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Force Equalization for Active/Active Redundant Actuation System Involving Servo-hydraulic and Electro-mechanical Technologies

Doctoral Thesis

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To everyone I love!
Abstract

**Keywords:** force equalization, active/active, hybrid actuation, redundant, servo-hydraulic, electro-mechanical

On the way to more electric aircraft (MEA), more and more power-by-wire (PBW) actuators are involved in the flight control system. For a hybrid redundant actuation system composed by the conventional hydraulically powered actuators and the PBW actuators, one major issue while they operate on active/active mode is the force fighting between channels. As the grave influence of force fighting on accelerating material fatigue and increasing power consumption, it must be addressed with attention. This thesis was aiming at proposing some effective force equalization control strategies for the hybrid actuation system involving one servo-hydraulic actuator (SHA) and one electro-mechanical actuator (EMA). For this objective, the position controllers for SHA and EMA were designed and validated as a first step. Then, a virtual test bench regarding to the realistic behaviors was built in the AMESim simulation environment to accelerate the controller design and enable the robustness study. Following this, 2 static force equalization control strategies were proposed and experimentally validated. The first strategy that introduced integral force fighting signal to compensate the actuator position control was proved a good candidate solution. In the next part, 3 dynamic force equalization strategies were proposed and assessed on the virtual test bench. Their performance sensitivities to the parameter uncertainties were studied through Monte-Carlo method. The first strategy that introduced velocity and acceleration feed-forwards to force the SHA and EMA having similar pursuit dynamics showed a good force equalization performance as well as good segregation and good robustness. In the end, the work presented in thesis was concluded and perspective was given to the ongoing work.

Résumé

**Mots Clés:** Égalisation d’effort, actif/actif, l’actionnement hybride, redondant, servo-hydraulique, électromécanique

L’évolution vers les avions plus électriques engendre des efforts importants pour développer des actionneurs à source de puissance électrique pour les commandes de vol. Pour de telles applications critiques, il est peut être intéressant dans le futur d’associer à une même surface de contrôle un actionneur conventionnel à source de puissance hydraulique et un actionneur à source de puissance électrique mais ceci pose un problème important lorsque les deux actionneurs sont actifs simultanément: comme chacun essaye d'imposer sa position à l'autre, les deux actionneurs luttent l’un contre l'autre en développant des efforts néfastes qui ne sont pas utilisés par la charge. L’objet du présent travail est de proposer des stratégies d’égualisation d’effort pour un système d’actionnement impliquant ces deux types d’actionneurs opérant en mode actif-actif. La première étape est de concevoir leur commande en position et de la valider sur banc d’essai. Un banc d'essai virtuel fidèle à la réalité est ensuite réalisé dans l’environnement de simulation AMESim pour pouvoir évaluer facilement les différentes stratégies d’égualisation d’effort entre les deux actionneurs. Ces stratégies sont proposées et évaluées virtuellement en deux étapes, statique puis dynamique. Pour finir, une étude de robustesse est réalisée a posteriori pour évaluer la sensibilité des indicateurs de performance aux incertitudes sur les modèles de simulation et sur les points et les conditions de fonctionnement.
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<tr>
<td>$\alpha_1$</td>
<td>EMA open loop pursuit gain [m/(A·s)]</td>
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<tr>
<td>$\beta$</td>
<td>SHA open loop rejection gain [m/N]</td>
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<td>$E_y$</td>
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<td>$F_e$</td>
<td>EMA output force [N]</td>
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<td>$F_f$</td>
<td>Jack friction [N]</td>
</tr>
<tr>
<td>$F_{jf}$</td>
<td>Joint friction between SHA and EMA [N]</td>
</tr>
</tbody>
</table>
Nomenclature

$F_{jct}$ Joint Coulomb friction $[N]$
$F_s$ SHA output force $[N]$
$F_{cl}$ Jack Coulomb friction $[N]$
$F_{st}$ Jack Strubeck friction $[N]$
$G_1$ SHA closed loop characteristics polynomial
$G_2$ SHA closed loop rejection function
$H_1$ EMA closed loop characteristics polynomial
$H_2$ EMA closed loop rejection function
$I_{rv}$ Servovalve input current $[A]$
$J_m$ Total inertia in motor side $[Kg \cdot m^2]$
$K_{ac}$ Jack leakage coefficient $[(m^3/s)/Pa]$
$K_c$ SHA total flow/pressure gain $[(m^3/s)/Pa]$
$K_{eb}$ EMA position control gain $[1/s]$
$K_{ec}$ EMA velocity control gain $[N \cdot s]$
$K_{ef}$ EMA force control proportional gain $[1/m]$
$K_{sb}$ SHA position control gain $[A/m]$
$K_{sf}$ SHA force control proportional gain $[A/N]$
$K_{sc}$ Servovalve flow/pressure gain $[(m^3/s)/Pa]$
$K_{sg}$ Servovalve flow/current gain $[(m^3/s)/A]$
$K_{sq}$ Servovalve flow/opening gain at null opening drop $[m^2/s]$
$K_{sv}$ Servovalve opening/current gain $[m/A]$
$K_p$ Force fighting feedback PID controller P gain $[m/N]$
$K_i$ Force fighting feedback PID controller I gain $[m/Ns]$
$K_d$ Force fighting feedback PID controller D gain $[ms/N]$
$K_{sp}$ SHA PID force controller P gain $[A/N]$
$K_{si}$ SHA PID force controller I gain $[A/Ns]$
$K_{sd}$ SHA PID force controller D gain $[As/N]$
$K_{vc}$ SHA velocity feed-forward compensation gain $[As/m]$
$M_e$ EMA rod level $[Kg]$
$M_s$ SHA rod level $[Kg]$
$M_t$ Surface equivalent mass on rod level $[Kg]$
$Q_{sv}$ Servovalve output flow $[m^3/s]$
$R$ Reliability index
$P_1$ SHA jack chamber 1 pressure $[Pa]$
$P_2$ SHA jack chamber 2 pressure $[Pa]$
$P_f$ SHA load pressure $[Pa]$
$P_r$ Hydraulic return pressure $[Pa]$
$P_s$ Hydraulic supply pressure $[Pa]$
$S_h$ SHA hydraulic stiffness $[N/m]$
$S_L$ Load closed loop static stiffness $[N/m]$
$S_{rs}$ Roller-screw stiffness $[N/m]$
$S_{em}$ EMA body static stiffness $[N/m]$
$S_{sm}$ SHA body static stiffness $[N/m]$
$S_{ep}$ EMA closed loop static stiffness $[N/m]$
$S_{sp}$ SHA closed loop static stiffness $[N/m]$

**Nomenclature**

<table>
<thead>
<tr>
<th>Symbol</th>
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<tbody>
<tr>
<td>$S_{ct}$</td>
<td>EMA transmission stiffness $[N/m]$</td>
</tr>
<tr>
<td>$S_{st}$</td>
<td>SHA transmission stiffness $[N/m]$</td>
</tr>
<tr>
<td>$S_{ec}$</td>
<td>EMA transmission stiffness out of backlash domain $[N/m]$</td>
</tr>
<tr>
<td>$S_{sc}$</td>
<td>SHA transmission stiffness out of backlash domain $[N/m]$</td>
</tr>
<tr>
<td>$T_d$</td>
<td>EMA motor torque demand $[N \cdot m]$</td>
</tr>
<tr>
<td>$T_e$</td>
<td>EMA Mechanical output torque $[N \cdot m]$</td>
</tr>
<tr>
<td>$T_f$</td>
<td>EMA roller-screw friction torque $[N \cdot m]$</td>
</tr>
<tr>
<td>$T_c$</td>
<td>EMA roller-screw Coulomb friction torque $[N \cdot m]$</td>
</tr>
<tr>
<td>$T_{st}$</td>
<td>EMA roller-screw Striebeck friction torque $[N \cdot m]$</td>
</tr>
<tr>
<td>$T_m$</td>
<td>EMA motor electrical output torque $[N \cdot m]$</td>
</tr>
<tr>
<td>$V_1$</td>
<td>SHA jack chamber 1 volume $[m^3]$</td>
</tr>
<tr>
<td>$V_2$</td>
<td>SHA jack chamber 2 volume $[m^3]$</td>
</tr>
<tr>
<td>$V_t$</td>
<td>Jack total equivalent volume $[m^3]$</td>
</tr>
<tr>
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<td>Jack friction windage coefficient $[N \cdot s^2/m]$</td>
</tr>
<tr>
<td>$X_e$</td>
<td>EMA output position $[m]$</td>
</tr>
<tr>
<td>$X_r$</td>
<td>Actuator position demand $[m]$</td>
</tr>
<tr>
<td>$X_s$</td>
<td>SHA output position $[m]$</td>
</tr>
<tr>
<td>$X_{ecr}$</td>
<td>EMA transmission part backlash factor $[m]$</td>
</tr>
<tr>
<td>$X_{scr}$</td>
<td>SHA transmission part backlash factor $[m]$</td>
</tr>
<tr>
<td>$X_{ec}$</td>
<td>Transmission part compression between EMA and load $[m]$</td>
</tr>
<tr>
<td>$X_{sc}$</td>
<td>Transmission part compression between SHA and load $[m]$</td>
</tr>
<tr>
<td>$X_{sv}$</td>
<td>Servovalve spool displacement $[m]$</td>
</tr>
<tr>
<td>$X_t$</td>
<td>Load position $[m]$</td>
</tr>
</tbody>
</table>

**Acronyms**

- FE: Force Equalization
- FF: Force Fighting
- BLDC: Brushless DC motor
- DFE: Dynamic Force Equalization
- DFF: Dynamic Force Fighting
- EBHA: Electro Backup Hydrostatic Actuator
- EHA: Electro-Hydrostatic Actuator
- EMA: Electro-Mechanical Actuator
- FBW: Fly By Wire
- FBWL: Fly By Wireless
- FBL: Fly By Light
- FFC: Flight Control Computer
- HMA: Hydro-Mechanical Actuator
- MDE: Motor Drive Electronics
- MEA: More Electric Aircraft
- MMI: Man-Machine Interface
- MSV: Mode Select Valve
- NRC: Non Recurring Cost
- PBW: Power By Wire
- RC: Recurring Cost
- SFE: Static Force Equalization
- SFF: Static Force Fighting
<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
</tr>
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<tbody>
<tr>
<td>SHA</td>
<td>Servo-Hydraulic Actuator</td>
</tr>
<tr>
<td>THS(A)</td>
<td>Trimmable Horizontal Stabiliser (Actuator)</td>
</tr>
<tr>
<td>TRL</td>
<td>Technology Readiness Level</td>
</tr>
<tr>
<td>XPC</td>
<td>Real Time Control Computer on Test Bench</td>
</tr>
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</table>
Introduction

As the rapid growth of air travel in recent years, the aircraft industry contributes an increasing carbon emission. In European Union, the greenhouse gas emissions from aviation increased by 87% between 1990 and 2006 [1]. According to data of the United Nations Intergovernmental Panel on Climate Change (IPCC), aviation has produced around 2% of the world's manmade emissions of carbon dioxide (CO$_2$) in 2011. As aviation grows to meet increasing demand, the IPCC forecasts that its share of global manmade CO$_2$ emissions will increase to around 3% in 2050 [2]. However, despite a growth in passenger numbers at an average of 5% each year, the aviation has managed to decouple its emissions growth by around 3% (or some 20 million tons annually). This is achieved through massive investment in new technology and operating procedures. As the world’s two biggest airliner producers, Airbus and Boeing make a great effort on improving their best selling single aisle aircrafts A320 and B737 to meet this objective. Their new developing models are A320Neo and B737Max, which have a better fuel-efficient than the current models [3, 4].

Meanwhile a constant investment is being done in aviation for the next generation of airliners, which are required to be much greener, cheaper and safer with an increased passenger appeal.

As indicated in Fig 1, except for employing better fuel efficient engines, lighter composite
material airframe and better aerodynamic design, the technology improvement of onboard systems also plays an important role in each innovation. The onboard systems are desired to be much quieter, lighter, safer and more efficient.

As indicated in Fig 2, in these onboard systems, the hydraulic networks, flight control system and landing gear system have the biggest potential of improvement. On current aircrafts, hydraulics is the main secondary power network. It is energized by the engines and supplies power for flight control actuation and landing gear system by heavy and bulky pipes. As the flight control actuation is one of the major secondary power consumers, the effect of its efficiency improving is significant. The efficiency of hydraulic network is lower in comparison with the electric network. So it is attractive to introduce more electric power user to enable replacing the hydraulic ones. The heavy pipes shall be replaced by lighter wires and the energy loss in power networks will be reduced. The energy efficiency of actuation is also greatly improved as the actuators can operate on a power-on-demand principle.

On recently developed commercial airliners Airbus A380 and Boeing B787, the electric power networks have been used to replace one hydraulic network. Due to the lack of maturity of electrically powered actuators, they are only used as the backup of the hydraulically powered actuators. However, with the constant efforts of aviation, the electrically powered actuators will certainly be more involved on the next generation of airliners. They will work together with the hydraulic actuators in primary flight control system because it is still unacceptable to replace all the hydraulic actuators for safety and maturity reasons for the time being.
As shown in Fig 3, for such a hybrid actuation system operating on active/active mode, one major issue is the force fighting between actuators. Because of their dissimilar technology and absolutely different behaviors, the actuators cannot output exactly the same position and will fight with the other one to position the flight control surface. The force fighting can accelerate the fatigue of material, reduce service life and increase energy consumption. Therefore, the actuators output forces must be equaled to keep the force fighting within an admissible range to reduce its grave influence.

This thesis is proposed to address the force equalization issue of a hybrid actuation system composed by one servo-hydraulic actuator (SHA, hydraulically powered) and one electro-mechanical actuator (EMA, electrically powered), which operates on active/active mode and is sized for the ailerons of single aisle airliners.

The thesis is organized as shown in Fig 4:

In chapter 1, state of the art of the flight control actuation is introduced and the challenges of this field in recent period and future are discussed. Some applications of flight control system
are presented. The redundant design necessity of actuation system is studied and the general redundant configurations are summarized. In the end, the hybrid actuation test bench used to help performing the research is introduced and the detailed objective of thesis is proposed.

In chapter 2, the position controllers of SHA and EMA are designed on the basis of the linear approach and experimentally validated on the test bench. Then, with the linear prototype of hybrid redundant actuation system built in Matlab/Simulink simulation environment [5], the force fighting is primarily studied and pointed out.

