# Energy Management of Hydraulic Flight Control Actuation Systems using Multi-Mode Control

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## Abstract

This paper presents an analysis of a multi-mode control hydraulic flight control actuation system. The multi-mode control enables the actuator to be active in different modes and thereby reduces the required pump flow. The hydraulic supply system can be down-sized saving weight, space, and power consumption. This directly translates to aircraft fuel savings. The multi-mode control actuator is applied to an electro-hydraulic local supply system where the pumps are driven from electric motors instead of the engine mounted gearbox. An additional mode setting is the pump pressure setting. The work compares the multi-mode system to a traditional hydraulic flight control system. A simulation framework is developed that allows to analyze the required pump flow for various flight conditions and maneuvers. The information is used to estimate the total weight of both systems. A strategy is proposed how to integrate the system simulation information into an aircraft design framework that includes the systems architecture effects on aircraft sizing. Although the multi-mode system requires more components, the results show that the weight of both systems are similar. This is thanks to a 60% pump flow savings using the mode shifting strategy.

**Keywords**: Hydraulic actuator, energy management, flight control

## 1 Introduction

This paper investigates a multi-mode hydraulic flight control actuation system compared to a traditional system based on a proposed methodology. A traditional hydraulic flight control system is based on a centralized redundant architecture with gearbox driven pumps supplying oil at constant pressure. Here the flow requirements are derived from the maximum actuator load and speed. However, since the maximum load and speed do not occur at the same time, introducing mode-control can downsize the pump flow requirements and save system weight and size. The maximum load varies with the flight conditions, varying the system pressure can save on the power delivered by the pumps in certain flight conditions.

The multimode system is combined with an electrified local supply system. Electrifying the hydraulic supply system gives benefits such as increased efficiency and a higher degree of flexibility in terms of control, packaging, and integration with other electrical systems. Electrical systems are in general heavier than their hydraulic counterparts and more components are needed. The introduction of mode-control can overcome these drawbacks to some extent.

Mode shifting actuators is uncommon in aircraft applications. A mode shifting actuator was developed and successfully tested in the demonstrator aircraft YF-23 developed by Northrop/McDonnell Douglas, with hydraulic flow savings up to 60%, [1]. This actuator uses a tandem configuration cylinder with different area sizes. Switching modes is done for a low aerodynamic pressure range and high pressure range.

This type of cylinder with different area sizes is also known as a multi-chamber cylinder. There are many studies within the field of digital hydraulic implementing such a cylinder in mobile machines in order to overcome the typical throttling losses of traditional hydraulic systems. This was proven in [2] where a sophisticated system was implemented based on the multi-mode cylinder. The concept of digital hydraulics using the multi-chamber cylinder has been investigated in several studies for flight control actuation. A summary of several concepts is found in [3]. This approach has been proven to drastically increase the energy efficiency of the hydraulic actuator but comes with an increased complexity and requires advanced control strategies.

Local electro-hydraulic systems have been developed in [4] and [5]. These units are defined as being important steps towards more electrification of the aircraft systems. For large commercial aircraft the local hydraulic systems can be placed near the consumers avoiding long pipes. The electro-hydrostatic unit in [5] handles all the nose gear functions. Local electro-hydraulic systems are already flying on the Airbus A380 as a back-up system, thus eliminating one of the hydraulic systems [6].

Other means to save hydraulic power is presented in [7], with variable system pressure as one of the biggest contributors. However, varying the system pressure is complex and requires knowledge on the required actuator force, according to [8].

A framework is prepared for a conceptual comparison of the two architectures both at a subsystem level and an aircraft level. This framework is based on work presented in [9]. At a subsystem level the power extraction, cooling needs, and mass are estimated. This information is integrated into an aircraft model where the impact of the system characteristics is accounted for. The main focus in this work is the subsystem level, but a discussion is made of the aircraft level integration studies. The intended case-study is a fighter aircraft, for which the systems weight are estimated and compared.

## 2 Delimitations

This work only considers the primary flight control actuation system with the intention to highlight possible benefits with a mode-controlled hydraulic actuator. Although synergies exist several other subsystems and airframe, these are not included in order to narrow the scope for the sake of this study. The aircraft and mission used for integration are solely fictive but serve the purpose of creating a realistic integration environment for the actuation system.

The intention with the study is to remain at a higher abstraction level, meaning that higher order dynamics, details, and advanced control are excluded to facilitate the comparison at an architectural level. The integration at aircraft level is primarily illustrative for this work but highlights the process.

