finer tunning and testing of component parameter prior to a future test bench assembly.

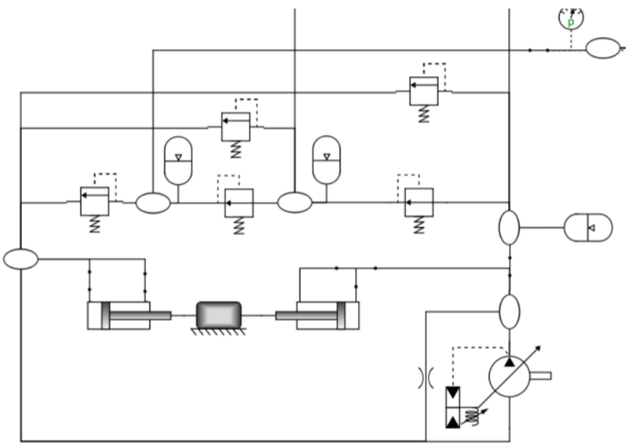

Figure 4: Hopsan simulation model.

4.1 System parameters

As mentioned before, the required pressure ratio (k), in eq. (2), is around 10. Which means that the pressure $p_1$is 10 times smaller than p_2. To achieve that, the actuators were selected based on components available in the market. The result was the cylinder 2A1 with 63 mm in diameter and piston rod $h_1$with 45 mm in diameter, and the cylinder 2A2 with 32 mm in diameter, with 14 mm in h_2 diameter. Beyond that, the cylinders are implemented as C-type pistons, with a 0.4 m stroke and 7.27 × 10^{-5} m^3 of dead volume in all chambers. Default values are assigned for other parameters.

The DHA is simulated based in [6], with the aim of evaluating the proposed supply system performance operating with the digital hydraulic actuator. It presents a multi-chamber actuator with areas A_A 13.44 × 10^{-4} m^2, A_B 7.07 ×10^{-4} m^2, A_C 11.02 × 10^{-4} m^2, and A_D 15.07 × 10^{-4} m^2. The stroke is 0.2 m. The actuator is operated by the set of 12 on/off valves presented in [6], providing 81 possible force outputs. Following the approach detailed in the prior publication, the controller employs a cost function to select the valve combination providing the nearest feasible force output relative to the system demand.

Additionally, the system proposed by [6] is supplied by a high-pressure line of 75 bar, a medium-pressure line of 45 bar and 7.5 bar low-pressure line that is also the reservoir line. In the present model, as aforementioned, the reservoir line is separated from the low-pressure line. Thus, a fourth line is added and the low-pressure line is set to 10 bar. The 7.5 bar line simulated is the reservoir one.

To achieve these pressures, as mentioned in Section 2.1, the pressure relief and pressure reducing valves are adjusted to cause pressure hysteresis. The values of 43 to 45 bar are set in the medium line and 8 to 12 bar in the low pressure. The upper limits are set to the reference pressures of the hydraulic pressure relief valves in the simulation. Such components are parametrized with 10mm of spool diameter, 18 kN/m of spring constant, and a flow coefficient of 0.7. The lower pressure limits are set to the reference pressure of the hydraulic pressure reducing valves in Hopsan. A nominal flow rate of 0.001 m^3/s at the reference pressure are assigned to the valves. Other parameters are left as default.

Accumulators are simulated as gas accumulators, all with 0.001 m^3 as volume and a polytropic index of 1.4. Preload pressures of 72 bar, 40 bar, and 8 bar are assigned to the high, medium and low pressure accumulators, respectively.

The high pressure is maintained by the pressure-compensated variable displacement pump, simulated as a Q-type pump with pressure control in Hopsan. A reference pressure difference of 75 bar, a rotational frequency of 1800 rpm, and a time response of 0.15 s from min to full displacement and of 0.12 s in the opposite direction are parametrized to the pump. The minimal flow rate is set to 0 and the nominal flow rate is set to give 23 Lpm (0.00038 m^3/s) at maximum displacement, 13 cm^3/rev, running at 1800 rpm, which is able of maintaining the maximum speed required by the actuator of 0.15 m/s.