In chapter 3, a virtual prototype of the test bench is built and detailed in the LMS_AMESim [6] simulation environment in order to more accurately represent the behaviors of the real one. Many advanced nonlinear models are considered, like the servovalve spool dynamics, flow /pressure characteristics, jack friction, roller-screw friction, backlash, joints compliance, and so on. Some partial experiments are done to validate the virtual test bench. In the end, the force fighting is represented and deeply studied through the virtual test bench.

In chapter 4, on the basis of virtual test bench, the static force equalization of the redundant actuation system is addressed. At first, the origins of static force fighting are studied in details. Then, two static force equalization control strategies are proposed and analyzed. In the end, the strategies are pre-validated on the virtual test bench and then experimentally validated on the real test bench.

In chapter 5, three dynamic force equalization control strategies are proposed and assessed on the virtual test bench. Then, the robustness of these control strategies is studied through evaluating the performance sensitivity to parameter variation with the Monte-Carlo method [7, 8]. In the end, the proposed force equalization strategies are compared in a more complete way considering the surface position performance, force equalization, robustness, segregation and complexity.

In chapter 6, the present work is concluded and an outlook over the further improvements and perspectives is provided.
Chapter 1

State of the Art and Test Bench

The thesis is about the force equalization of redundant actuation systems involving dissimilar technology actuators which operate on active/active mode. So in present chapter, the state of the art of flight control actuation will be introduced. To start with, the operation functions and development history of flight control actuation is presented. The necessity and configurations of redundant design in actuation field is studied as following. After that, the mainstream fly-by-wire (FBW) actuators are introduced. In the final part, the test bench used for thesis is displayed and the final objective is given out.

1.1 State of the Art

One hundred years ago, on December 17, 1903, two Americans, the Wright brothers, Orville and Wilbur, invented and built the world’s first successful aircraft and made the first controlled, powered and sustained heavier-than-air human flight [9]. From then on, human have produced thousands of various aircrafts, which have greatly changed the world.

1.1.1 Flight Control Actuation System

The aircraft is a complex vehicle flying on air and composed by thousands of components. In order to ensure safe flying, all the components are desired to accurately work and cooperate with each other. As shown in Fig 1-1, the aircraft is composed by three main parts: airframe, engine and onboard systems [10]. The airframe is the mechanical structure to support the engines and onboard systems; the engine supplies propulsion for aircraft and generates secondary power for onboard systems; the onboard systems are in charge of all the other works, flight control, landing gear, fuel, environment, anti-ice, navigation, communication, and so on.
Chapter 1 State of the Art and Test Bench

Fig 1-1 Configuration of aircrafts

The secondary power includes hydraulic, electric, pneumatic and mechanic types. The flight control actuation is powered by the hydraulic and electric ones. It is in charge of controlling the aircraft attitude and aerodynamic configuration by moving the flight control surfaces. It is divided into primary level and secondary level. The primary flight control system controls the attitude in the yaw axis (by rudders), pitch axis (by elevators) and roll axis (by ailerons). The secondary flight control system controls the aircraft aerodynamic configuration through the high lift (by slats and flaps), trim (by trimmable horizontal stabiliser, THS) and the air brake (spoilers and some special surfaces) [11, 12]. As a safety critical application, the primary flight control system is strictly required to have a very good performance on statics, dynamics and reliability.

Fig 1-2 Flight control surfaces of a commercial airliner
As shown in Fig 1-2, the flight control surfaces on a commercial airliner are presented [13]. These surfaces are driven by actuators. For the primary flight control system, redundant actuators are generally employed to drive a common surface; for the secondary flight control system, one actuator with internally redundant arrangements is used for each function [14].

The operation architecture of these actuators is displayed in Fig 1-3. It can be represented considering the signaling and powering flows [15]. Following these two lines, there are four main sections: man-machine interface (MMI), flight control computer (FCC), power source and actuators. MMI is the interface between pilots and aircraft, like the control sticks, pedals and display screens. It is in charge of communicating with pilots, receiving orders and displaying the aircraft operating information. The autopilot is a common device on modern aircraft to substitute pilots controlling the flying trajectory and keeping stabilization during cruise. The pilots and autopilot give out the position demand for actuators according to flight attitude and transfer it to the FCC. FCC collects the position demand and sensors’ signals and calculates the control orders for power modulating device. Meanwhile FCC supplies the flight envelope protection to prevent the pilots from control commands that would force the aircrafts to exceed its structural and aerodynamic operating designs [16]. In the end, the actuator drives the flight control surface to the demanded position in response to the demand signals.

![Fig 1-3 Configuration of flight control system](image)

Generally the performance of flight control actuation is assessed by two items. As shown in Fig 1-4, the first item is pursuit performance. The actuator is controlled to force the angle $\theta$ of
flight control surface to follow the pilot demanded angle $\theta^*$. The pursuit is desired to be quick, accurate and stable. The second item is rejection performance. Under the influences from air load and disturbance, the flight control surface is desired to be not sensitive to them and the output angle error must remains within the allowable range.

As the actuator is the mechanical executive device in this flight control actuation system, its statics and dynamics behaviors have significant effects on making the upper performance to be satisfied. Therefore, sustained efforts are given on the technology innovation of actuators. In Tab 1-1, the history of its innovations on commercial aerospace is summarized and future developing tendency is also forecasted [12, 15, 17].

As indicated in Tab 1-1, on the earlier aircrafts until 1940s, the control stick and pedals were directly connected to the flight control surfaces by rods and cables. Not only the orders, but also the powers were coming from the pilots. So for the early pilots, operating an aircraft
needed both skills and power. But as the aircraft developed to larger and faster, the air load on
the flight control surfaces became bigger and bigger that made it more and more difficult for
pilots to supply so big power.

Until 1950s, the hydraulic actuator was developed and introduced into flight control system
to assist the pilots controlling the surfaces. In this system, the pilot orders were transmitted
by mechanical cables to control a hydraulic valve, which fixes the jack position by controlling
the flow according to the position error. This kind of actuators is called as hydro-mechanical
actuators (HMAs). The actuation power was supplied by an engine-driven pump. Meanwhile a
feel device was installed to help pilots knowing the surface response because the control stick
and pedals were no more directly connected to the surfaces and could not restitution their force
feedback to the pilots.

In the late of 1950s, the aircraft stability became very worse in case flying at a much higher
speed. This issue cannot be well solved by only adjusting the mechanical flight control system
or airframe aerodynamic. Then in the 1960s, the rate gyroscope and accelerometer were
introduced to construct a stability augmentation device. Its output was combined with the
pilot orders to control the hydraulic valve. This device was proved significantly increasing the
aircraft stability at high speed.

Then from 1970s, the rapidly developed electric devices were widely used in various fields. In
the 1980s, electric computer and electro-hydraulic servovalve were introduced into the flight
control system, which firstly has the full configuration as displayed in Fig 1-3. The pilot orders
were no more transmitted by mechanical cables but by electric wires. The power modulating
device, the servovalve, was also no more ordered by mechanical signal but by electric signal.
FCC became the middle section between pilots and actuators. It collected the necessary
information and generated the orders to servovalve. Because all the mechanical cables for
signaling were replaced by the electric wires, this kind of flight control actuation system was
called as fly-by-wire (FBW) system [18]. In this period, the hydraulic was still the unique
power source for actuation [19, 20]. The common employed actuators were servo-hydraulic
actuators (SHAs).
The first analogue FBW system for a commercial aircraft was installed in the Concorde in the 1970s [21]. After that, as its excellent performance, the FBW systems became widely used in the commercial aircraft designs. Now the FBW system already innovates from analog to digital, which is much more powerful and interference free [18].

In the 2000s, the electrically powered actuators were developed and started to be introduced into the flight control actuation system to replace the hydraulically powered actuators. The actuation power was no more generated by the engine-driven pumps but by engine-driven electrical generators and transmitted by electric wires. So this kind of flight control system was called as power-by-wire (PBW) system. Eliminating the heavy and bulky hydraulic pipes saved a great weight and meanwhile removed the pipes vibration problem. In the view of power transmission and modulating, the electric is also more efficient than the hydraulic.

The electrically powered actuators are the key components in the PBW system. Until now, two configurations have been developed, the electro-hydrostatic actuator (EHA) and the electro-mechanical actuator (EMA) [23].

As safety is a major requirement in commercial aviation, considering the lack of maturity of electrically powered actuators, the hydraulic actuators are still the dominant choice for flight control actuation system, even in the next decades. So in recent applications, the electrically powered actuators are mainly employed as the backup of hydraulic actuators, as on Airbus A380 and Boeing B787.

As shown in Fig 1-5, on the earlier airliners employing FBW technology, like the Airbus A320, all the actuation were hydraulically powered [24]. Now on recently developed Airbus A380, 2 electric power networks are employed to work together with the 2 hydraulic power networks. Under these 2 electric power networks, 8 EHAs are involved in inboard ailerons and 8 electric backup hydrostatic actuators (EBHAs) are involved in elevators and spoilers [25]. And on Boeing B787, 5 EMAs are involved in the spoilers and THS [26]. As mentioned upper, all these electrically powered actuators are working as the backup of SHAs.
But with the constant investment of aviation, the PBW actuators absolutely will be introduced step by step in flight control systems and contribute more to the more-electric-aircraft (MEA). On the MEAs, both the signal and power will be transmitted by electric wires [18]. One developing example is the military fighter JSF F-35 II, on which the surfaces are all driven by the PBW and FBW actuators [27].

Two other developing directions are all about the signal transmission. The first one replaces the electric wires by optical fiber cables so it is called as fly-by-light (FBL). It can transfer data at higher speed and volume. Meanwhile it is immune to electromagnetic interferences. The second one totally removes the wires and uses a wireless protocol to transfer data, so it is called as fly-by-wireless (FBWL). It can decrease the weight of wiring and have the potential to reduce costs throughout an aircraft’s life cycle [18].

1.1.2 Reliability Requirements and Redundant Design

According to the regulations of Federal Aviation Administration, the severity classification and authorized probability in commercial aerospace are summarized as Tab 1-2 [28, 29, 30].
## Tab 1-2 Commercial aircraft severity and authorized probability

<table>
<thead>
<tr>
<th>Severity Classification</th>
<th>Authorized Probability</th>
<th>Quantitative Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Catastrophic</strong></td>
<td><strong>Extremely Improbable</strong></td>
<td>&lt;10^{-9} per flight hour</td>
</tr>
<tr>
<td>Death, system or aircraft loss, permanent total disability.</td>
<td>So unlikely, not expected in system</td>
<td></td>
</tr>
<tr>
<td><strong>Hazardous</strong></td>
<td><strong>Extremely Remote</strong></td>
<td>10^{-7}-10^{-9} per flight hour</td>
</tr>
<tr>
<td>Severe injury or major aircraft or system damage.</td>
<td>Unlikely for item, may occur few in system</td>
<td></td>
</tr>
<tr>
<td><strong>Major</strong></td>
<td><strong>Remote</strong></td>
<td>10^{-5}-10^{-7} per flight hour</td>
</tr>
<tr>
<td>Minor injury or minor aircraft or system damage</td>
<td>Possible for each item, several for system</td>
<td></td>
</tr>
<tr>
<td><strong>Minor</strong></td>
<td><strong>Probable</strong></td>
<td>10^{-6}-10^{-5} per flight hour</td>
</tr>
<tr>
<td>Less than minor injury or aircraft or system damage</td>
<td>Likely to occur in lifetime of each system</td>
<td></td>
</tr>
<tr>
<td><strong>No Safety Effect</strong></td>
<td>/</td>
<td>/</td>
</tr>
</tbody>
</table>

Where the consequence of a failure is low, it is admitted to occur more frequently than when it is very critical. In order to meet these safety requirements, the aircraft must be designed to include additional mitigation, as incorporation of redundant or more reliable components or systems. In flight control system, the actuation systems are generally made redundant. The common solution is that [22]:

- The electric devices are triplex or quadruplex redundant as their reliability is a little low.
- The hydraulic devices are dual or triplex redundant.
- The mechanical devices have a higher reliability, so normally simplex or dual redundant.

Generally, for redundant actuation systems, the reliability of single channel can be written as:

\[ R(t) = e^{-\lambda t} \]  \hspace{1cm} (1.1)

where \( \lambda \) is the hazard rate and \( t \) is the service time.

A good reliability means the term \( \lambda t \) is very small and \( R \) is approximate to 1. Regarding to the experiences in aviation, improving the reliability of a single channel is very heavy and costly. Especially when the reliability is approaching to 1, the cost will increase exponentially [31].

In other side, for redundant actuation systems in which \( n \) channels are connected in parallel and work on active/active mode, the total reliability of system can be calculated as [32]:

\[ R_{\text{sys}}(t) = (1 - R_{\text{fail}})^n \]
\[ R = 1 - \prod_{i=1}^{n} (1 - R_i) \]  
\[ (1.2) \]

where \( R_i \) is the individual reliability of each channel.

For a redundant system of this kind, in case all channels have a constant reliability of 0.8: the dual redundant will have a reliability of 0.96 and the triplex redundant will be 0.992. This shows that the reliability of whole system is rapidly improved and the cost and weight are still kept being acceptable [22]. That is why the redundant design is widely employed in aviation.

Many redundant actuation configurations have been developed according to the requirements of applications. One major issue is ensuring the actuators to be safe for other systems in case of failure (fail safe). This plays a major role on selecting actuators and designing mechanical configurations. The typical types of fail safe actuators are summarized in Tab 1-3 [33].

<table>
<thead>
<tr>
<th>Types</th>
<th>Description</th>
<th>Example</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fail Frozen</td>
<td>Actuator is wished to be locked in failure position.</td>
<td>THSA</td>
</tr>
<tr>
<td>Fail Passive</td>
<td>Actuator is wished to be on damping or free mode.</td>
<td>Ailerons, Elevators</td>
</tr>
<tr>
<td>Fail Neutral</td>
<td>Actuator is wished to return to a defined position.</td>
<td>Engine inlet guide valve actuators</td>
</tr>
</tbody>
</table>

As shown in Tab 1-3, the EMAs can be chosen for designing the THSAs as they could be easily locked at the position after a jamming failure. About the ailerons, rudders and elevators, the actuators with hydraulic jack (SHAs and EHAs) are preferred as the jacks are improbable to be jammed, meanwhile the damping and free moving can be easily achieved by bypass valves after failure. Oppositely, EMAs introduce a new issue for these applications as the jammed-free problem is still not well solved.