## 3 Method

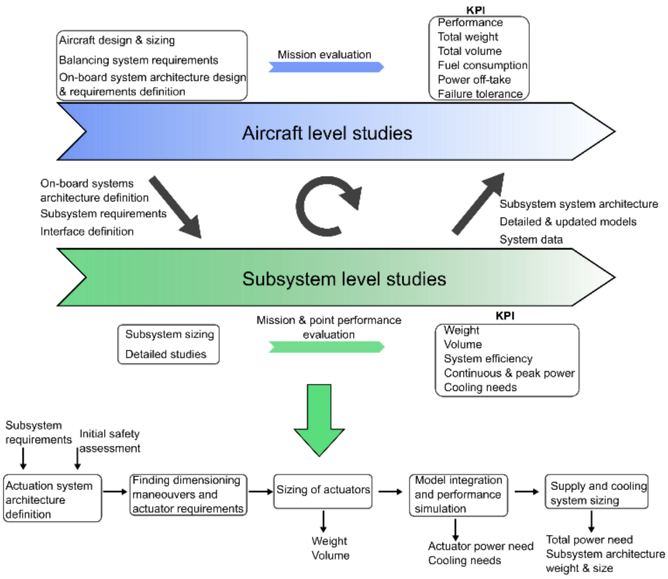


*Figure 1: Aircraft system design and on-board systems interaction, [9]*

A framework and methodology were developed in [9] for comparing different flight control actuation system architectures during the early aircraft conceptual design phase, where also a comprehensive description of the framework is presented. As shown in figure 1 studies should take place both at subsystem level and aircraft level providing a complete analysis of the system's performance, characteristics, and impact on the aircraft design. Following the steps of the defined methodology, an integrated environment allows to size and analyze the complete architecture, including actuators and supply system.

Information on the actuation system's characteristics and power requirements are propagated to the aircraft model. The design process is iterative during the entire phase. At an early stage, subsystem studies can benefit from a generic aircraft model of a certain configuration to analyze power and cooling needs, as well as initial sizing. This means that while the aircraft design is shaping up, subsystem studies can continuously provide the necessary information.

A generic fighter aircraft model of delta-canard configuration is used as a baseline for the actuation system study. The model is defined and described in [10]. By integrating the actuation system configurations an understanding of how the power and cooling needs vary for different flight maneuvers can be analyzed. Although a different design might eventually take place at aircraft level, this information is very useful to include early on. A proposal on how this is made is provided. The subsystem modelling and analysis have been done using the Matlab/Simulink tool version 2022b, while the aircraft integration has been done using the Desmo 9.0 tool from Pace.

## 4 System description and mode analysis

### 4.1 Mode control hydraulic actuator description

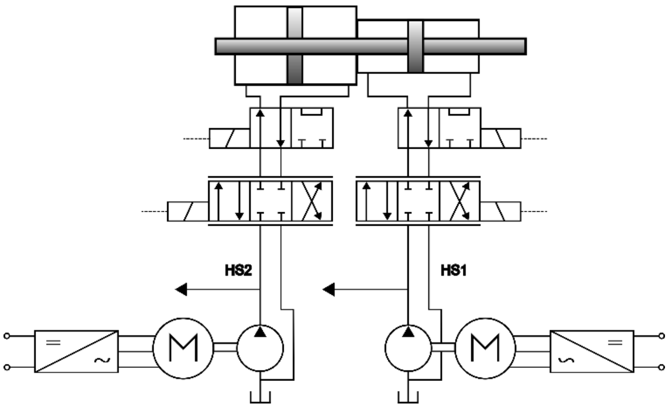


*Figure 2: Multi-mode hydraulic actuator.*

The reference hydraulic actuator consists of a symmetrical tandem cylinder supplied by two separated systems with gearbox driven pumps (the gearbox is attached to the engine). This is driven by safety requirements and the tandem cylinder ensures operation, although with limited power, in case of single failure. Two servo valves control the position of each half of the cylinder. For each side there is a by-pass valve connected both cylinder chambers in case of loss of hydraulic power.

The proposed system is shown in figure 2. It is similar to the reference with the main difference that each half of cylinder have different area sizes. Since the by-pass valves enable to deactivate one side of the cylinder, they can be used to choose how to operate the actuator: either the small side, the big side, or both sides. The actuator can be set in three different modes depending on the operating conditions. The total actuator force is defined by the required stall load, as for the reference actuator. The respective size of each cylinder side can be defined to minimize the total flow requirement.

Although the mode-controlled actuator works in a traditional centralized supply system, local electro-hydraulic drives are studied here. It is assumed that the motors run from a 270 VDC electric supply system.

### 4.2 Requirement analysis

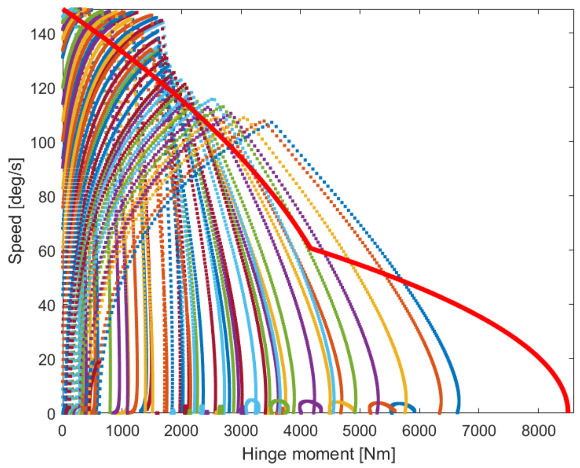


*Figure 3: Force and speed requirements.*

The required actuator force and speed can be found by simulating the aircraft model for various flight conditions and recording the control surface hinge moment and rate. An example is shown in figure 3, where a distinct separation is notable for higher aerodynamic pressure. As the aerodynamic pressure increases, the maximum control surface deflection is reduced and thereby the maximum speed is also reduced. It should be noted that the required speed and force looks different for other aircraft configurations, this is just one example. An approximative function is defined bounding the requirements region, shown as the bold red curve. The function is used for defining the actuator flow requirements.