Although other pump control strategies can be used, in this paper will only focus on providing and maintaining the constant pressure levels as originally required by the hydraulic system in [6]. In the future, different strategies could be implemented and optimized to further develop the total system's efficiency, performance, or functionality.

4.2 Step input responses

Considering the hydraulic supply system (Figure 2) coupled to the digital hydraulic actuator (Figure 1), results for a sequence of step inputs are discussed. As shown in Figure 5, a reference step input is applied at every five seconds. The reference signal is shown in blue, the actuator position in red and in yellow the response of an equivalent SHA system (dimensioned for the same use case) for comparison.

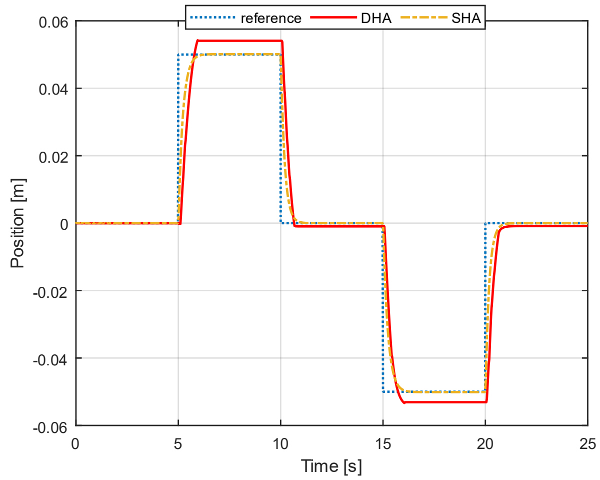

Figure 5: Actuator position for step input.

Figure 6 shows the pressure behavior in each of the supply lines for the same simulation.

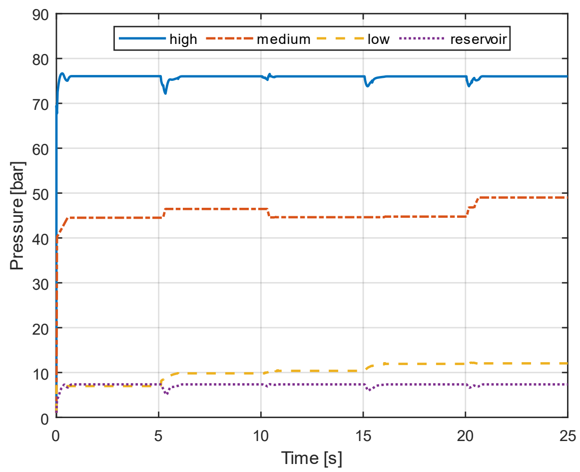

Figure 6: Pressure behavior for step input.

As it can be seen, at five-second intervals, the actuator movement causes a slight pressure drop in the high-pressure line as the actuator demands fluid supply. The fluctuation, however, is less than 5 %, demonstrating the system ability to maintain the defined pressure limits.

Additionally, at 5 and 20 seconds, the medium-pressure line exhibits an increase in pressure value. This fluctuation indicates that the line receive fluid from the actuator, indicating that the flow (and, consequently, the energy) is recovered at the intermediary pressure. A similar behavior occurs in the low-pressure line. The reservoir pressure line follows the high-pressure behavior due to the pressurizing device shown in Figure 3. Figure 7 shows the piston rod displacement in the bootstrap reservoir.

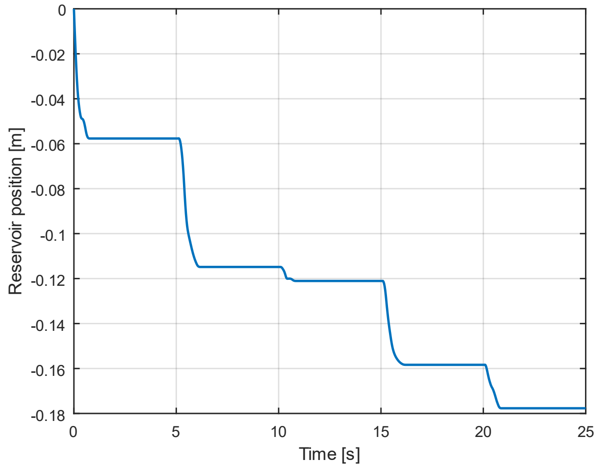

Figure 7: Reservoir position for step input.