Then on the basis of requirements in Tab 1-3, the existing redundant actuation configurations and their properties and applications are summarized in Tab 1-4 [14, 22]. The dual redundant designs are studied here, the triplex redundant is similar.
### Tab 1-4 Summary of redundant actuation configuration

<table>
<thead>
<tr>
<th>Configuration</th>
<th>Actuators</th>
<th>Properties</th>
<th>Application</th>
</tr>
</thead>
<tbody>
<tr>
<td>a) Parallel Force Summing</td>
<td>Channel 1: SHAs &amp; EHAs (future EMAs?)</td>
<td>All channels’ rods are connected to flight control surface. The final output is the summing of all channels’ forces. <strong>Advantage:</strong> structure is simple; suitable for all kinds of actuators; normally could be fail operation/fail safe. <strong>Disadvantage:</strong> there is force fighting between actuators on active/active mode.</td>
<td>The most common choice of primary flight control system for airliners, like A320, A380, B747 and B787.</td>
</tr>
<tr>
<td>b) Parallel Position Summing</td>
<td>Channel 1: SHAs (future EHAs &amp; EMAs?)</td>
<td>All channels’ rods are connected to a differential linkage and the final output is the mean position. <strong>Advantage:</strong> suitable for all kinds of actuator; there is no force fighting. <strong>Disadvantage:</strong> the position summing linkage is complex and not suitable for more than dual redundant; each individual actuator needs a larger stroke than the minimum allowable output stroke to accommodate the failure channel, this restricts its application to only small displacement situations.</td>
<td>Boeing B737 dual yaw damper [14].</td>
</tr>
<tr>
<td>A variation of a) Tandem Force Summing</td>
<td>Channel 1: SHAs</td>
<td>All channels drive a common rod and only actuators with hydraulic jack as final linear motion output device can be involved. <strong>Advantage:</strong> compact structure; there is no force fighting on A/S mode and very small on A/A mode. <strong>Disadvantage:</strong> too long for more than dual redundant.</td>
<td>A common choice in the primary system of military fighters, as the tail of F-16, ailerons, canard and flaperons of F-15.</td>
</tr>
<tr>
<td>A variation of b) Tandem Position Summing</td>
<td>Channel 1: EMAs</td>
<td>One EMA is nested inside another one. The nut envelop of inner screw is stiffly connected to the outer screw. The final output is the relative displacement of outer screw plus the relative displacement of inner screw. <strong>Advantage:</strong> there is no force fighting; one channel failure (jammed) can be compensated by the rest correctly working channel; compact structure. <strong>Disadvantage:</strong> each individual channel needs a larger stroke than the minimum allowable; the mechanization is complex.</td>
<td>HEAT/ACT for helicopter [34].</td>
</tr>
<tr>
<td>A variation of b) Speed Summing</td>
<td>Channel 1: EMAs</td>
<td>Two motors are connected to a speed summing gearbox. Its output drives the nut and screw. <strong>Advantage:</strong> one motor failure can be compensated by the rest correctly working one. <strong>Disadvantage:</strong> nut-screw jamming will lead to the whole system failure; reducer and clutch needed.</td>
<td>THSA of Boeing B787 [26].</td>
</tr>
</tbody>
</table>
As mentioned upper, the parallel force summing configuration is the most common choice for the primary flight control system on commercial airliners. Because in most cases they operate on active/active mode, the force fighting between actuators is serious and must be addressed. Two solutions have been proposed: one is scheduled maintenance, adjusting the devices to reduce the statics difference between channels; one is introducing force equalization control strategy to continually balance the mean and static force of all channels. The first solution is widely used in the hydro-mechanical actuation systems. In FBW systems, the second solution is preferred as the force equalization can be conveniently done by the FCC and meanwhile the maintenance cost is reduced.

While designing the force equalization control strategies, the segregation between channels should be concerned. The cross links between channels introduced by the force equalization should be limited in an admissible range to prevent from any failure to propagate from one channel to the other channels that could reduce the overall reliability.

For the redundant actuation systems only composed by SHAs, the common force equalization strategy is introducing a voted load pressure signal to compensate the position controllers of each channel [35]. However, for the redundant actuators composed by dissimilar technology actuators working on next generation airliners, the new force equalization control strategy is necessary to be developed. As EHAs are considered as the transitional form of electrically powered actuators to EMAs, the redundant actuators composed by one SHA and one EMA is selected to be studied in present thesis.

1.1.3 FBW Actuators

In this section, the current mainstream FBW actuators are presented in detail, including SHAs, EHAs, EMAs and EBHAs [36]. The operation function of actuators is introduced firstly. Then their properties are summarized and compared.

a) Servo-Hydraulic Actuator (SHA)

As shown in Fig 1-6, the typical structure of SHA is presented. The SHAs use hydraulic power throughout the flow. A servovalve is working as the modulating device between power source
Chapter 1 State of the Art and Test Bench

Fig 1-6 Schematic of SHA

and jack. The power modulation is controlling the variable hydraulic resistance in the flow path to jack by adjusting the servovalve spool opening. Inside the servovalve, an electro-hydraulic-mechanical amplifier which is signaled by current and powered by hydraulic is employed to control the spool opening with sufficient power. The current signal (normally 10-15mA) is generated by the FCC and relative control electronics. The FCC & control electronics, hydraulic servovalve, jack and sensors construct a closed position loop to control the rod extension, and by the way the surface deflection. The key component in this control loop is the servovalve which has decisive effect on system statics and dynamics behaviors. When the SHA is not active, the mode select valve (MSV) will force the actuator to operate on damping mode, the two chambers of the jack are bypassed.

b) Electro-Hydrostatic Actuator (EHA)

As shown in Fig 1-7, the EHAs involve two types of power, electric from power source to

Fig 1-7 Schematic and some application of EHA
modulation and hydraulic from modulation to transformer. The power modulation device is the motor drive electronics (MDE) which controls the brushless DC (BLDC) motor. A cascade controller is built for the motor with speed and current control loops. A fixed displacement pump is connected to the motor. Its flow is leaded to control the rod extension. The core of this actuation system is the motor and pump, which determine mainly the EHAs performance. Especially for high speed pump, the short service life has restricted its use in aviation. The response to failure is similar as SHAs. The mode select valve can force the two chambers of jack to bypass mode [38].

c) Electro Backup Hydrostatic Actuator (EBHA)

As shown in Fig 1-8, the EBHA could be considered as a combination of one SHA and one EHA which share a common jack. The EHA works as the backup of SHA and the operating mode is changed by a mode select valve. In case both the SHA and EHA are failed, the bypass valve will be de-energized to connect the two chambers. The benefit is achieving dissimilar redundancy with a so compact structure. As an excellent transition from hydraulic to electric, it has been employed in recent European applications.

d) Electro-Mechanical Actuator (EMA)

As shown in Fig 1-9, the biggest difference between EHA and EMA is the power type between power modulation and transformer, EHAs use hydraulics and EMAs use mechanics. While the motor transfers electric power to mechanic rotary motion, EHAs transfer the rotary mechanic power to hydraulic power and finally transfer it to linear motion; EMAs transfer the rotary
mechanic power to linear motion through gearbox and/or nut-screw, the middle hydraulic section is eliminated. So theoretically, the EMAs have a higher efficiency than EHAs. However, the reducer is often needed to save weight by using lighter electrical motor.

According to the motion transfer way, EMAs can be divided into two types: direct drive and gear drive [41], as shown in the upper right corner of Fig 1-9. In direct drive, the nut/screw is directly driven by the motor. The nut is stiffly connected to the motor rotor or stator, with the rotary of motor, the screw is driven to extend and retract. In gear drive, the nut/screw and motor are individually installed. A gearbox is used to connect motor and nut. Generally the direct drive type is more compact and lighter. The gear drive type is more maintenance friendly. The common used screw technique is ball-screw and roller-screw. The ball-screw has lower friction but with lower load capacity. The roller-screw has better load capacity but with higher friction.

On the basis of above study, the advantages and disadvantages of SHAs, EHAs and EMAs are summarized in Tab 1-5 [22, 42, 43, 44].
As summarized in Tab 1-5, SHAs are the most mature and attractive FBW actuators until now. For the time being, EHAs and EMAs are still not mature enough to totally replace the SHAs in primary flight control systems, at least in front line.

### 1.2 Introduction of Test Bench

For carrying out the research, a hybrid actuation test bench composed by one SHA and one EMA is employed. The test bench was built for evaluating the behaviors of one EMA prototype and placed in the hydraulic laboratory of ICA, INSA Toulouse, France, as shown in figure 1-10 [41, 45]. The test bench will be used to demonstrate the performance of redundant actuation
system in laboratory environment as the level 4 of technology readiness level (TRL) [46]. The force equalization control strategies to be proposed shall be validated on this test bench (In practice, due to the unrealistic backlash and no way to simulate external load, the validation could not be completely performed on it.).

Fig 1-10 Photograph and schematic of test bench

Mechanical Configuration

The test bench includes one industrial SHA and one prototype of EMA which combines a BLDC motor driving an inverted roller-screw. One mechanism with two round iron lumps in the end is used to represent the surface inertia, as shown in the bottom of Fig 1-10. It is connected to the actuators’ rod end by a leverage linkage. The actuators are fixed on the bench by two anchorages, which can simulate the linkage compliance to airframe and their value can be adjusted based on requirements. In addition two force sensors are installed on the rod end of actuators to measure their output forces, and therefore the amount of the force fighting.
The two actuators are fixed in symmetrical position, one push and one pull to position the load. Although not like the parallel arrangement in Tab 1-4, the system performances are quite similar. The results got on this test bench can be applied on the parallel arrangement redundant actuation system only with some adjustments over +/- signs. Meanwhile, this arrangement is more efficient on studying the single actuator’s property. In case one actuator is studied, the other one can be force controlled to simulate the airload.

The following table summarizes the main properties of test bench [47, 48]:

<table>
<thead>
<tr>
<th>SHA</th>
<th>EMA</th>
<th>Load</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stroke: 100mm</td>
<td>Stroke: 90mm</td>
<td>Equivalent Mass at rod level: 600Kg</td>
</tr>
<tr>
<td>Max Speed: 160mm/s</td>
<td>Max Speed: 200mm/s</td>
<td></td>
</tr>
<tr>
<td>Max Force: 26KN at a</td>
<td>Max Force: 50KN</td>
<td></td>
</tr>
<tr>
<td>supply pressure of 8.5MPa</td>
<td>Supply Voltage: 540V/26.8A</td>
<td></td>
</tr>
</tbody>
</table>

**Electric Architecture**

The schematic of electric system is presented in Fig 1-11:

![Control system schematic of test bench](image)

where SC1, 2, 3, 4 and 5 are signal conditioners.

The control of the redundant actuation system is done by a XPC Target computer [49], which is used to simulate the two FCCs. The output range of sensors is +/-10VDC. They are collected by the XPC. Two analog outputs whose ranges are also +/-10VDC are calculated by the XPC on the basis of the designed control laws and separately sent to the SHA servo valve voltage-
current amplifier and Parvex drive that is associated with the EMA motor. The current signal outputted by amplifier can control the servovalve opening. Meanwhile, the analog output from XPC to Parvex is transferred as the motor torque demand and a motor torque control loop is built within the Parvex drive [50].

The PC connected to XPC and Parvex is not inside the control loop. It is in charge of some offline programming, parameter setting and data collection. The control laws are designed in the Matlab/Simulink simulation environment on PC, and then downloaded to be executed on the XPC Target computer. Meanwhile the experiments results stored in XPC can be uploaded to PC. All the data transmission between PC and XPC is on the basis of TCP/IP protocol. And the data transmission between PC and Parvex is on the basis of Can-bus protocol. In addition, one motor management software PME is installed on PC to set the parameters of Parvex, by which the motor torque control properties are adjusted [50].

1.3 Performance Requirements

For the studied active/active position control redundant actuation system, the objective of thesis is designing the position controllers with force equalization features for SHA and EMA to ensure that the following performances can be obtained:

- Pursuit performance: the surface position magnitude ratio is greater than -3dB and the phase lag is lower than 45° for a 3Hz/1mm magnitude sine input;
- Rejection performance: the surface angle $\theta$ must show a static dependency to the airload corresponding to a closed loop static stiffness greater than $2\times10^8$N/m;
- Force equalization performance: the static force fighting (force difference between the actuators) should be lower than 4% of the rated force (50KN) and the dynamic force fighting should be fewer than 20% of this rated force;
- Segregation: the cross links between channels should be minimized and limited to ensure the immunity against failure propagation from one channel to another channel;
- Robustness: the system with force equalization should be robust for keeping the actuation performance in the presence of modeling error and unconsidered uncertainties;
- Complexity: the strategy needs short computation time and few numbers of sensors.
As mentioned in chapter 1, the objective of thesis is designing controllers for SHA and EMA to ensure the actuators’ position performance and the force equalization. Satisfying the position requirements is the basic and force equalization is the further objective. Therefore, the first step is designing position controllers for SHA and EMA. The design process is on the basis of the linear approach. The proposed position controllers are to be experimentally validated on the real test bench. Then, the force fighting between SHA and EMA with position controller involved is primarily pointed out on the linear prototype of the hybrid actuation system that is built in present step.

2.1 SHA Position Controller Design

This section aims at designing the position controller for SHA. The EMA is disconnected to not alter the load characteristics and the whole surface is driven by the SHA.

2.1.1 Linear Prototype of SHA

The schematic of SHA system is displayed in Fig 2-1.
where:

- $\theta_t$ Flight control surface rotary angle [°]
- $A_t$ Jack effective area [$m^2$]
- $B_s$ Jack viscous damping coefficient [$N\cdot s/m$]
- $C_a$ Air load on surface [N]
- $F_L$ Equivalent air load force on rod level [N]
- $I_{sv}$ Servovalve input current [A]
- $M_t$ Equivalent mass of surface [Kg]
- $P_1$ Pressure of jack chamber 1 [Pa]
- $P_2$ Pressure of jack chamber 2 [Pa]
- $P_s$ Supply pressure of hydraulic power [Pa]
- $P_r$ Return pressure of hydraulic power [Pa]
- $Q_{sv}$ Output Flow of Servovalve [m$^3$/s]
- $Q_{sv}$ Output Flow of Servovalve [m$^3$/s]
- $V_1$ Volume of jack chamber 1 [$m^3$]
- $V_2$ Volume of jack chamber 2 [$m^3$]
- $X_t$ Equivalent surface displacement [m]
- $X_s$ Extension of rod [m]

The system has two inputs: servovalve current input $I_{sv}$ and air load torque $C_a$. The output is the surface rotary angle $\theta_t$. As shown in Fig 2-1, the flight control surface is in rotary motion and SHA rod is in linear motion, in order to simplify the following research, an equivalent load displacement $X_t$ and an equivalent air load $F_L$ are introduced to represent the surface effect on SHA rod level. In small rod extension range, they can be considered as being proportional to the surface rotary angle $\theta_t$ and air load $C_a$ (neglecting the change in lever arm). So in following parts, the system output being controlled is the $X_t$.