Different control modes of the actuator are possible, a few are presented here in figure 4. The first approach, similar to the case with the YF-23 [1], is to choose control mode with respect to the maximum aerodynamic pressure, (q). The maximum q is given by the current flight condition: flight altitude and speed. In this way the mode shifting is not very frequent and depends on relatively slow changing variables compared to the actuator performance. Two strategies are evaluated based on q with two modes and three modes respectively. In both cases the maximum force is given by activating both cylinder sides. For the two-mode case, either the smaller piston area is activated or both, depending on the q. The piston and valve sizes are selected to fulfill the required actuator speed and the resulting flow and speed are marked as “2M, q-cntrl” in the figure. The three-mode case is similar with three different q zones. The required speed has been traded to keep the required flow low. This case is marked as “3M, q-cntrl”.

The other two strategies are based on a continuous mode shift depending on the actual actuator load. The load depends on the flight conditions and actuator position. This means that the actuator changes mode during its own operation. As the actuator move and the load increases, a mode shift occurs when the load reaches a certain limit. Both two modes and three modes are evaluated where the piston area and valve sizes are selected to minimize the required flow to maintain the required speed and fulfill the required stall load.

The most interesting approaches are the two-mode q-control and the three-mode continuous control strategies. Both fulfill the required speed while reducing the required flow by 50% and 62% respectively. The advantage with the q-control strategy is that mode-shifting only depends on the flight conditions. It will take place as the aircraft transition from one region of the flight envelope to another. Mode shifting takes place less frequently and not during actuator movement, thus, it could be easier to handle and less noticeable in actuator performance. The continuous control strategy shifts mode during actuator movement and will be more sensitive to handle. Nevertheless, the load is still dependent on the flight conditions and up to three mode shifts will only occur in the highest q-region. The main advantage is the additional reduction in required flow, that will affect the entire supply chain and installation.

The investigation continues with the three-mode strategy, focusing on integration aspects. Steady-state models are used for the analysis, while dynamic aspects and mode-shifting are left for future work.

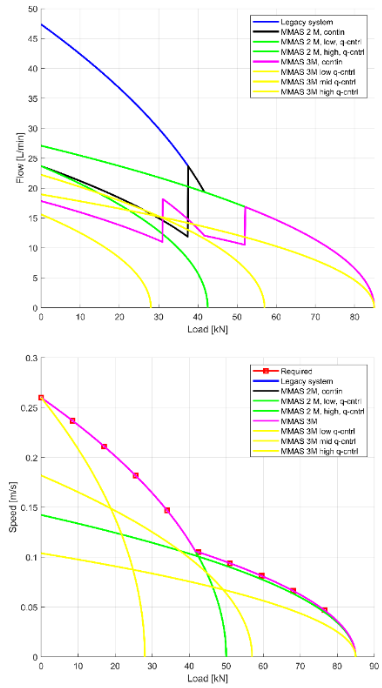


*Figure 4: Various mode selection strategies.*

### 4.3 Architecture definition

Two actuation system architectures are considered in this work as illustrated in figures 5, the reference system, and 6, the multi-mode system. Since the example aircraft has one engine, only one gearbox is available. Although no detailed safety analysis has been made, the two architectures have been designed with resilience to single fault failure in mind, that is, no single fault shall lead to a catastrophic event. There are in total seven control surfaces, each connects to a dual redundant actuator.

The reference system is supplied by a centralized hydraulic system. There a two separated system supplying each actuator. The pumps are attached to the same gearbox driven by the engine. Cooling is considered both for the hydraulic oil and the gearbox. The pump pressure is set to a constant 28 MPa and has a variable displacement, providing only the required amount of oil to move the control surface, as well as to cover for internal leakage. The two separate hydraulic systems ensure functionality in case of any fault in the hydraulic chain. An electrohydraulic unit supplied from a 270 VDC battery provides hydraulic power to one system if gearbox drive is lost. Additional safety measures could be necessary in practice.

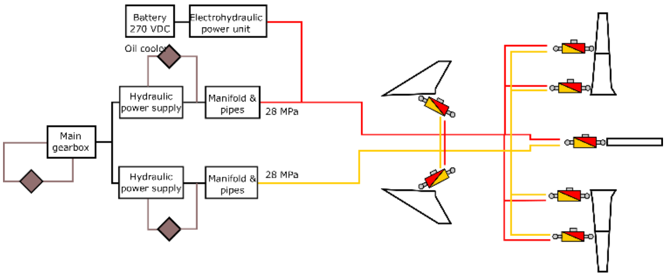


*Figure 5: Reference hydraulic architecture.*

The multi-mode system is designed in a similar way. Two redundant electro-hydraulic units supply oil to the rear actuators (elevators, ailerons, and rudder) and a forward unit supplies the canard actuators. Each unit consists of two fixed displacement electric motor driven pumps that share the same reservoir. A 270 VDC electric system is assumed. The electric system consists of a wild frequency generator and rectifying control unit, both systems driven by the same gearbox. Oil cooling is assumed for the electric power generation and gearbox. Two separate electric power systems ensure actuator drive in case for any single fault. If the gearbox drive is lost a battery pack can supply one of the electric systems.