It is noteworthy that, as the DHA moves, the reservoir cylinder rod displaces to the left (negative displacement). Therefore, the 2A2 actuator rod occupy part of the internal volume of the actuator, meaning that the reservoir is reducing the fluid volume. Consequently, the pressure lines are increasing their fluid volume. As shown in Figure 6, the medium- and low-pressure lines experience a pressure increasing during the DHA movement and, consequently, store fluid in the supply system. Figure 8 provides a detailed view of the fluid volume stored in the accumulators.

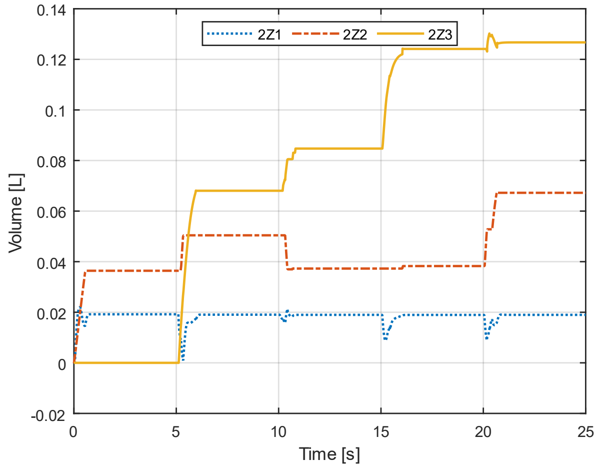

Figure 8: Fluid volume stored in the accumulators.

Figure 9 show the displacement for the sequence of step inputs (parametrization shown in section 4.1). As shown in Figure 9, the pump changes its displacement to provide fluid to the system when the pressure in the high-pressure line reduces, that is, when the DHA moves.

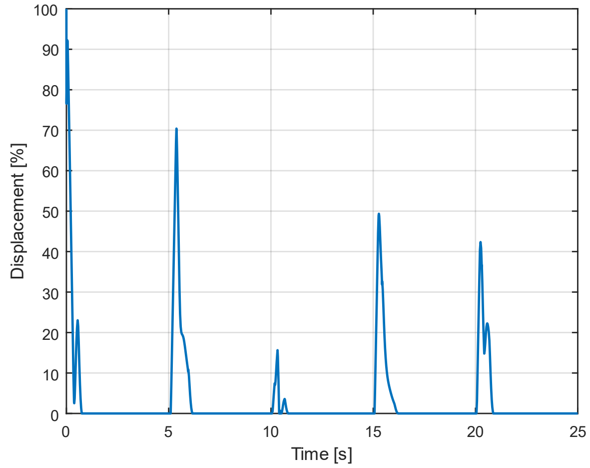

Figure 9: Pump displacement for the step inputs.

The Figure 10 shows the flow rates through the pressure valves. The valve labels are identified in in Figure 2. The valves stay closed practically the whole time except for the relief valve of the low-pressure line, as this line receives enough fluid for its pressure to reach the set pressure and it opens to reduce the pressure.