At first, the system physical equations are obtained according to signaling and powering flow. The signaling flow is from servovalve current input to spool displacement (valve opening); the powering flow is from hydraulic power source to surface displacement in counteracting the air load.

The signaling flow from current to servovalve spool displacement is presented in equation 2.1. It is a function of supplied pressure and input current. The spool statics is mainly driven by the input current and the dynamics is influenced by the supplied pressure. Normally, the typical dynamics of servovalve is around 100Hz which is much higher than the jack position control dynamics only a few Hz, so the dynamics is always neglected for simplifying research.

$$X_{sv} = f_{sv}(P_s, I_{sv})$$

(2.1)

where:

- $X_{sv}$ Servovalve spool displacement [m]
- $f_{sv}$ Valve opening function

The hydraulic power is modulated by servovalve. The valve flow can be expressed as equation 2.2 [22]. It is influenced by the power supply, load pressure and valve opening. The first item
shows the flow/valve opening characteristics and the second item shows the flow/pressure characteristics.

\[ Q_{sv} = K_{sq} X_{sv} \sqrt{1 - \frac{P_f}{P_s - P_r}} \text{sgn} \left( X_{sv} \right) \text{sgn} \left( 1 - \frac{P_f}{P_s - P_r} \text{sgn} \left( X_{sv} \right) \right) - K_{sc} P_f \] (2.2)

where:

- \( K_{sq} \) Servovalve flow/opening gain at null pressure drop \([m^2/s]\)
- \( K_{sc} \) Servovalve flow/pressure gain \([(m^3/s)/Pa]\)
- \( P_f \) Load pressure \( P_f=P_{1}-P_{2} [Pa] \)

The flow delivered by the servovalve is consumed by the jack functional need (rod velocity) and the parasitic effects (hydraulic compression and leakage). According to this, the flow to jack chamber 1 \( Q_{j1} \) can be expressed as:

\[ Q_{j1} = A_s \dot{X}_s + \frac{V_1 + A_s X_s}{E_y} \dot{P}_1 + K_{sc} P_f \] (2.3)

where:

- \( K_{sc} \) Jack leakage coefficient \([(m^3/s)/Pa]\)
- \( E_y \) Oil effective bulk modulus \([N/m^2]\)

Same to the chamber 1, the flow from the jack chamber 2 \( Q_{j2} \) can be expressed as:

\[ Q_{j2} = A_s \dot{X}_s - \frac{V_2 - A_s X_s}{E_y} \dot{P}_2 + K_{sc} P_f \] (2.4)

With the pressure difference between two chambers, the jack output force \( F_s \) can be written as the summing of hydrostatic force and friction:

\[ F_s = A_s P_f - F_{jf} \] (2.5)

where:

- \( F_s \) SHA jack output force \([N]\)
- \( F_{jf} \) SHA jack friction \([N]\)

In the end, according to Newton Second Law, the surface dynamic equation can be expressed as equation 2.6. The jack rod mass \( M_s \) is so small in comparison with the load equivalent mass \( M_t \) that it is neglected.

\[ F_s - F_L = M_s \ddot{X}_s \] (2.6)

Following this, the upper physical equations are linearized to get a linear approach of SHA to
help designing the position controller.

In equation 2.1, the influence of supply pressure is neglected. The valve opening is linearized as a proportional section to current input.

\[ X_{sv} = K_{sv} I_{sv} \]  

(2.7)

where:

- \( K_{sv} \) Servovalve opening/current gain \([\text{m/A}]\)

In equation 2.2, the servovalve flow is linearized around null opening and null load pressure (in other operation points, the gains will change but the magnitude is small \([53]\)) leading to:

\[ Q_{sv} = K_{qs} X_{sv} - K_{sc} P_f \]  

(2.8)

In equation 2.3 and 2.4, the jack flow continuity equation is linearized around central position. The chambers volume \(V_1\) and \(V_2\) are considered as identical and half of the total jack volume.

\[ Q_{sv} = \frac{Q_{12} + Q_{22}}{2} = A_s \ddot{X}_s + \frac{V_s}{4E_y} \dot{P}_f + K_{sc} P_f \]  

(2.9)

where:

- \( V_t \) Jack total effective volume \(V_t = V_1 + V_2\) \([\text{m}^3]\)

In equation 2.5, the jack friction is linearized as a viscous section. The Coulomb and Stribeck effects are neglected, this leads to:

\[ F_s = A_s P_f - B_s \dot{X}_s \]  

(2.10)

In addition, as the stiffness between rod and surface is very high, its compression is much smaller in comparison with the rod displacement, its influence is neglected for now. The rod displacement is considered as equaling to the load displacement.

\[ X_s = X_r \]  

(2.11)

With this, the Laplace transformation is done on upper equations to get a linear prototype of the SHA, as displayed in Fig2-2.
With this block diagram, the transfer function of SHA can be got as:

\[
X_i(s) = \frac{K_{sg} I_{in}(s) - \left(\frac{V_t}{4E_c A_s^2} s + \frac{K_c}{A_s^2}\right) F_L(s)}{s \left[ \frac{V_t M_t}{4E_c A_s^2} s^2 + \left( \frac{M_t K_c}{A_s^2} + \frac{V_t B_s}{4E_c A_s^2} \right) s + \left( \frac{B_s K_c}{A_s^2} + 1 \right) \right]} 
\]

(2.12)

where:

- \(K_{sg}\): Servovalve flow/current gain \(K_{sg} = K_{sv} K_{sq}\) \([m^3/(A\cdot s)]\)
- \(K_c\): SHA total flow/pressure gain \(K_c = K_m + K_{ac}\) \([m^2/(Pa\cdot s)]\)

For further study, the parameters in upper transfer function are valued according to the SHA on test bench, as summarized in Tab 2-1.

### Tab 2-1 Parameter values of SHA linear prototype

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Value</th>
<th>Unit</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>(A_t)</td>
<td>3.1×10^{-3}</td>
<td>m^2</td>
<td>Accurate Value</td>
</tr>
<tr>
<td>(K_{sq})</td>
<td>8.3×10^{-3}</td>
<td>m^3/(A\cdot s)</td>
<td>Linearized</td>
</tr>
<tr>
<td>(V_t)</td>
<td>3.1×10^{-4}</td>
<td>m^3</td>
<td>Accurate Value</td>
</tr>
<tr>
<td>(E_y)</td>
<td>8×10^8</td>
<td>Pa</td>
<td>Assumed Value</td>
</tr>
<tr>
<td>(K_c)</td>
<td>7.94×10^{-13}</td>
<td>m^2/(Pa\cdot s)</td>
<td>Linearized</td>
</tr>
<tr>
<td>(M_t)</td>
<td>600</td>
<td>Kg</td>
<td>Accurate Value</td>
</tr>
<tr>
<td>(B_s)</td>
<td>2300</td>
<td>N/(m/s)</td>
<td>Linearized</td>
</tr>
</tbody>
</table>

Then the canonical form of transfer function can be presented as:

\[
X_i = \frac{a I_{in} - \beta(1 + \tau_s s) F_L}{s \left( 1 + \frac{2\zeta}{\omega_s} s + \frac{1}{\omega_s^2} s^2 \right)}
\]

(2.13)
where:
\( \alpha \)  SHA open loop pursuit gain \([m/A]\)
\( \beta \)  SHA open loop rejection gain \([m/N]\)
\( \tau_s \)  SHA rejection time constant \([s]\)
\( \xi_s \)  SHA open loop dimensionless damping factor
\( \omega_s \)  SHA open loop natural frequency \([Hz]\)

and,
\[
\alpha = \frac{A_2 K_{se}}{B_5 K_c + A_1^2} \approx \frac{K_{se}}{A_1} = 2.677 \text{[m/A]} \tag{2.14}
\]
\[
\beta = \frac{K_c}{B_5 K_c + A_1^2} \approx \frac{K_c}{A_1^2} = 8.26 \times 10^{-6} \text{[m/N]} \tag{2.15}
\]
\[
\tau_s = \frac{V_t}{4E_s K_c} \approx 0.122 \text{[s]} \tag{2.16}
\]
\[
\omega_s = \sqrt{\frac{4E_s (B_5 K_c + A_1^2)}{V_t M_t}} \approx \sqrt{\frac{4E_s A_1^2}{V_t M_t}} = 40.66 \text{[rad/s]} = 64.7 \text{[Hz]} \tag{2.17}
\]
\[
\xi_s = \frac{4E_s M_t K_c + V_t B_s}{2\sqrt{4E_s (B_5 K_c + A_1^2)V_t M_t}} \approx \frac{4E_s M_t K_c + V_t B_s}{2\sqrt{4E_s A_1^2 V_t M_t}} = 0.0418 \tag{2.18}
\]

As shown in equation 2.14, the SHA open loop pursuit gain \( \alpha \) is mainly fixed by the servovalve flow/current gain, so the following designed position controller is going to regulate this value. As indicated in equation 2.15, the rejection performance to external load and disturbance at open loop is driven by the SHA total flow/pressure gain. In equation 2.17, the SHA natural frequency \( \omega_s = 64.7 \text{Hz} \), which is fixed by the fluid compression and load mass. This frequency will change under different connecting conditions. While the EMA is connected, SHA and EMA operate on active/active mode, if the load mass is assumed equally shared, this frequency will be 2 square root times of the one with only SHA connected. In addition, as in equation 2.18, the SHA open loop damping factor \( \xi_s = 0.0418 \). It is so small that the SHA open loop stability is poor and needs to be paid more attention during position controller design.

Moreover, as shown in equation 2.12, the SHA open loop linear model displays an integral. This means even with a pure P controller, the position static error will be null in an ideal situation. But in fact, due to the Coulomb friction of jack and servovalve hysteresis caused by leakage around null opening, the real position static error will not be zero. So generally, the
linear model has good performance on setting controllers for stability but not representative enough for static accuracy.

In addition, of course, the servovalve dynamics (about 80Hz) can be added as a well damped second-order section in the present study without difficulty. It is not mentioned here for simplification but its influence cannot be totally ignored: the lowest hydro-mechanical mode (centered rod) is not much lower than that of the servovalve.

2.1.2 Position Controller Design

As concluded in upper part, the position controller is designed to adjust the forward pursuit gain. The P controller is the most common choice for this kind of task, as displayed in Fig 2-3.

\[ X_r(s) = \frac{1}{4E_A K_{sg} K_{sb}} s^3 + \frac{V_M}{4E_A K_{sg} K_{sb}} s^2 + \frac{B K_s + A^2}{A K_{sg} K_{sb}} s + 1 \]

where \( K_{sb} \) is the P controller, unit [A/m].

After regulation, the SHA closed loop transfer function can be presented as:

\[ X_c(s) = \frac{K_c}{A K_{sg} K_{sb}} \left( \frac{V_x}{4E_A K_c} s + 1 \right) F_l(s) \]

On the basis of this transfer function, the P control gain \( K_{sb} \) is calculated to ensure satisfying the requirements on stability, pursuit and rejection:

<table>
<thead>
<tr>
<th>Stability</th>
<th>According to Routh Stability Criterion: ( 0&lt;K_{sb}&lt;4.51A/m )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pursuit</td>
<td>( \geq-3dB/-45^\circ ) for 1mm/3Hz Sine Input</td>
</tr>
<tr>
<td>Rejection</td>
<td>( \geq 2\times 10^8N/m )</td>
</tr>
</tbody>
</table>

According to Bode graph of transfer function \( X_c/X_r \):

\( K_{sb}\geq 7.1A/m \)

According to the statics of transfer function \( F_l/X_r \):

\( K_{sb}\geq 6.2A/m \)

On the basis of upper results, it is impossible to find a \( K_{sb} \) satisfying all the requirements at the same time. The stability request and pursuit dynamics request are inharmonic. This can be found from the Bode graph of equation 2.19 with different \( K_{sb} \) value, as shown in Fig 2-4.
As indicated in Fig 2-4, while $K_{sb}=4A/m$, SHA is stable but the pursuit dynamics and rejection are very poor and far away from the requirements; while $K_{sb}=7.1A/m$, the critical value of the SHA pursuit dynamics, as the green lines, the pursuit performance is satisfied but the system becomes unstable; while $K_{sb}=10A/m$, as the red lines, the system is still unstable. The badly damped closed loop resonance occurs at 64.7Hz which is close to the SHA natural frequency. The common solution for damping this resonance in SHA control is inserting a notch filter. As shown in equation 2.20.

$$F_N = \frac{1}{s^2} \frac{\frac{2\xi_d}{\omega_n} s + 1}{s^2 + \frac{2\xi_n}{\omega_n} s + 1} \quad (2.20)$$

The $\omega_n$ is the notch frequency of filter. In order to damp the hydro-mechanical resonance, it is set to 407rad/s (64.7Hz). The $\xi_d$ and $\xi_n$ are used to set the damping ratio, $\xi_n$ is the reference value and $\xi_d$ is adjusted to set the damping effect, typically they are set as $\xi_d=0.01$ and $\xi_n=0.7$.

The improved SHA position controller is displayed in Fig 2-5.
Chapter 2 Actuator Position Controller Design

Under this control strategy (common in flight control actuation), while the notch frequency is assumed exactly equaling to the hydraulic natural frequency, the SHA closed loop resonance shall be well damped. The effect of this notch filter is assessed through Bode graph:

As shown in Fig 2-6, with the notch filter, the hydraulic resonance at 64.7Hz is well damped. The system is stable under the two tested value $K_{sb}=10\text{A/m}$ and $K_{sb}=15\text{A/m}$. Meanwhile, the system position pursuit dynamics and rejection can also meet the requirements.

On the basis of equation 2.19, while the system stability is ensured, the bigger the $K_{sb}$ is, the better the pursuit dynamics and rejection. But higher dynamics needs more supply power. While the power demand is over the limit, the dynamics shall be not realized and the speed is saturated. So a balance value is selected as $K_{sb}=15\text{A/m}$.

2.1.3 Validation of SHA Position Controller

As mentioned above, a P plus notch filter position controller is proposed for the SHA. It has
been validated in frequency domain on the basis of linear approach. Now it is introduced into SHA for validating in time domain through experiments. Three experiments are done to assess the system stability, position pursuit dynamics and rejection.

- First: to test pursuit dynamics, the EMA is disconnected; the position demand of SHA is a 1mm/3Hz sine signal.
- Second: to test system stability, the EMA is disconnected; the position demand of SHA is a 2mm step signal at 5s.
- Third: to test rejection, the EMA is connected and force controlled to load the SHA; the force demand of the EMA is slowly changed as a ramp signal from -20KN to 20KN; the position demand of SHA is null.