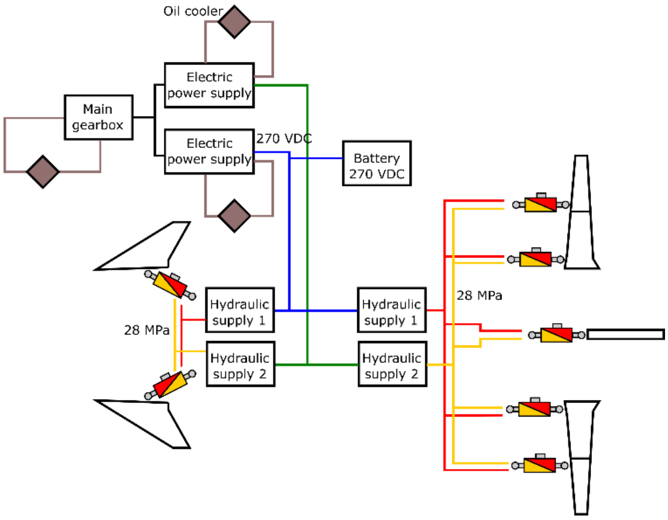


*Figure 6: Multi-mode actuation system architecture.*

## 5 System integration

### 5.1 Actuation system modelling

The system modelling has taken place in the Matlab/Simulink package using the Simscape library, shown in figure 7. Both the reference actuator and the multi-mode actuator are modelled in a similar way. The difference is in the piston sizing and control strategy.

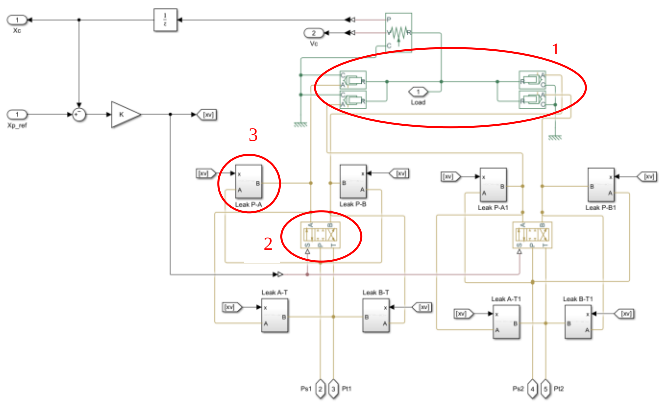


*Figure 7: Actuator simulation model. 1) shows the multi-chamber cylinder, 2) the servo valve model, and 3) the leakage model.*

The servo valves are represented by a 4-way directional valve considering the transition from laminar to turbulent flow. The discharge coefficient is set to 0.7 and a maximum valve opening to 1 mm. The cylinder is represented by four hydro-mechanical converters. No oil compression is considered. The piston size is defined by the stall load requirements according to eq. 1, assuming zero tank pressure.

$$A_p = \frac{F_{max}}{2p_s} \tag{1}$$

A leakage function is inserted for each of the valve spool lands, as shown in figure 8. This function gives maximum leakage in the centered and opening position, but decays as the spool closes on the corresponding side. The leakage will thus add to the total output flow, although other approaches are possible.

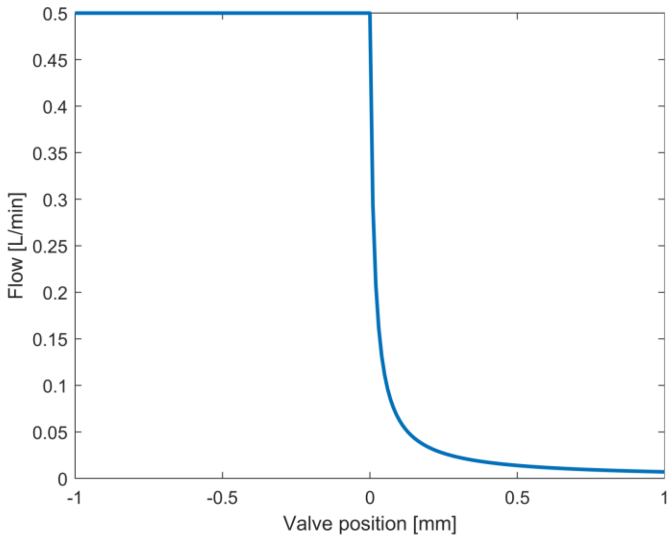


*Figure 8: Servo valve leakage curve.*

The actuator control loop is based on a generic approach. The actuator with its controller can be represented by the block diagram in figure 9, where Kp is the controller gain, Kq the plant gain, Ap the piston size, and s the Laplace operator.