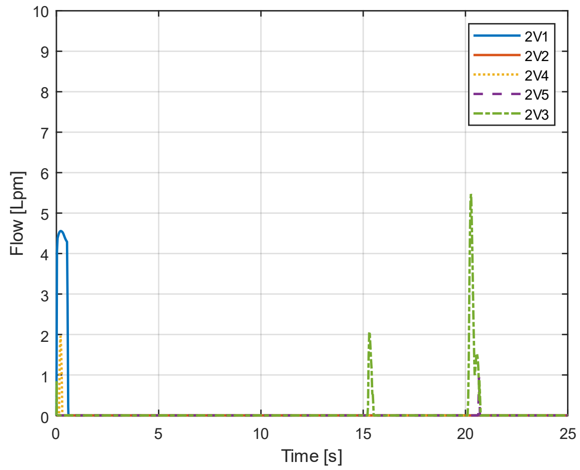

Figure 10: Valve flow for the step input

The variables used for efficiency calculations as shown in Section 3 are extracted from the simulation. In the simulated scenery, the pump provided 2295 J of energy to the system. The DHA consumed around 628 J, which results in around 27% of efficiency. Figure 11 shows the energy supplied by the pump and consumed by the actuator.

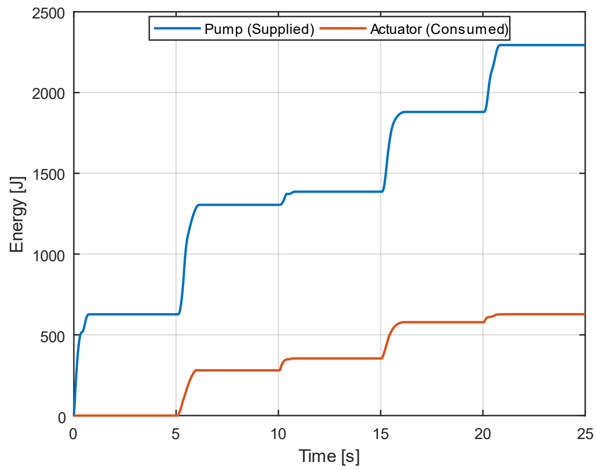

Figure 11: Energy Supplied by the Pump and Consumed by the Actuator for step input.

4.3 Sinusoidal input responses

Results for a sinusoidal input for the actuator positioning system are discussed in this section. The sine input has a frequency of 0.2 Hz and amplitude of 0.09 m. Figure 12 shows, in blue, the reference signal, in red, the actuator position and in yellow the response of an equivalent SHA system for comparison:

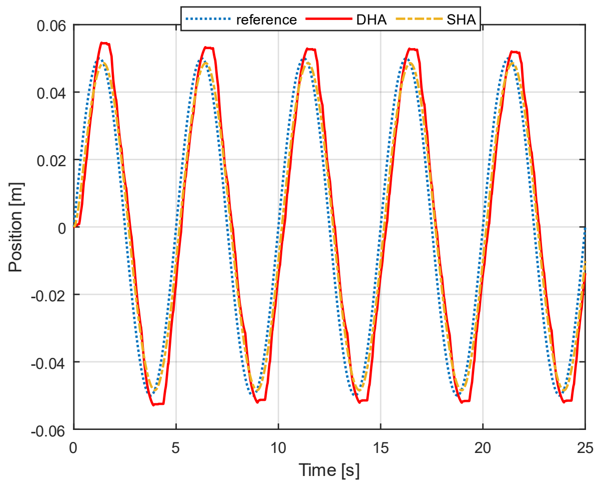

Figure 12: Actuator position for sinusoidal input.

The proposed supply system does not affect the system ability to follow the reference, as the simulation result is similar as the ones shown in [7].

Figure 13 shows the pressure behavior for the sinusoidal input. As can be seen, the pressure varies in all four pressure lines throughout the simulation. This behavior is expected, as the actuator continuous movement at different velocities enables the system to optimize the fluid utilization. To enhance efficiency, the system dynamically switches the pressure lines connected to each actuator chamber and, consequently, the lines constantly receive and supply fluid.

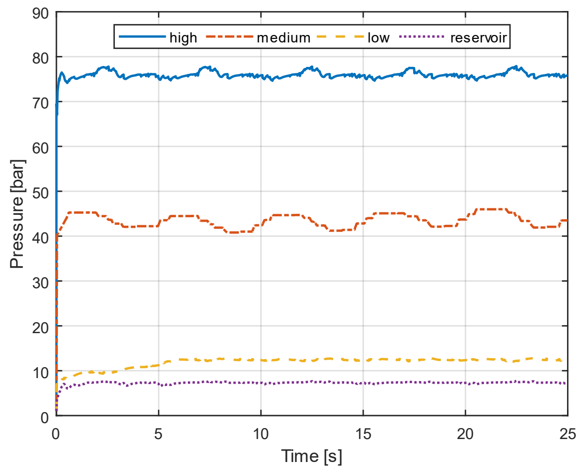

Figure 13: Pressure behavior for sinusoidal input.