The experiment results are displayed in Fig 2-7.

![Fig 2-7 Position controller performance on real SHA](image)

The P plus notch filter controller shows a good performance on meeting the requirements of stability, position pursuit dynamics and rejection. The closed loop static rejection stiffness is close to $5 \times 10^8 \text{N/m}$ which is much higher than the required value.

Meanwhile, the simulation results on the basis of SHA linear prototype of equation 2.19 are
displayed to compare with the experimental results. It is found that the linear model does not represent well the real behaviors. As for the step position demand experiment, the simulated response is smooth and without static error, but the experimental result has some vibrations in transient period and the static error is not zero. This is caused by the non modeled effects and the nonlinearities existing on test bench, such as the backlash between SHA rod and load, servovalve dynamics, flow around null valve opening, and so on. This proves the need of a realistic nonlinear virtual test bench to enable the model based design and assessment of the force equalization.

2.2 EMA Position Controller Design

This section aims at designing position controller for the EMA. The SHA is disconnected for the same reason as explained above.

2.2.1 Linear Prototype of EMA

The schematic of EMA is displayed in Fig 2-8. It includes a BLDC motor, roller-screw and flight control surface. The EMA system has two inputs: motor torque demand $T_d$ and air load torque $C_a$. The output is the surface rotary angle $\theta$. Same as SHA, for simplifying the calculation, the $X_e$ and $F_L$ are used to present the surface load effects on EMA rod level.

![Fig 2-8 Schematic of EMA](image)

where:

- $B_e$: Roller-screw viscous damping [N·m/(rad/s)]
- $T_d$: BLDC motor torque demand [N·m]
- $J_m$: Total rotary inertia in motor side [Kg·m²]
- $X_e$: EMA rod displacement [m]
According to Fig 2-8, the system physical equations are obtained following the signaling flow and powering flow. The signaling flow is from motor torque demand to Parvex motor power drive (MPD). A motor torque control loop is implemented within the Parvex MPD through controlling the motor current. As indicated by the supplier, the dynamics of this torque loop can be represented as a second-order filter with a natural frequency of $\omega_m=600\text{Hz}$ and a dimensionless damping factor $\xi_m=0.707$ [50]. As this dynamics is much higher than that of the position loop domain which is only a few Hz, its influence can be neglected, leading to:

$$T_m = T_d \quad (2.21)$$

where:

- $T_m$: Motor electrical output torque [$N\cdot\text{m}$]
- $T_d$: Torque demand [$N\cdot\text{m}$]

Then the torque balance at motor shaft can be written as:

$$T_e = T_m - T_f - J_m \omega_e \quad (2.22)$$

where:

- $T_e$: Mechanical output torque [$N\cdot\text{m}$]
- $T_f$: Overall mechanical friction torque [$N\cdot\text{m}$]
- $J_m$: Motor inertia [$\text{kg}\cdot\text{m}^2$]
- $\omega_e$: Motor angular velocity [rad/s]

As shown in equation 2.22, the power generated by motor is partly consumed by inertia and mechanical friction effects. The friction comes from the mechanical transmission elements, as the bearings and nut-screw. It is a very complex phenomenon including many nonlinearities. The common mechanical friction model includes the Coulomb, Stribeck and viscous effects. In order to represent the friction in a linear approach, it is linearized as a pure viscous term:

$$T_f = B_e \omega_e \quad (2.23)$$

Then through the roller-screw, the rotational power is transferred into translational form:

$$\dot{X}_e = \frac{l}{2\pi} \omega_e \quad (2.24)$$

$$F_e = \frac{2\pi}{l} T_e \quad (2.25)$$

where:

- $F_e$: EMA output force [$N$]
- $l$: Lead of roller-screw [$m$]
Finally, according to *Newton Second Law*, the load dynamic equation can be written as:

\[ F_e - F_L = M_j \ddot{X}_e \]  \hspace{1cm} (2.26)

Same to SHA, in EMA, the stiffness between screw and load is also neglected, leading to \( X_e = X_t \).

After that, the *Laplace* transformation of upper equations is done to get the linear prototype of EMA, as displayed in Fig 2-9.

On the basis of this block diagram, the transfer function of EMA can be written as:

\[ X_i(s) = \frac{\frac{2\pi}{l} T_d(s) - F_L(s)}{s \left[ M_j + \frac{4\pi^2}{l^2} J_m \right] s + \frac{4\pi^2}{l^2} B_e} \]  \hspace{1cm} (2.27)

The parameters in upper transfer function are valued regarding to the EMA on test bench, as summarized in Tab 2-3.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Value</th>
<th>Unit</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>( l )</td>
<td>( 3 \times 10^{-3} ) m</td>
<td>Accurate Value</td>
<td></td>
</tr>
<tr>
<td>( J_m )</td>
<td>( 9.2 \times 10^{-3} ) Kg·m²</td>
<td>Approximate Value</td>
<td></td>
</tr>
<tr>
<td>( B_e )</td>
<td>0.053</td>
<td>N·m/(rad/s)</td>
<td>Linearized</td>
</tr>
</tbody>
</table>

The canonical form of transfer function can be expressed as:

\[ X_i = \frac{\alpha_1 T_d - \beta_1 F_L}{s[\tau_e s + 1]} \]  \hspace{1cm} (2.28)

where:
- \( \alpha_1 \) \hspace{0.5cm} EMA open loop pursuit gain \([m/(A \cdot s)]\)
- \( \beta_1 \) \hspace{0.5cm} EMA open loop rejection gain \([m/(N \cdot s)]\)
- \( \tau_e \) \hspace{0.5cm} EMA electro-mechanical time constant \([s]\)
Chapter 2 Actuator Position Controller Design

and:

\[ \alpha_i = \frac{l}{2\pi B_c} = 0.009 [1/N] \] \hspace{1cm} (2.29)

\[ \beta_i = \frac{l^2}{4\pi^2 B_c} = 4.302 \times 10^{-6} [m/N] \] \hspace{1cm} (2.30)

\[ \tau_i = \frac{M_l l^2}{4\pi^2 B_c} + \frac{J_m}{B_c} = 0.1762 [s] \] \hspace{1cm} (2.31)

\[ M_l + \frac{4\pi^2}{l^2} J_m = 600 + 40356 = 40956 [Kg] \] \hspace{1cm} (2.32)

The open loop EMA is a second order system including one integral. Similar to the SHA, under
the effect of this integral, the position static error of EMA linear prototype in pursuit case is
theoretically null while a P controller is introduced. But in practice, mainly due to the large
amount of Coulomb friction of roller-screw, the EMA static error is not null.

As indicated in equation 2.30, in comparison with SHA, the EMA open loop rejection is a little
poor only with \( 4.3 \times 10^{-6} [m/N] \). Meanwhile, the EMA rotary inertia is so huge that it represents
several ten times of the surface mass when it is reflected on the rod level, as indicated in
equation 2.32. This strongly alters the EMA dynamics. With the effect of this inertia, the EMA
natural dynamics is only 6rad/s (\( \approx 1 \text{Hz} \)). The poor natural dynamics and rejection means a
large regulation control gain is needed during designing the position controller to meet the
closed loop requirements.

2.2.2 Position Controller Design

On the basis of above analysis, a P controller is proposed for the EMA to start with, as shown
in Fig 2-10.

\[ K_{eb} \]

\[ T_d \]

\[ \text{EMA} \]

\[ F_L \]

\[ X_i \]

\[ X_r \]

\[ \text{Fig 2-10} \text{ P controller for EMA} \]

where \( K_{eb} \) is the P control gain, unit [N].
Chapter 2 Actuator Position Controller Design

The transfer function of EMA with the P position controller can be expressed as:

\[
X_r(s) = \frac{X_r(s) - \frac{l}{2\pi K_{eb}} F_L(s)}{\left(\frac{M l}{K_{eb}} + \frac{2\pi J_m}{K_{eb}}\right) s^2 + \frac{2\pi B}{K_{eb}} s + 1} = \frac{X_r - \frac{4.775 \times 10^{-4}}{K_{eb}} F_L}{\frac{19.555}{K_{eb}} s^2 + \frac{111}{K_{eb}} s + 1}
\]

(2.33)

As shown in equation 2.33, the closed loop EMA is a second order system. According to Routh Stability Criterion, while \( K_{eb} \) is greater than zero, EMA is always stable. So the system stability will be not the key for valuing \( K_{eb} \). Then, in order to meet the rejection requirement, \( K_{eb} \) needs to be greater than 95500[N]. But while \( K_{eb} \) equals to this value, the system position dynamics is about a natural frequency of 70rad/s and a dimensionless damping factor of 0.041. The damping factor is so small that the system position response will display large overshoot and vibration. Meanwhile, according to calculations, the greater the \( K_{eb} \) the smaller the damping factor. For addressing this kind of problem, the common solution is introducing a velocity feedback, as displayed in Fig 2-11.

![Fig 2-11 Cascade controller for EMA](image)

where the control parameters are the position gain \( K_{eb} \) [1/s] and the velocity gain \( K_{ec} \) [N·s].

The velocity feedback normally uses the motor rotary speed \( \omega_e \) as this signal already exists for the motor control. In our system, because the motor shaft is considered as being rigidly connected to the surface, as shown in equation 2.24, the rod velocity \( \dot{X}_r \) has the same effect as \( \omega_e \). So in order to simplify the calculation, the rod velocity \( \dot{X}_r \) is used in following part.

The improved position controller is a cascade controller with an inner velocity control loop. In fact, if the motor torque control loop is also considered, this cascade controller shall have 3 stages. The motor torque loop is the 2nd inner loop of outer position loop.

After that, the transfer function of EMA under cascade controller can be written as:
As shown in equation 2.34, now two control parameters \( K_{eb} \) and \( K_{ec} \) can be used to set the two canonical parameters: EMA closed loop natural frequency \( \omega \) and damping factor \( \xi \). Moreover, the EMA static rejection stiffness is coupled with the natural frequency. So in order to meet the rejection requirement at the same time, the adjusting objective is set as \( \omega = 70 \text{ rad/s} \) and \( \xi = 1.0 \), leading to:

\[
\begin{align*}
\omega &= \sqrt{\frac{K_{eb}K_{cc}}{19.555}} = 70 \text{[rad/s]} \\
\xi &= 35 \times \left( \frac{111}{K_{eb}K_{cc} + \frac{1}{K_{eb}}} \right) = 1.0
\end{align*}
\]

Then the above proposed cascade controller is assessed in the frequency domain through the Bode graph of equation 2.34, as shown in Fig 2-12.
As shown in upper Bode graph, based on the linear prototype of EMA, the proposed cascade controller shows a good performance in frequency domain meeting all the requirements of stability, pursuit dynamics and rejection.

In addition, the integral control conventionally used in aerospace applications involving the cascade controller is not considered here. It is removed for two reasons: first, the I action causes strong nonlinearities and introduces limit cycles; and second, the I action can disturb the force equalization strategy.

### 2.2.3 Validation of EMA Position Controller

After being pre-validated in frequency domain, the cascade controller is to be experimentally validated in time domain on the EMA that is involved on the test bench. Same as SHA, three experiments are done to evaluate the system stability, pursuit dynamics and rejection.

- **First:** to test pursuit dynamics, the SHA is disconnected; the position demand of EMA is a 1mm/3Hz sine signal.
- **Second:** to test system stability, the SHA is disconnected; the position demand of EMA is a 4mm step signal at 7.5s.
- **Third:** to test rejection, the SHA is connected and force controlled to simulate the external air load on EMA; the force demand of SHA is a slowly changed ramp signal from -20KN to 20KN; the position demand of EMA is null.

The experimental results are shown in Fig 2-13.

It is observed that the EMA stability, position pursuit dynamics and rejection can all meet the requirements. The closed loop static rejection stiffness is close to $2 \times 10^8$ N/m which equals to the critical value.

Similar to SHA, the simulations on the basis of EMA linear prototype are also done to compare with the experimental results. It has a good representation of the dynamic behaviors, but the system statics is still not well represented. Like the strong nonlinearities of EMA rejection, as shown in the right graph of Fig 2-13, this is mainly due to the large Coulomb friction of roller-screw which is not considered in the controller design.
Moreover, the EMA pursuit curve to step position demand is very smooth in comparison with the SHA. It is close to the simulation results of linear prototype. This shows a smaller backlash between EMA rod and surface than that of SHA.

### 2.3 Primarily Study of Force Fighting on Linear Model

Now, the position controllers of the SHA and EMA have been designed and validated on test bench individually. In this section, the linear prototype of the SHA and EMA built for designing position controller in above part will be combined to construct a complete linear prototype of the hybrid actuation system. The force fighting between two channels shall be primarily studied and pointed out with this linear prototype. Moreover, the large backlash on test bench makes the real tests unrealistic to display the force fighting in aerospace conditions.

#### 2.3.1 Hybrid Linear Model

While combining the linear models of SHA and EMA together, the transmission compliances between actuators and load are introduced. In former analysis of single actuator, they are not considered as their compressions are so small in comparison with the rod displacement. They have less influence on the single actuator performance. But for the hybrid actuation system,
especially for the study of force fighting, they play a very important role, which requires their
effects to be introduced. In addition, in response to the causality issue, the rod masses are
introduced to connect the actuators and load. In former study, they are considered with the
load mass together. The schematic of this hybrid actuation system is shown in Fig 2-14.

Fig 2-14 Schematic of active/active redundant actuation system

where:

- \( M_{SHA} \) SHA rod mass, typical value 10\([\text{Kg}]\)
- \( M_{EMA} \) EMA rod mass, typical value 15\([\text{Kg}]\)
- \( S_{st} \) Transmission stiffness from SHA to load
  typical value 4.0×10^{8}\([\text{N/m}]\)
- \( S_{et} \) Transmission stiffness from EMA to load
  typical value 4.0×10^{8}\([\text{N/m}]\)

In this approach, the actuator output forces are calculated from the transmission stiffnesses
model as:

\[
\begin{align*}
F_s &= (X_s - X_i)S_{st} \\
F_e &= (X_e - X_i)S_{et}
\end{align*}
\]

(2.36)

Then the load dynamic equation can be written as:

\[
F_s + F_e - F_L = M_j \ddot{X}_j
\]

(2.37)

According to the equations 2.36 and 2.37, the block diagram of this hybrid actuation system is
got, as displayed in Fig 2-15.

Fig 2-15 Block diagram of hybrid actuation system
After that, the transfer functions of the hybrid actuation system are calculated.