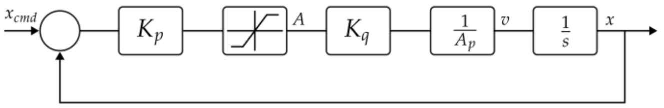


*Figure 9: Hydraulic actuator block diagram.*

The block diagram can be transformed into a first order transfer function on the form in eq. 2. The details of this operation are covered in reference [9].

$$G = \frac{1}{\tau \cdot s + 1} \qquad (2)$$

By identification the controller gain is defined as in eq. 3.

$$K_p = \frac{A_p}{\tau K_q} \qquad (3)$$

The maximum opening area is defined by eq. 4.

$$A_{max} = \frac{q_{max}}{C_q \sqrt{\frac{1}{\rho}(p_s - p_{nom})}} \qquad (4)$$

The maximum flow, qmax, is defined by the speed requirements and piston size, the pump pressure is set to 28 MPa (tank pressure is assumed zero but is in practice a fraction of the pump pressure), and the nominal pressure, pnom, could be set to any desired value but is here selected to zero.

Although the control strategy is similar to the multi-mode actuator, the mode shifting needs to be considered. Besides from the maximum load, two more load levels are introduced, Flow and Fmid, with the relation according to eq. 5.

$$F_{max} = F_{low} + F_{mid} \qquad (5)$$

These two force levels depict when to change to a new mode, that is, to either activate one of the cylinder sides or both. The control strategy needs the actual actuator force to determine which mode to select. If the force is low than Flow than mode 1 is selected, if it is between Flow and Fmid mode 2 is selected, if it is higher than Fmid than mode 3, which activates both cylinder sides, is selected. The actuator is only allowed to stall when the maximum force for mode 3 is reached. For mode 1 and 2 the shift has to occur before Flow and Fmid are reached respectively. Therefore, the shift occurs at 1 kN before the predefined levels.

The definition of the maximum valve opening differs also to the reference actuator. The reference actuator is defined to give maximum flow at maximum pressure drop and is allowed to drop off as the load pressure reaches the stall load. The multi-mode actuator has to consider the performance for each mode and has to deliver sufficient flow for the entire mode 1 and 2 regions. The maximum flow is therefore defined to occur at a pressure drop corresponding to 1 kN of load force. To prevent the actuator to deliver more flow corresponding to the maximum speed requirements, a dynamic saturation is introduced. The saturation calculates the maximum allowed valve position based on the current force and mode. Equation 4 can again be used here considering that the pressure drop is calculated from the current measured force. To better shape the response of the actuator a pre-filter is introduced to the reference position. The pre-filter is a first order filter where the time response is tuned so that the actuator closed loop response matches the required response.

A validation test of the actuators' performance confirms the behavior of the reference actuator and mode-controlled actuator compared to the requirements. A step of 10 mm is applied as reference position and a generic spring load with a rate of $7 \cdot 10^6$ N/m is working against the actuators. The performance is shown in figure 10. The required response is valid for the no-load case and therefore the reference actuator's performance drops off as the load increases. The mode shifting of the multi-mode actuator is clearly seen in both position and speed, although, no oil compression is included that could cause increased disturbance in position and speed control. At this point the multi-mode actuator's behavior is similar to the reference actuator.

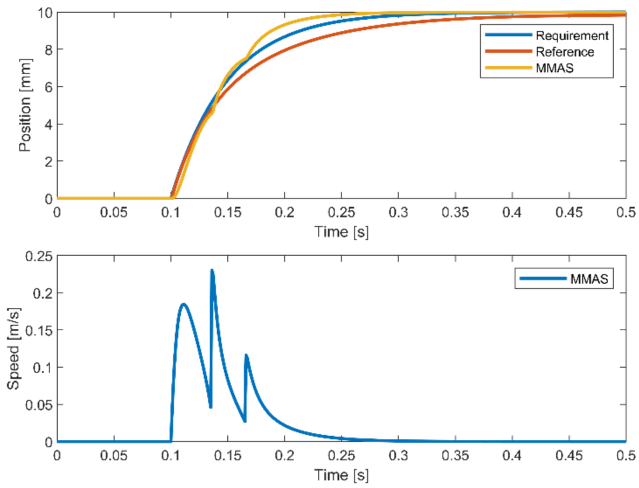


*Figure 10: Actuator response and speed.*

#### 5.1.1 Varying the supply pressure

A brief analysis of introducing variable pressure with a mode-controlled actuator is provided here.

Varying the supply pressure is a basic feature in any load sensing hydraulic system used to reduce the energy consumption, typically seen on mobile hydraulic machines. Some studies can be found on this for servo hydraulic systems in aircraft, as mentioned by [8]. The challenge is to anticipate the actuator power need without implementing a very complex system. Varying the system supply pressure has a very positive effect on life cycle cost savings, as reported in [7].