Figure 14 shows the position of the bootstrap reservoir cylinder. Similar to what was observed in Figure 7, Figure 14 shows that, as the system pressurizes the lines, the reservoir is displaced to the left, reducing its fluid volume and supplying it to the system. However, it can be noted that after 5 seconds, the system begins to reach a steady-state condition. As seen in Figure 13, the pressures oscillate within a defined range. Consequently, the fluid volume stored in the accumulators does not change considerably, instead of that, it fluctuates along with the pressure oscillations. Figure 15 provides a detailed view of this behavior.

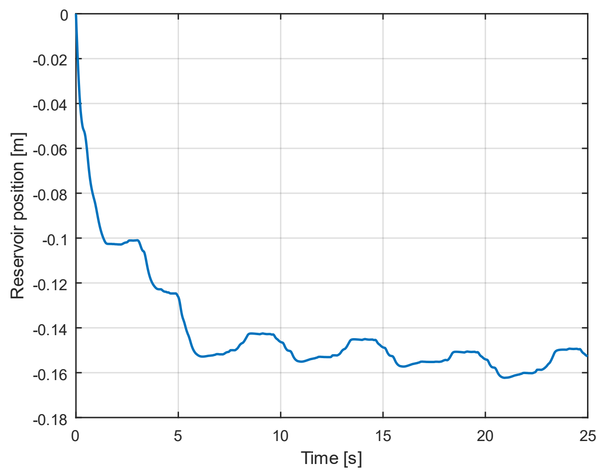

Figure 14: Reservoir position for sinusoidal input.

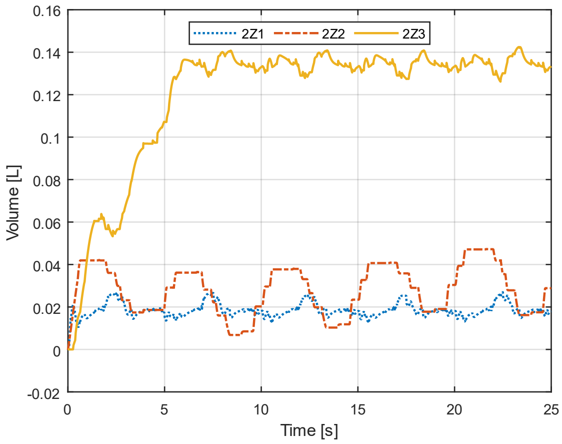

Figure 15: Fluid volume stored in the accumulators for sinusoidal input.

Figure 16 shows the pump displacement for the sinusoidal input.

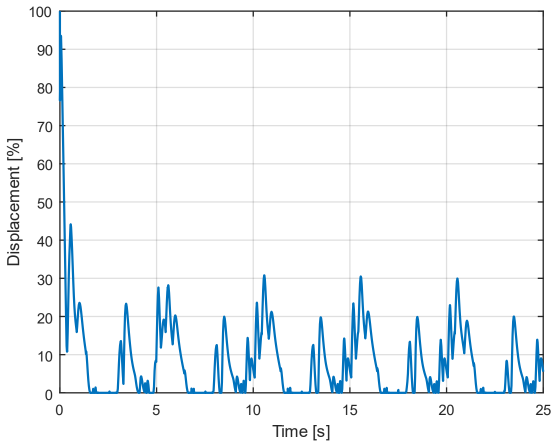

Figure 16: Pump displacement for the sinusoidal input.

As can be seen in Figure 16, the pump is constantly changing its displacement in response to the DHA movement. Since the pressure in the high-pressure line is constantly changing, the pump is constantly compensating the pressure drops.