The SHA position transfer function, according to equation 2.19:

\[ X_s = \frac{X_r - G_2(s)F_r}{G_1(s)} \]  \hspace{1cm} (2.38)

where:

\[ G_1(s) = \frac{V_i M_i}{4E_y A K_{sg} K_{sb} F_N} s^3 + \frac{4E_y M_i K_c + V_i B_i}{4E_y A K_{sg} K_{sb} F_N} s^2 + \frac{B_i K_c + A_i^2}{4E_y K_c} s + 1 \]

\[ G_2(s) = \frac{K_c}{A_i K_{sg} K_{sb} F_N} \left( \frac{V_i}{4E_i K_c} s + 1 \right) \]

The EMA position transfer function, according to equation 2.34:

\[ X_e = \frac{X_r - H_2(s)F_r}{H_1(s)} \]  \hspace{1cm} (2.39)

where:

\[ H_1(s) = \left( \frac{M_i l}{K_{eb} K_{ec} 2\pi} + \frac{2\pi J_m}{K_{eb} K_{ec} l} \right) s^2 + \left( \frac{2\pi B_e}{K_{eb} K_{ec} l} + \frac{1}{K_{eb}} \right) s + 1 \]

\[ H_2(s) = \frac{l}{2\pi K_{eb} K_{ec}} \]

Then according to equations 2.38 and 2.39, the hybrid actuation system can be presented as:

Based on this, the load displacement can be expressed as:

\[ X_l = \frac{\left[ S_{\alpha} (H_1 + H_2 S_{\alpha}) + S_{\alpha} (G_1 + G_2 S_{\alpha}) \right] X_r - (G_1 + G_2 S_{\alpha}) (H_1 + H_2 S_{\alpha}) F_l}{(G_1 + G_2 S_{\alpha}) (H_1 + H_2 S_{\alpha}) M_1 s^3 + G_1 S_{\alpha} (H_1 + H_2 S_{\alpha}) + H_1 S_{\alpha} (G_1 + G_2 S_{\alpha})} \]  \hspace{1cm} (2.40)
Chapter 2 Actuator Position Controller Design

About the force fighting between SHA and EMA, we propose using a dimensionless indicator $\gamma_d$ as the ratio of the force difference to the rated force:

$$\gamma_d = \frac{F_s - F_e}{F_{\text{rated}}}$$  \hspace{1cm} (2.41)

As the rated force $F_{\text{rated}}$ is constant, for simplifying the expressions, the force fighting can be written as:

$$\gamma = \gamma_d F_{\text{rated}} = \frac{F_s - F_e}{F_{\text{rated}}} = \frac{\left[ S_p \left( H_1 + H_2 S_{\text{et}} \right) - S_{\text{et}} \left( G_1 + G_2 S_{\text{et}} \right) \right] M_s s^2 + 2H S_{\text{et}} S_{\text{et}} - 2G_1 S_{\text{et}} S_{\text{et}}}{(G_1 + G_2 S_{\text{et}}) \left( H_1 + H_2 S_{\text{et}} \right) M_s s^2 + G_1 S_{\text{et}} \left( H_1 + H_2 S_{\text{et}} \right) + H_1 S_{\text{et}} \left( G_1 + G_2 S_{\text{et}} \right)} X_r \; (2.42)$$

According to equations 2.40 and 2.42, the position pursuit static error and static force fighting caused by position demand $X_r$ are always null no matter the values of stiffnesses $S_{\text{et}}$ and $S_{\text{et}}$. But this is the result of ideal case. In fact, on real test bench, due to the offset of sensors, DA transfer devices and AD transfer devices, servovalve leakage at null opening, Coulomb friction, etc, the actuators position static error and static force fighting in pursuit case are not null. That is why the proposed force equalization strategy should present a good robustness.

The load static rejection stiffness can be expressed as:

$$S_\text{static} = \frac{S_p \left( 1 + \frac{S_{\text{et}}}{S_{\text{sp}}} \right) + S_{\text{et}} \left( 1 + \frac{S_{\text{et}}}{S_{\text{sp}}} \right)}{\left( 1 + \frac{S_{\text{et}}}{S_{\text{sp}}} \right) \left( 1 + \frac{S_{\text{et}}}{S_{\text{sp}}} \right)} = 3.53 \times 10^8 [N/m]$$  \hspace{1cm} (2.43)

where:

- $S_{\text{sp}}$ is the SHA closed loop stiffness at low frequency domain [$N/m$]:

$$S_{\text{sp}} = \frac{A K_{\text{rs}} K_{\text{sb}}}{K_c}$$

- $S_{\text{et}}$ is the EMA closed loop stiffness at low frequency domain [$N/m$]:

$$S_{\text{et}} = \frac{2\pi K_{ee} K_{\text{et}}}{l}$$

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As shown in equation 2.43, the rejection of hybrid actuation system can meet the requirement. But the value is greatly influenced by the transmission stiffness. While the stiffness is too low, the rejection requirement shall be very difficult to meet.

The static force fighting caused by external load $F_L$ can be expressed as:

$$\gamma_{	ext{off}} = \frac{S_{st} \left( 1 + \frac{S_{et}}{S_{sp}} \right) - S_{et} \left( 1 + \frac{S_{st}}{S_{sp}} \right)}{S_{st} \left( 1 + \frac{S_{et}}{S_{sp}} \right) + S_{et} \left( 1 + \frac{S_{st}}{S_{sp}} \right)} F_L = 0.231 \times F_L$$

(2.44)

It is shown that the static force fighting in rejection condition is mainly driven by the actuator closed loop static stiffness and the transmission stiffness. With the above designed position controllers, while the external force $F_L$ is large enough, the static force fighting cannot meet the requirements. But this can be improved by resetting the position control parameters if it is consistent with the position performance requirements.

Then, as the dynamic performances of hybrid actuation system are too complex to be studied formally, it will be analyzed through numerical simulations.

2.3.2 Simulation of Force Fighting

For running simulation, the linear hybrid actuation system is represented in Matlab/Simulink, as shown in Fig 2-17.

![Fig 2-17 Prototype of hybrid actuation system built in Matlab/Simulink](image)

Then some simulations are run to show the dynamics of hybrid actuation system.
Chapter 2 Actuator Position Controller Design

a) Position demand step & external force step

This simulation is run to point out the dynamic force fighting under pursuit condition. The input is a 1mm position demand step (2% of full stroke) applied at 0.1s followed by a 10KN external force step (20% of rated force) occurring at 1s.

Two reasons for choosing 1mm as position demand magnitude:

- to have sufficiently high enough control signals to remove the influence of static offset;
- to not reach the saturation nonlinearities.

![Position Pursuit and Load Rejection](image)

**Fig 2-18** Simulation results of hybrid system to step input

The system pursuit performance to step position demand is shown in the left part of Fig 2-18. Under the same input $X_r$, as the different pursuit dynamics, SHA and EMA do not output the exactly same position. The dynamics of SHA is better, so it is faster than the EMA. According to equation 2.36, different $X_s$ and $X_e$ leads to different actuator output force $F_s$ and $F_e$, so the SHA is always pulling the EMA. Through the comparison between actuator position difference $X_s - X_e$ and force fighting $\gamma$, it is shown that these quantities have same evolution and the peaks are located at the same time (0.116s).

Meanwhile, with only a position demand of 2% full stroke, the dynamic force fighting already
reaches 35KN that represents 70% of the rated force. Opposite to this, with the external force input of 20% rated force, the max force fighting is only 14KN that is 28% of rated force. This highlights that the dynamic force fighting in pursuit condition is much more serious than that in rejection condition even the model is quite simple. This has been proven in practice.

As shown in equation 2.37, while $F_L$ is null, the load dynamics is driven by the sum of $F_s$ and $F_e$. But as indicated in the lower left graph, only a few part of power is used to drive the load and most of the power is counteracted by the force fighting between SHA and EMA. The summing magnitude of $F_s$ and $F_e$ is less than 1KN, opposite to the 35KN force fighting. This shows the force fighting greatly increases the total power consumption and stresses the airframe, flight control surface and actuator. This absolutely will accelerate the fatigue of connecting material and reduce their service life. As these grave influences, the force fighting must be addressed with more attention.

The system rejection performance to external load step is shown in the right part of Fig 2-18. The simulation results about statics are consistent with the calculations of equation 2.43 and 2.44. About the dynamics, except for the low frequency response of position loop, there is still a high frequency resonance (around 140Hz). This resonance is caused by the spring/mass effect driven by the overall stiffness in both SHA and EMA channels. It can be explained using figure 2-19:

```
Fig 2-19 Structural stiffness in hybrid actuation system
```

where $S_h$ is the SHA hydraulic stiffness, $S_{sm}$ is the SHA body stiffness, $S_{rs}$ is the roller-screw stiffness and $S_{em}$ is the EMA body stiffness. The hydraulic stiffness is:

\[
S_h = \frac{4E_s A_t^2}{V_t} = 9.92 \times 10^7 [N/m]
\]

(2.45)

As the EMA equivalent mass is so huge equal to 40371Kg (see equation 2.32), in comparison
with the 600Kg load mass, it could be considered as fixed at the first natural frequency of oscillation. Therefore, the influence of \( J_m \) and \( S_{em} \) is neglected. The actuator rod masses \( M_s \) (10Kg) and \( M_e \) (15Kg) are so small in comparison with the load mass that their influence is also neglected. The SHA body stiffness and EMA roller-screw stiffness are considered as very high for now, so their influences are not considered. Finally, the effective parts include:

![Diagram](image.png)

**Fig 2-20** Effective stiffness in linear hybrid actuation system

According to Fig 2-20, the frequency of resonance can be calculated as:

\[
\omega = \sqrt{\frac{S_{st} + \frac{S_s S_{st}}{S_e + S_{st}}}{M_t}} \approx 893 \text{rad/s} = 142 \text{Hz}
\]  

(2.46)

That is very close to the simulation one. The frequency of this resonance is far higher than the position control domain (only a few Hz). Meanwhile, as the load displacement information is not involved in the actuator position controller (only the actuator rod extension is controlled), it is not easy to damp this vibration only with the notch filter. The dynamic pressure feedback could be introduced in the SHA control for this issue.

**b) Position demand sine & external force step**

This simulation is run to evaluate the load position pursuit dynamics under external load. The input is a 1mm/3Hz sine position demand from the beginning and a 10KN external force step applied at 1s.

As indicated in Fig 2-21, on the basis of this linear prototype, the position pursuit dynamics of the hybrid actuation system can meet the requirements even under 10KN external force input. Once applied the external force, there is a small offset on pursuit, but the response is globally satisfactory.
In summary, through the two simulations, the designed position controllers for SHA and EMA make the hybrid actuation system meeting the requirements of position pursuit and rejection. Meanwhile, the force fighting between actuators is pointed out. The dynamic force fighting in pursuit condition is found much graver than that in rejection condition.

### 2.4 Conclusion of Chapter 2

The main challenge of present chapter was to introduce the force fighting through very simple modeling. On the basis of the linear approach, the position controllers for SHA and EMA have been firstly designed: pure P plus notch filter controller for the SHA and cascade controller (velocity and position) for the EMA. Then, the position controllers were validated on test bench and showed a good performance on meeting the requirements. While designing the position controllers, the studied actuator was demanded to drive the full load and the other actuator was disconnected. This was consistent with the conditions of active/standby mode. Spending time to design sophistical controllers (e.g. pole placement with state feedback) was out of scope of the present work and not consistent with simplicity and cost requirements.

With the above designed position controllers, the load pursuit and rejection performances of the hybrid actuation system was proved meeting the requirements on the basis of the hybrid
linear prototype. However, the force fighting between SHA and EMA was found serious. As the difference on their individual statics and dynamics, the SHA and EMA do not always output exactly the same position during transients that makes them not able to share the load equally. In view of the grave effects of force fighting on accelerating fatigue and increasing power consumption, this chapter has shown how important it is to address the static and dynamic force equalization with care.

In order to summarize the analysis, the force fighting is categorized as in Tab 2-3.

<table>
<thead>
<tr>
<th>Force fighting caused by pilot demand (pursuit condition)</th>
<th>Statics (hard)</th>
<th>Mainly driven by the actuators position offset, while the position static errors of SHA and EMA are identical or similar; this force fighting could be decreased.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Dynamics (grave)</td>
<td>Mainly driven by the actuators position pursuit dynamics, while the SHA and EMA have identical or similar pursuit dynamics, this force fighting could be decreased.</td>
<td></td>
</tr>
<tr>
<td>Force fighting caused by external load (rejection condition)</td>
<td>Statics (hard)</td>
<td>Mainly driven by the actuators closed loop stiffness and transmission stiffness, while the SHA and EMA have identical or similar position rejection stiffness, this force fighting could be decreased.</td>
</tr>
<tr>
<td>Dynamics (light)</td>
<td>Mainly driven by the actuator position rejection dynamics and overall equivalent stiffness, as the frequency of oscillation in this case is much higher than the position control domain, it is not easy to damp it.</td>
<td></td>
</tr>
</tbody>
</table>

According to Tab 2-3, the force equalization mainly focuses on the static force fighting under all conditions and the dynamic force fighting under pursuit condition. In the following parts, the force equalization control strategies shall be proposed based on the position controllers designed for the SHA and EMA in present chapter. The static force equalization issue will be addressed as the first step and then the dynamic force equalization issue will be dealt with.

But before this, a more accurate prototype of the hybrid actuation system is necessary to be built to accelerate the model based design of force equalization and to enable the robustness analysis. As displayed in Fig 2-7 and Fig 2-13, although the linear models can represent the dynamic performance of actuators, the static accuracy is not acceptable because many effects are linearized or neglected. In fact, a lot of nonlinearities have important effects on system performance, like the servovalve flow/pressure characteristics, roller-screw friction, backlash etc. The system force response is very sensitive to these nonlinearities. So, for designing force
equalization strategies, a virtual prototype including all the important nonlinearity effects has to be built, as reported in the next chapter.

Important conclusions of present chapter:

- **Conclusion 1**: The linear model is unable to show the static force fighting under null load as far as no offset is considered.
- **Conclusion 2**: Addressing force fighting with virtual prototype requires modeling with care a lot of nonlinearities and paying attention to structural compliances.
- **Conclusion 2**: the dynamic force fighting in pursuit condition is much more critical than that in load rejection condition.
Chapter 3
Virtual Test Bench Building

As already mentioned in chapter 2, introducing with care the nonlinear effects into the virtual prototype is mandatory to get realistic conditions for studying the force equalization. But making nonlinear and detailed model is a cumbersome task in Matlab/Simulink environment as all the models should be built from zero. Opposite to this, it is easier in LMS_AMESim environment as many common engineering models are already available in its library. It is possible to set the nonlinearity on the basis of these submodels and this saves much time and work. The effort can be put on design instead of the numerical issues such as integration continues, integration access discontinuities, discrete events, and so on. Meanwhile, it can be used to numerically study the system robustness.