Anticipating the actuator power needs for controlling the supply pressure could be done in the same way as for the mode shifting based on the aerodynamic pressure. By controlling the supply pressure based on the flight conditions the load dependency between actuators is eliminated. Assuming only two pressure levels give an indication on possible benefits. The advantage is limited for a mode-controlled actuator since the full system pressure is used for a larger portion of the actuator's working envelope. The analysis is shown in figure 11. The required actuator load pressure is calculated from the load force and piston size, top plot. The two pressure levels are 15 MPa and 28 MPa. The required actuator power (ideally, since no losses are considered here) is shown in the bottom plot. Since the actuator remains in the low force region for the larger portion of the mission, great power savings can be expected. This would not affect the supply system sizing, but is interesting from an energy management perspective, increasing the overall system efficiency. Additionally, saving power could also reduce the heat load and thereby save on the heat exchanger weight. This possibility is part of future work, and a constant supply pressure is assumed for the remainder of the work, including more advanced pressure control strategies.

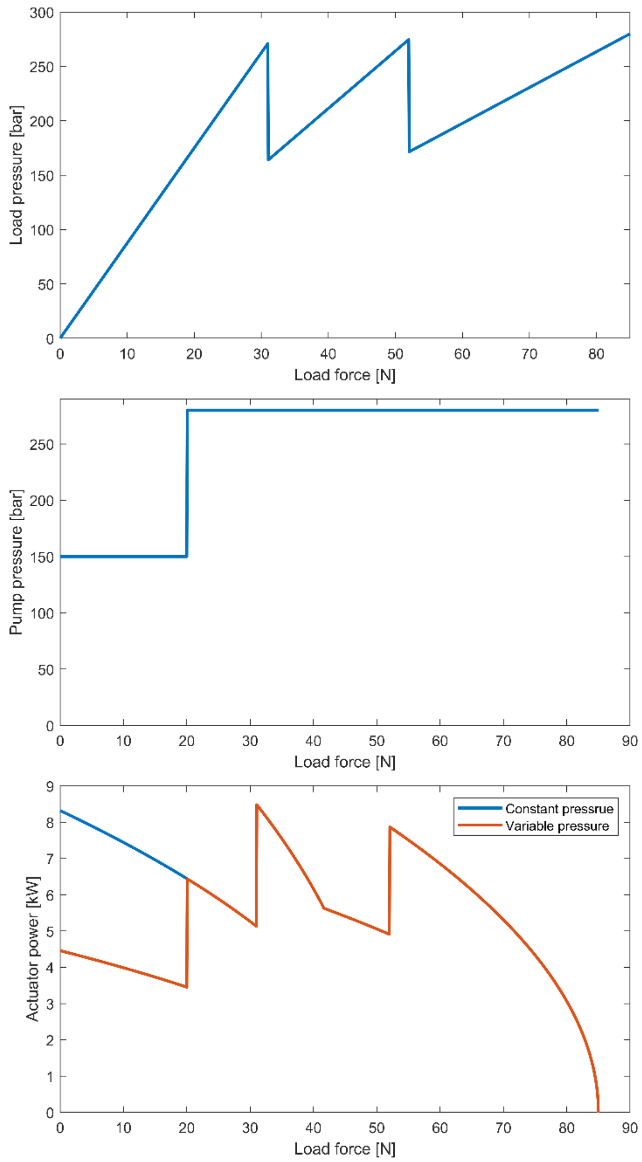


*Figure 11: Varying supply pressure analysis.*

### 5.2 System simulation integration

The system integration model was developed in reference [9], shown in figure 12. The framework is used both for supporting the system sizing and performance analysis. A 6 DOF fighter model is included, as well as aerodynamic data for calculating the control surface hinge moments. The model includes the seven control surfaces and the supply system. It is possible to simulate single maneuvers or an entire mission. The advantage of this approach is that a more realistic scenario for which the system can be analyzed is given. It is also advantageous from a sizing perspective since all actuators will not work equally all the time. A more accurate flow demand is estimated.

The actuator models are the previously defined. The hydraulic and electrical supply systems are represented by taking the sum of the actuator power, that is pressure times flow and applying a constant efficiency term. The efficiencies are assumed to be 80% for the pumps, and 90% for the electric motor and generators, and 90% for the gearbox.

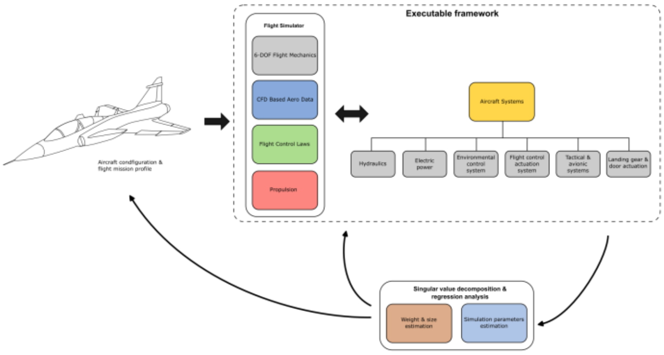


*Figure 12: System simulation framework, [9].*

### 5.3 System sizing and weight estimation

Among the most important characteristics when considering aircraft integration are the subsystems' weight, size, power consumption, and cooling needs. The focus here is all the mentioned besides the size.