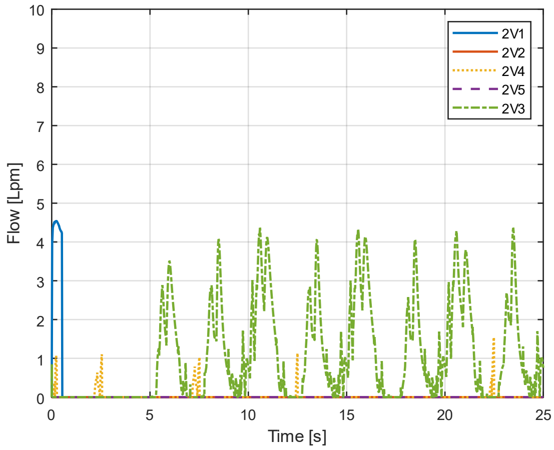

Figure 17: Valve flow for the sinusoidal input.

As it is shown in Figure 17, in the sinusoidal input the valves also stay closed the whole time, except for the relief valve of the low-pressure line. As the DHA is constantly discharging fluid to this line, its pressure increases due to the increase volume in the accumulator, so the relief valve is constantly opening to maintain the pressure in this line.

As previously discussed, the system was designed with pressure hysteresis in each line to facilitate fluid to be moving between the pressure lines rather than discharging to the reservoir. For the sine input the pump provided 5280 J and the DHA consumed around 3100 J. In this case, the system achieved an energy efficiency of approximately 59%. The energy supplied by the pump and consumed by the actuator is shown in Figure 18:

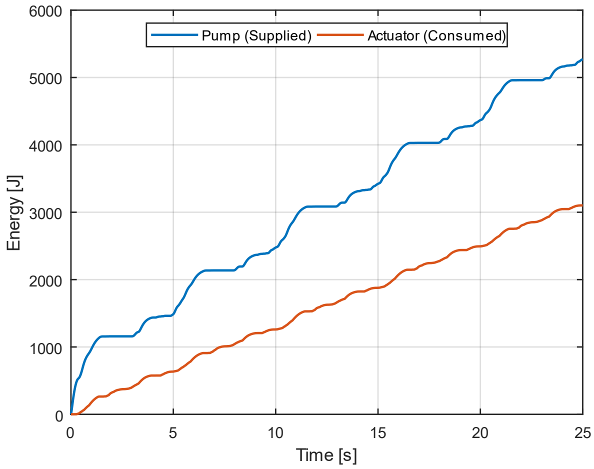

Figure 18: Energy supplied by the pump and consumed by the actuator.

The difference in energy efficiency from the previous case can be explained by the fact that in the sine input, the actuator is constantly moving, thus, the system is constantly using the energy that was provided.

5 Conclusion

Based on the simulation results, the proposed multiple pressure supply system successfully achieves its intended purpose of supplying hydraulic power to the DHA without compromising its performance and maintaining smooth operation. This facilitates the application of digital hydraulics with reliable power supplies, especially with mobile and aeronautical applications as the system is simple and lean.

Additionally, the system kept the reservoir line pressurized and avoided cavitation at all times. This prevents potential damage to hydraulic components, disregarding the orientation of the vehicle, and ensures fluid supply whenever the pump's displacement needed is higher.

Beyond that, the system also maintained the pressure within the designated range for all pressure lines. This validates its ability to handle pressure fluctuations and ensure fluid distribution when needed.

The overall efficiency of the system was significantly higher than conventional hydraulics applied in aviation (as seen in [5], that can be as low as 2%) in all cases simulated, achieving up to 59% as seen in section 4.3. This confirms the efficiency benefits shown by DHA when the supply scheme is also considered, and infatuates the promising alternative that digital hydraulic is for flight control actuation.

The DHA concept and its Hopsan model have been experimentally validated in prior work [6]. The development of this novel supply system proposed in this study is scheduled for experimental validation, with a dedicated test bench currently under assembly to evaluate its functionality and efficiency under real-world operating conditions.

Acknowledgement

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