AMESim is an advanced modeling environment for performing the simulations of engineering systems. It is based on an intuitive graphical interface in which the system is displayed by symbols throughout the simulation process. AMESim uses symbols to represent individual components within system, like the ISO symbols for hydraulic components, block diagram symbols for control systems and the easily recognizable pictorial symbols for mechanic components [6].

In present chapter, a virtual prototype of the hybrid redundant actuation system is built in the AMESim with special attention. The key nonlinearity effects within system are introduced and validated through partial experiments. Then, on the basis of this validated virtual test bench, the force fighting between actuators is more accurately observed, highlighted and analyzed.

3.1 Nonlinear Effect Modeling

In this section, the characteristics of key nonlinearity effects and their modeling in AMESim are studied in detail.
3.1.1 Electric Signaling

In chapter 2, the sensors dynamics, sampling, quantization and saturation have all been neglected. These effects are all within the position control loop and they have significant influence on system statics and dynamics. Their characteristics can be got from the datasheet supplied by suppliers and by discrete control choices. They are represented by the models in control system library, as summarized in Tab 3-1.

<table>
<thead>
<tr>
<th>Items</th>
<th>Description</th>
<th>Submodels</th>
</tr>
</thead>
<tbody>
<tr>
<td>Electric Devices Equivalent Dynamics (linear effect)</td>
<td>First Order Filter placed on any A/D converter input</td>
<td>( k(1+T_3) ) ( t=0.33\text{ms} )</td>
</tr>
<tr>
<td></td>
<td>Second Order Filter for SHA LVDT conditioner (SHA position measurement)</td>
<td>( \frac{k_0}{\omega^2+2\xi\omega+1} ) ( \omega=440\text{Hz}, \xi=0.9 )</td>
</tr>
<tr>
<td></td>
<td>Third Order Filter: Butterworth filter 500Hz for SHA pressure sensor conditioners and force sensor conditioners</td>
<td>( b_0, b_1, b_2, a_0, a_1, a_2 = 1, 1/500\pi, 2/(1000\pi)^2, 1/(1000\pi)^3 )</td>
</tr>
<tr>
<td>Sampling</td>
<td>Sampling rate for control: 0.6ms (1666.7Hz)</td>
<td>( t_s=0.6\text{ms} )</td>
</tr>
<tr>
<td>Quantization</td>
<td>Quantization rate of A/D converter device: 14 bits for ±10V full scale</td>
<td>Quantizer: 0.001V</td>
</tr>
<tr>
<td>Saturation</td>
<td>Saturation of DAC device: ±10V for analog outputs to servovalve amplifier and EMA motor driver Parvex (torque demand)</td>
<td>Saturation element: High limit +10V, low limit -10V</td>
</tr>
</tbody>
</table>

3.1.2 SHA Servovalve

In chapter 2, the servovalve was linearized as a pure proportional effect from current to flow, as shown in equation 2.8. The dynamics of servovalve spool, the influence of load pressure on flow and the influence of spool opening on leakage were all neglected. The servovalve is the key component in SHA and these characteristics play a major role on SHA performance.

The servovalve is modeled as displayed in Fig 3-1.
The virtual servovalve includes three parts: the left part is the servovalve voltage to current amplifier, the gain is 5mA/V; the middle part presents the features of pilot stage from input current $I_{sv}$ to main valve opening $X_{sv}$; the right part presents the features of power stage from valve opening to powered flow $Q_{sv}$. The servovalve performance is mainly driven by the last two parts. They will be modeled in details hereunder.

a) Pilot Stage

In servovalve, the pilot stage is an electro-hydraulic-mechanical amplifier (including electric torque motor, double flapper-nozzle hydraulic amplifier and power spool) which changes the input current to spool opening [60]. It is a very complex mechanization. Modeling this stage regarding to physical structure is a very cumbersome task. As the characteristics of this stage are almost constant while the system supply pressure is constant, this stage is presented with a signal view only. A more detailed modeling considering the input magnitude, the supply pressure and the output flow effects can be found in [71].

The performance of pilot stage is reproduced by statics and dynamics.

Statics of Valve Opening:

The valve opening statics is obtained through partial experiments. The servovalve current is inputted with a pure sine wave having a 50mA magnitude and a 0.5 Hz frequency. The spool displacement is measured by a LVDT attached to the spool. The hydraulic supply pressure is kept at a constant value of 85bars. In addition, in order to operate without the influence of flow force acting on the spool, the use ports A and B of servovalve are interconnected. This feature can be easily enabled by the SHA manifold which includes function to bypass the
cylinder chambers [41]. The experiment result is shown in Fig 3-2.

![Fig 3-2 Experiment about spool displacement statics](image)

As shown in Fig 3-2, the servovalve rated current is ±15mA and the corresponding rated valve opening is ±213μm. But while the current is over this rated area, the servovalve still works well. It can open to 600μm at 50mA, so the practical operating area is ±50mA. In the whole operating domain, the linear area is ±30mA and ±426μm, where the gain is 14.2μm/mA. While the current increases over this linear area, the valve opening is close to saturation.

The time delay effect found on the experimental curve comes from the dynamics of servovalve. A much slower sine input shows that no hysteresis effect is observed. Therefore, the statics of valve opening is considered as the mid plot, as indicated in Fig 3-2. 201 points are sampled from the mid plot to form an ASCII file, which can reproduce the statics from current to valve opening in AMESim over the full ±50mA range.

**Dynamics of Valve Opening**

The valve opening dynamics is also obtained through experiments. The experimental setting is similar as statics. The servovalve input current is a sine wave, whose magnitude is the rated value 15mA and the frequency changes from 0.5Hz to 80Hz. The experimental results are summarized in Tab 3-2.
Tab 3-2 Summary of spool dynamics experiment

<table>
<thead>
<tr>
<th>Frequency</th>
<th>0.5Hz</th>
<th>10Hz</th>
<th>20Hz</th>
<th>30Hz</th>
<th>40Hz</th>
<th>50Hz</th>
<th>60Hz</th>
<th>70Hz</th>
<th>80Hz</th>
</tr>
</thead>
<tbody>
<tr>
<td>Magnitude Ratio</td>
<td>0</td>
<td>-0.42dB</td>
<td>-1.12dB</td>
<td>-2.22dB</td>
<td>-3.33dB</td>
<td>-4.83dB</td>
<td>-5.92dB</td>
<td>-7.07dB</td>
<td>-8.17dB</td>
</tr>
<tr>
<td>Phase Delay</td>
<td>-1.42°</td>
<td>-20.88°</td>
<td>-35.28°</td>
<td>-50.18°</td>
<td>-61.63°</td>
<td>-72.72°</td>
<td>-79.49°</td>
<td>-86.44°</td>
<td>-92.16°</td>
</tr>
</tbody>
</table>

On the basis of these experimental results, the dynamics of valve opening is identified as a second-order filter with a natural frequency of 75.7 Hz and a dimensionless damping factor of 1.283. The identification results are also shown in Fig 3-3.

It can be noticed that the servovalve dimensionless damping factor is over damped. It has to be kept in mind that the servovalve opening gain depends directly on the supply pressure. As servovalve is set in factory to be stable at the maximum supply pressure of 250 bars [60], it becomes over damped while the supply pressure is decreasing.

b) Power Stage

The power stage transfers valve opening to powered flow. It is constructed by the cylindrical spool and valve housing. As displayed in Fig 3-1, this stage is modeled in mechanical view. It is on the basis of the spool submodels with rounded edges and annular orifices in hydraulic component library. It can well reproduce the effect of spool/housing clearance, the rounded edges of the metering wall and the effect of reduced flow number. The main issue comes from the difficulty to set the parameters to reproduce the flow/opening gain and flow/pressure...
Chapter 3 Virtual Test Bench Building

gain. This has been achieved by a curve fitting approach as detailed in [56] (for each orifice, the spool housing clearance, the rounded edges radius and under lap). In Tab 3-3, the geometry and identified parameters of power stage are summarized.

Tab 3-3 Summary of servovalve power stage parameters

<table>
<thead>
<tr>
<th>Known Geometry Parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Spool diameter: 6.9mm</td>
</tr>
<tr>
<td>Width of slot: 3mm</td>
</tr>
<tr>
<td>Number of slots in each pocket: 4</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Identified Parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Variable orifice 1 underlap: 13.5µm</td>
</tr>
<tr>
<td>Variable orifice 3 underlap: 16µm</td>
</tr>
<tr>
<td>Variable orifice 4 underlap: 13.5µm</td>
</tr>
<tr>
<td>Orifice rounded corner radius: 7.9µm</td>
</tr>
</tbody>
</table>

Then, according to the experimental results in [56], some simulations are run to evaluate the performances of this virtual servovalve on reproducing the characteristics of pressure drop at null opening, flow/pressure gain and flow/opening gain.

**Pressure Drop and Flow/Pressure Gain**

It should be noticed that as the hydraulic cylinder is a naturally integrator in position control the servovalve operates averagely around the null opening. The pressure drop and leakage around null opening play an important role on studying force equalization as it drives the SHA force at very low speed.

The simulation is run in following conditions: the valve opening is measured as the input and driven by a very low frequency sine wave current, its varying range is ±60µm; the use ports A and B are blocked to force the valve not delivering any flow to jack; the pressure at use ports A and B are measured to give the pressure/valve opening gain; the flow at oil supply P port is measured to give the flow/pressure gain; the hydraulic supply pressure is 80bars at 50°C, the fluid viscosity is 27cp. The simulation results are shown in Fig 3-4.

As shown in Fig 3-4, the pressure drop at null opening is sufficiently well represented by the virtual prototype. The simulated leakage has some error from the measurement result. This is because the measured flow is got by difference between the total flow and the pilot flow that
is assumed to be constant. Generally, the pressure/flow characteristic is also sufficiently well reproduced.

**Flow/Valve Opening Gain**

In order to validate the flow/opening characteristics of virtual servovalve, a simulation is run in following conditions: the valve opening is forced to a sine wave of 600μm/0.1Hz by acting on the input current; the use ports A and B are short-circled to get null load pressure as required of defining the flow gain; the flow from port A to port B is measured to give the flow/opening plot; the hydraulic supply pressure is 80bars at 50°C, the fluid viscosity is 27cp. The simulation result is compared with the manufacturer data [60], as shown in Fig 3-5.
It can be observed that the virtual servovalve well reproduces the flow characteristics. In 5% nominal opening area, its flow gain is close to 140% of the nominal value, which is within the normal range of 50% to 200%. For making clear what causes this tolerance, the robustness study over parameter uncertainties is suggested. Meanwhile, in larger valve opening area, it becomes very close to the nominal flow gain.

In summary, the effort placed on accurate modeling of servovalve provides us with a realistic virtual servovalve. It is considered as mandatory for later validation of the force equalization control strategies.

3.1.3 SHA Jack Friction

The friction between two contact objects is a very complex phenomenon which is influenced by many factors, such as the sliding velocity, external load, environment, service time, and so on [72]. In chapter 2, only an equivalent viscous effect has been considered as it makes the friction model linear vs. velocity. In order to more accurately reproduce the characteristics of jack, the neglected friction effects should be added.

The common effects considered within the jack friction are summarized as following:

- **Coulomb effect**: a constant friction that does not dependant on sliding velocity.
- **Velocity dependent effects**: it can include a Stribeck effect (rapid change of friction at low velocity), a viscous effect (friction is proportional to velocity) and a windage effect (friction is proportional to acceleration).
- **Load dependent effect**: the upper effects are all subjected to change vs. load and even vs. the sign of the power transmitted to load. Although this effect is often neglected, it plays a major role in friction [41].

In view of this, the jack friction is identified through a curve fitting approach in two steps: the first step identifies the Coulomb effect and velocity dependent effects; the second step identifies the load dependent effect.

a) **Coulomb Effect and Velocity Dependent Effects**

At first, a friction model involving Coulomb effect and velocity dependent effects is proposed
as equation 3.1.

\[ F_{jj} = \left[ F_{cl} + (F_{st} - F_{cl}) e^{\frac{\nu_r}{\nu_r}} \right] \text{sgn} \left( \dot{X}_s \right) + B_s \ddot{X}_s + W_s \dddot{X}_s \] (3.1)

where:

- \( F_{jj} \): Jack friction [N]
- \( F_{cl} \): Coulomb friction [N]
- \( F_{st} \): Stribeck friction [N]
- \( \nu_r \): Stribeck reference velocity [m/s]
- \( B_s \): Viscous friction coefficient [N·s/m]
- \( W_s \): Windage friction coefficient [N·s²/m]

The model identification is on the basis of experimental results. The EMA and surface are all disconnected to force null external load on SHA. The SHA is position controlled and the position demand is a triangular wave to get a constant velocity motion of a range \( \pm 80 \text{mm/s} \). The hydraulic supply pressure and the fluid temperature are kept constant at 85 bars and 50°C respectively. The experimental and identification results are displayed in Fig 3-6.

The identified friction parameters are: \( F_{cl} = 590 \text{N} \), \( F_{st} = 1269 \text{N} \), \( \nu_r = 7.2 \times 10^{-3} \text{m/s} \), \( B_s = 800 \text{N·s/m} \) and \( W_s = 0 \). As shown in Fig 3-6, the identified model well reproduces the jack friction under null external load condition. The Coulomb and Stribeck effects are observed playing a major role in jack friction. The influence of viscous effect is very small and the influence of windage effect is considered as null.

Once the parameters are identified, the accuracy of this friction model is evaluated through
several sine velocity experiments. As shown in Fig 3-7, the friction model is accurate in all the operating range. The lower plots seem to suggest the presence of hysteresis. However, they come from the dynamics of linear filter that is used to get velocity from the measured position signal. As observed, this effect is well reproduced when the filter model is introduced.

**Fig 3-7** Jack friction at sine velocity and null external force

**b) Load Dependent Effect**

While the jack is loaded, the pressure change of two cylinder chambers will lead the seals to be differently stressed and consequently modify the amount of friction. At first, a jack friction model involving external load effect is proposed as equation 3.2:

\[
F_{ff} = \left( F_{cl} + (F_{cl} - F_{cl}) e^{-\frac{|\dot{x}|}{\gamma}} + |F_{ex}|(a + b \text{sgn}(F_{ex}\dot{x})) \right) \text{sgn}(\dot{x}) + B_x\dot{x} \tag{3.2}
\]

where \(a\) and \(b\) are the dimensionless factors of jack friction dependency on external load. \(F_{ex}\) is the external resistant force acting on SHA rod level, unit [N].