The force and speed requirements are previously defined in chapter 4 by simulating a full stick input (pitch and roll) flight maneuver for various flight conditions. A safety factor of 1.2 is applied. This results in the following force and speed values used for sizing:

| Control surface | Force (kN) | Speed (m/s) |
|---|---|---|
| Elevator | 85 | 0.26 |
| Ailerons | 115 | 0.26 |
| Canards | 85 | 0.1 |
| Rudder | 50 | 0.26 |

The approach in this work is to use open data to develop weight estimation models. The advantage is that several samples are available to develop a more accurate model. The downside is that installation effects are not included. The availability of open data, especially for aircraft components, is also limited which will skew the results. Since this is a relative comparison between a traditional system and the multi-mode system, the lack of information applies to both systems. The weight estimation of each component follows the respective architectures. All estimation models are derived from linear regression analysis.

The pump model is based on data from [11]. Both the weight and displacement are estimated with the pump flow as input. The displacement is needed to calculate the electric motor torque for the electro-hydraulic system. The relations are defined in eq. 6 and 7.

$$W_{pump} = 0.066 \cdot q + 0.2 \text{ kg} \tag{6}$$

$$D_{pump} = 0.0006q^2 + 0.15q + 2.16 \text{ cm}^3\text{/rev} \tag{7}$$

The reservoir is assumed to be a bootstrap configuration. The weight estimation is based on data from reference [12] with total oil volume, V, as input. The relation is given in eq. 8.

$$Weight_{reserv} = 0.4 \cdot V + 5.7 \text{ kg} \tag{8}$$

The total oil volume is calculated from each actuator's piston size and stroke, and the oil in the hydraulic pipes, in addition 30%.

The gearbox weight is based on only one sample found in [13] and scaled linearly with a weight to power ratio of 0.5 kg/kW.

The electric power generation is also based on one sample from reference [14] and scaled linearly with a weight to power ratio of 0.23 kg/kW. This includes both generator and rectifier unit.

The electric motor weight estimation is based on several samples of permanent magnet motor for industrial use, without any forced cooling from reference [15]. The weight scales linearly with motor torque according to eq. 9.

$$Weight_{motor} = 0.33 \cdot T + 0.7 \text{ kg} \tag{9}$$

The electric motor torque is given from eq. 10 with the maximum pump pressure and a mechanical efficiency of 90%.

$$T = \frac{D_p p_s}{2\pi \cdot \eta} \tag{9}$$

The oil cooler is based on one sample from reference [16] and scaled linearly with the required cooling power using a weight to power ratio of 0.13 kg/kW. The required cooling power is given from the internal leakage in the system. Since the power required to drive the actuators is only intermittent during maneuvering, it is assumed that the main loss is from the leakage.

The weight for the hydraulic piping and electrical cables are given from the aircraft level integration model in chapter 5.4. This also gives the total weight of the oil.

### 5.4 Aircraft level integration

The Desmo 9.0 tool from Pace, [17], is used for aircraft level integration. The tool allows initial aircraft sizing based on a defined mission, as well as on-board systems architecture design and analysis. The architecture representation includes the weight, size, power consumption and bleed-air extraction from the engine. The physical representation of the components also gives a weight distribution along the aircraft. There is a built-in function that routes and sizes the power distribution elements, like hydraulic pipes and electrical cables.

The information on the subsystems' characteristics needs to be incorporated into the aircraft model. A full aircraft and system sizing is not provided in this work but a discussion on how the integration can take place. A fictive aircraft is designed based on one of the example models provided by Desmo, shown in figure 13. The figure shows the reference architecture, where all the green components represent the actuators and the hydraulic supply system. The red lines are the hydraulic distribution elements used for weight estimation of pipes and cables.

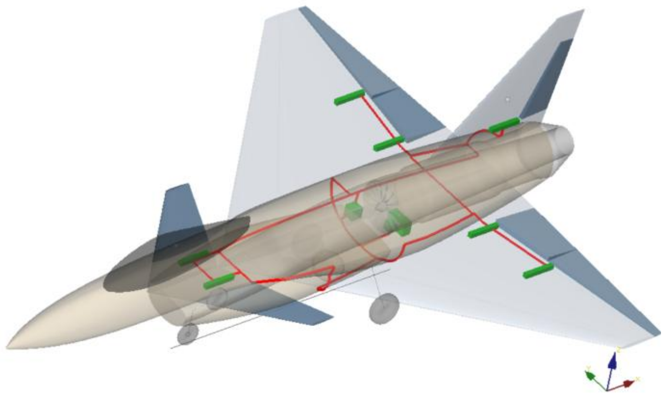

*Figure 13: Example aircraft model implemented in Desmo 9.0.*

Each component representation stores its own weight and power consumption. The flight mission is divided into several segments, like taxi, take-off, climb, cruise, etc. It is up to the user to define the granularity of the mission segments. The power consumption of each component is then defined for each segment. Since Desmo is not a time-based simulation tool, one approach to include the flight control actuators' power usage is to analyze the behavior from the time-based simulation for various maneuvers for a similar aircraft, and taking average power usage as input to that specific flight segment. An example of how this can be done is shown in figure 14. It shows the pump flow for the reference system during a sustained turn at Mach 0.9 at 5000 meters. A sustained turn segment can be implemented in Desmo where the average flow, or power, from the system simulation is used. In this way the different mission segments defined in Desmo can be populated with information from the system simulation model.

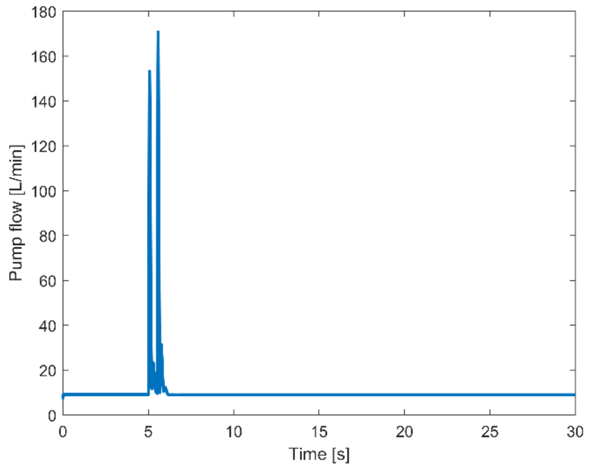


*Figure 14: Required pump flow during a sustained turn.*

## 6 Analysis

This analysis is focused on the weight comparison between the two systems shown in figures 15 and 16. The data from the systems simulation and aircraft integration has been used to estimate the weight of each component of the architectures. The total weight is similar, 115 kg compared to 125 kg, for the reference and multi-mode systems respectively. These numbers should not be seen as absolute. The quality of the results depends on the available data. Open reliable data is hard to find and for some components only one sample was used. The electrical motors are based on industrial data and are not optimized for aircraft applications. Installation effects are also not included since it would require detailed knowledge about the actual aircraft. Nevertheless, the results show that by using a multi-mode control approach, a local electro-hydraulic system can be significantly down-sized. This would, of course, also apply to a traditional architecture, though an electro-hydraulic system benefits from an increased flexibility and could easily share power between systems.

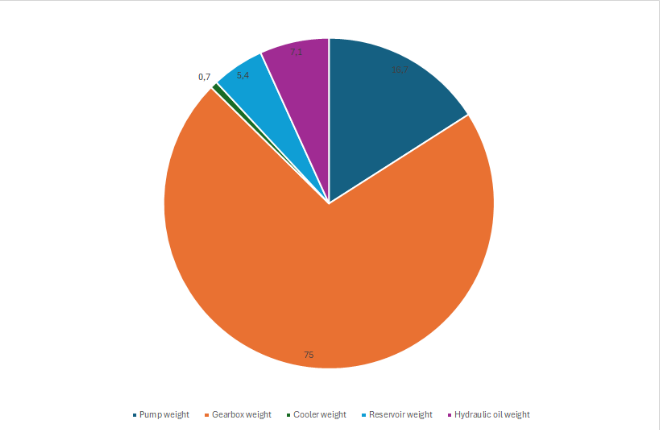


*Figure 15: Weight distribution reference system.*

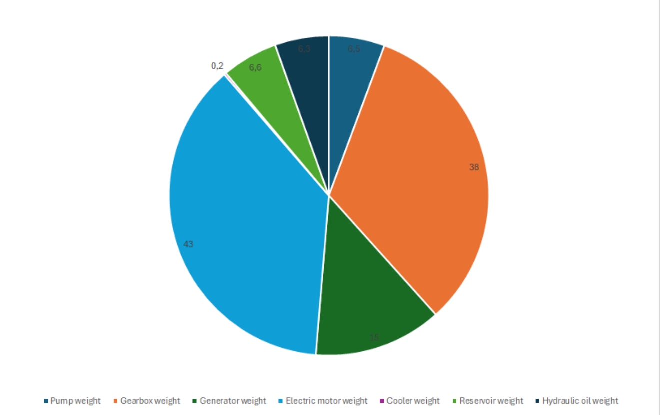


*Figure 16: Weight distribution multi-mode system.*

The presented work applies to top level studies in the design process. Later stages should include more detailed studies that might alter the outcome. Examples of such effects to include are hysteresis in the switching strategy, influence from natural dynamics and dynamic stiffness, excitation of natural modes coming from transients generated by the commutation, etc.

## 7 Conclusions

This work has shown an approach to size and analyze an hydraulic flight control system. The weight of traditional system was compared to a local electro-hydraulic multi-mode system. The multi-mode approach helps to significantly downsize the power requirements, up to 60% is possible to save. The work also highlights the interaction between subsystem level and aircraft level studies, with the aim to analyze the complete on-board systems architecture and its impact on aircraft performance. A method is proposed where system simulation is used to populate an aircraft sizing model with information on the flight control actuation power usage.

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