In order to study the influence of external load on jack friction, several experiments are done. The EMA is connected and position controlled. Its position demand is a triangular wave in order to get a constant velocity motion. The SHA is force controlled to generate a constant force whatever velocity. In this way, the influence of friction coming from connecting joints
between SHA and EMA is removed. The hydraulic supply pressure and fluid temperature are still kept at 85bars and 50°C. The experimental results are displayed in Fig 3-8.

On the basis of the experimental results, the model parameters are identified as: $F_{cl}=590\text{N}$, $F_{st}=1269\text{N}$, $v_r=7.2\times10^{-3}\text{m/s}$, $B_s=800\text{N·s/m}$, $a=0.0087$ and $b=0.0197$. The model results are also displayed in Fig 3-8.

It can be observed that the mean error between experimental results and model results is less than 130N that is only 10% of the friction magnitude. Meanwhile, the maximum jack friction 1.6KN under 10KN external force is only 6% of the SHA capability output force 26KN (85bars supply pressure). This means the friction modeling error is less than 0.5% of the SHA max force. So generally, the friction model is sufficiently accurate enough on reproducing the real jack friction.

Following that, the jack friction is reproduced in AMESim as shown in Fig 3-9.
Chapter 3 Virtual Test Bench Building

The jack friction model is made of two parts: the signal part to calculate the amount of friction; the mechanical part to add friction effect into system.

- **Signal part:** it is built with a two inputs function submodel, which is set on the basis of upper identified friction model. The two inputs are the SHA rod velocity and the external force acting on SHA rod. The output is sent to the mechanical friction submodel.

- **Mechanical part:** it is built on the basis of an existing linear friction submodel, which is connected to cylinder and rod models to reproduce the friction between them. It is set on the mode of using external input friction value. A tanh function is used on the input friction value to avoid the strong nonlinear step effect at null velocity, which can be observed in Fig 3-6. A critical velocity is set to limit the tanh effective range. In this model, it is set as 0.01mm/s.

3.1.4 **EMA Rotary Inertia and Roller-Screw Friction**

As indicated in Fig 2-9, the roller-screw friction considered on the motor side is magnified by $2\pi/l$ (2094) times when it is reflected on the rod side. As it has a significant influence on the EMA performance, an accurate representation of friction is absolutely necessary. Similar to the jack friction, the roller-screw friction was also linearized as a viscous effect in chapter 2. In the present section, the effects of Coulomb, Stribeck and external load shall be introduced.

The roller-screw friction has been proposed and identified by Doctor Wissam Karam in [41], as shown in equation 3.3.

$$
T_f = \left( T_{cl} + (T_{st} - T_{cl}) e^{-\frac{F_{ex} \omega_r}{c + d \text{sgn}(F_{ex} \omega_r)}} \right) \text{sgn}(\omega_r) + B_e \omega_r
$$

(3.3)

where:

\begin{align*}
T_f & \quad \text{Roller-screw friction torque [N·m]} \\
T_{cl} & \quad \text{Coulomb friction torque [N·m]} \\
T_{st} & \quad \text{Stribeck friction torque [N·m]} \\
\omega_r & \quad \text{Stribeck reference velocity [rad/s]} \\
c & \quad \text{External load dependency factor [m]} \\
d & \quad \text{External load dependency factor [m]}
\end{align*}

The identified friction model parameters were: $T_{cl}=3.624\text{Nm}$, $T_{st}=-2.245\text{Nm}$, $\omega_r=70.55\text{rad/s}$, $B_e=0.016\text{Nm/(rad/s)}$, $c=1.04\times10^{-4}\text{m}$ and $d=-6.21\times10^{-5}\text{m}$. Meanwhile, the EMA rotary inertia was also accurately identified as $J_m=0.00846\text{Kgm}^2$. 

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It can be observed that the Stribeck friction is negative. This is caused by the preloading on EMA to avoid backlash. In addition, the maximum roller-screw friction torque is close to 6Nm. While reflected on rod level, it is about 12.5KN which is 25% of the EMA rated force 50KN. In view of this, the importance of accurately modeling roller-screw friction becomes obvious.

The roller-screw friction is reproduced in AMESim as displayed in Fig 3-10.

Similar to the jack friction prototype as shown in Fig 3-9, this roller-screw friction prototype is also constructed by two parts: signal part and mechanical part.

- **Signal part:** it is built with a two inputs function submodel set on the basis of equation 3.3. The two inputs are the motor rotary velocity and the external force which is got from the force sensor attached on the EMA rod.

- **Mechanical part:** it is built on the basis of an existing rotary Coulomb friction submodel, which is connected to a torque summing rotary node. Within this rotary node, the torque at port 1 subtracts the torque at port 2 and the result is outputted at port 3; the velocity at port 3 is transferred to ports 1 and 2. Through this node, the roller-screw friction effect on motor shaft is reproduced. The friction submodel is set as the mode of using external input friction value. A tanh function is also used on the input friction value and the critical velocity is set as 0.1rev/min.

### 3.1.5 Joint Friction between SHA and EMA

While measuring the SHA jack friction under external load in section 3.1.3, it is found that the outputs of force sensors fixed on the rod of SHA and EMA are not identical. As the surface is in
constant velocity motion, the load dynamics influence can be neglected. Therefore, this should be caused by the friction existing inside the joints between SHA and EMA.

At first, this joint friction is measured on the basis of experimental results of Fig 3-8, as shown in Fig 3-11. It can be observed that this friction is mainly driven by the external force effect (the average of two force sensors which are attached on rods of SHA and EMA). The velocity dependent effects are so small that can be neglected. According to this, a friction model is proposed as shown in equation 3.4.

\[
F_{jpf} = F_{jtcl} + |F_{ex}| \left( e + f \text{ sgn}(F_{ex}) \right)
\]  

(3.4)

where:

- \(F_{jpf}\) Joint friction between SHA and EMA [N]
- \(F_{jtcl}\) Joint Coulomb friction [N]
- \(e\) External force dependent factor [null]
- \(f\) External force dependent factor [null]

The model parameters are identified as: \(F_{jtcl}=188\text{N}\), \(e=0.056\) and \(f=0.0085\). The model results are added to Fig 3-11.

The mean error between experimental and model results is less than 20N, which is only 10% of the minimum joint friction 200N. This shows the good representation of identified model.

The coming issue is that how this joint friction is distributed at SHA side and EMA side. It is infeasible to measure it on the basis of actual test bench because the joints are between the
two force sensors. For solving this problem, a distribution weight coefficient is introduced. It will be valued during the validation of virtual test bench to get a more representative result.

The joint friction is reproduced in AMESim as displayed in Fig 3-12. Two identical ones are involved, one for SHA and one for EMA.

![Diagram of joint friction in AMESim]

**Fig 3-12** Realization of joint friction in AMESim

The virtual joint friction is also constructed by two parts:

- **Signal part**: it has a one input function submodel and a proportional gain. The function is set on the basis of equation 3.4 and its input is the actuator force sensor's output. The proportional gain is used to set the distribution weight.

- **Mechanical part**: it is built on the basis of an existing linear friction submodel, which is connected to actuator rod and load to reproduce the friction between them. It is set as the mode of using external input friction value. The tanh function is not used as the velocity effect is not involved in this model.

### 3.1.6 Comparison of Friction Level

The main frictions on test bench have been accurately reproduced in the virtual test bench, including the hydraulic jack friction, roller-screw friction and joints friction. In order to make clear which friction plays a major role on system performance, they are simulated under same operating condition. The simulated velocity range is ±100mm/s and the external force is set to three constant values 0N, 5KN and 10KN. The simulation results are shown in Fig 3-13.
In Fig 3-13, the frictions acting on rod level are displayed. Obviously, the roller-screw friction has a larger magnitude whatever the operating conditions. The frictions under 10KN external force are summarized in Tab 3-4 for demonstration.

Tab 3-4 Friction level under 10KN external force

<table>
<thead>
<tr>
<th>Items</th>
<th>Roller-Screw Friction</th>
<th>Jack Friction</th>
<th>Joint Friction</th>
</tr>
</thead>
<tbody>
<tr>
<td>Resistant Load 100mm/s</td>
<td>8.02KN</td>
<td>955N</td>
<td>833N</td>
</tr>
<tr>
<td></td>
<td>16% EMA Max Force 50KN</td>
<td>3.67% SHA Max Force 26KN</td>
<td>3.2% SHA Max</td>
</tr>
<tr>
<td>Resistant Load 0.1mm/s</td>
<td>3.56KN</td>
<td>1.6KN</td>
<td>833N</td>
</tr>
<tr>
<td></td>
<td>7.12% EMA Max Force</td>
<td>6% SHA Max Force</td>
<td>3.2% SHA Max</td>
</tr>
<tr>
<td>Aiding Load -0.1mm/s</td>
<td>-6.16KN</td>
<td>-1.16KN</td>
<td>-833N</td>
</tr>
<tr>
<td></td>
<td>12.32% EMA Max Force</td>
<td>4% SHA Max Force</td>
<td>3.2% SHA Max</td>
</tr>
<tr>
<td>Aiding Load -100mm/s</td>
<td>-10.6KN</td>
<td>-561N</td>
<td>-833N</td>
</tr>
<tr>
<td></td>
<td>21.2% EMA Max Force</td>
<td>2.16% SHA Max Force</td>
<td>3.2% SHA Max</td>
</tr>
</tbody>
</table>

Through Fig 3-13, it is shown that the roller-screw friction plays a very important role for the EMA performance. Therefore, more attention should be paid on it while designing the force equalization control strategies. Oppositely, the influence of jack friction on SHA performance is much smaller.
3.1.7 Transmission Compliance between Actuator and Load

As shown in Fig 1-11, in this hybrid actuation system, only the actuator extensions are used for position closed loop. The load angular position is not measured (on aircraft, a sensor is used to measure the effective position of flight control surface for monitoring purpose only). The load is indirectly controlled through the rod extension of SHA and EMA. Therefore, as the intermediate links between actuators and load, the stiffness of transmission parts plays an important role on load performance. In chapter 2, the transmission stiffness was linearized as a proportional part. In fact, due to the existing of backlash, this compliance involves strong nonlinearity.

In order to make clear its characteristics, an experiment is done. The EMA is force controlled to load SHA and its force demand is a triangular wave slowly varying from -20KN to 20KN in 40s. The SHA is position controlled and has null position demand. The anchorages are locked to be at high stiff mode. The schematic of experimental setting is shown in Fig 3-14.

![Fig 3-14 Experimental setting to measure transmission compliance](image)

The total deformation of the transmission part can be measured by using EMA rod extension added to the SHA rod extension. The results are displayed in Fig 3-15.

It is shown that there is a 0.6mm backlash within the transmission part. Out of this domain, the stiffness increases rapidly to a constant value of $2.6 \times 10^7 \text{N/m}$ making a total deformation of 1mm at 20KN external force. This discontinuity of stiffness has an extremely grave effect on performance. The load pursuit, rejection and force equalization shall be all greatly influenced by this effect.

This backlash is much larger in comparison with that of real aircraft applications (typically...
50μm). It will introduce severe limitation in the experimental validation of force equalization strategy. Therefore, the force equalization strategy can only be validated on virtual test bench for now. This backlash effect is not involved while validating the force equalization strategy on virtual test bench in order to be more representative of a flight control actuation.

The following issue is how the backlash and compliance are distributed on SHA side and EMA side. This is infeasible through dynamic loading experiments, such as in Fig 3-15. Therefore, some static loading tests are done.

On test bench, the backlash mainly comes from the joint between actuator and load, as shown in Fig 3-16.
With static loading tests, the backlash and stiffness of each part are measured (Appendix B):

- **SHA part:** equivalent stiffness $4.5 \times 10^7 \text{N/m}$, total backlash 0.35mm.
- **EMA part:** equivalent stiffness $2.5 \times 10^8 \text{N/m}$, total backlash 0.03mm.
- **Fix pieces:** equivalent stiffness $8.0 \times 10^7 \text{N/m}$, total backlash 0.20mm.

Due to the limitation of test bench, the static loading force is not high enough, so the upper measured values are only approximate results. But it can be still observed that the spherical plain bearing of SHA and the fix pieces are the higher contributors to the backlash.

In order to reproduce the backlash effect, two transmission compliance models are proposed for SHA and EMA, as shown in equations 3.5 and 3.6.

For transmission stiffness between SHA and load:

$$S_{st} = \text{abs}\left(\tanh\left(\frac{X_{sc}}{X_{scr}}\right)\right) \times S_{sc}$$  \hspace{1cm} (3.5)

where:

- $S_{st}$ Stiffness between SHA and load [N/m]
- $X_{sc}$ Compression between SHA and load [m]
- $X_{scr}$ Backlash factor [m]
- $S_{sc}$ Stiffness out of backlash domain [N/m]

For transmission stiffness between EMA and load:

$$S_{et} = \text{abs}\left(\tanh\left(\frac{X_{ec}}{X_{ecr}}\right)\right) \times S_{ec}$$  \hspace{1cm} (3.6)

where:

- $S_{et}$ Stiffness between EMA and load [N/m]
- $X_{ec}$ Compression between EMA and load [m]
- $X_{ecr}$ Backlash factor [m]
- $S_{ec}$ Stiffness out of backlash domain [N/m]

Then, it should be noticed that while SHA and EMA are operating under different connecting condition, the equivalent compliance and backlash are different. In case only one actuator is connected to the load, the total equivalent compliance will involve the effect of fix pieces; but in case both actuators are connected, the effect of fix pieces shall be shared. The model setting in different condition is summarized in Tab 3-5.
### Tab 3-5 Summary of transmission compliance setting under different condition

<table>
<thead>
<tr>
<th>Operating Condition</th>
<th>SHA Transmission Compliance</th>
<th>EMA Transmission Compliance</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Individually</strong></td>
<td>$S_{e}=2.9\times10^7\text{N/m}$, $X_{ecr}=0.001\text{m}$</td>
<td>$S_{e}=6.0\times10^7\text{N/m}$, $X_{ecr}=0.0005\text{m}$</td>
</tr>
<tr>
<td><strong>Connected</strong></td>
<td>$S_{e}=2.9\times10^7\text{N/m}$, $X_{ecr}=0.001\text{m}$</td>
<td>$S_{e}=2.5\times10^8\text{N/m}$, $X_{ecr}=0.0001\text{m}$</td>
</tr>
</tbody>
</table>

In connected condition, because SHA has only one joint located on mid position of shaft, it is considered sharing the most part of fix pieces’ compliance effect.

In the end, these transmission compliances are reproduced in AMESim in same way.

![Fig 3-17 Realization of transmission compliance in AMESim](image)

As displayed in Fig 3-17, the virtual transmission section is composed by:

- **Signal part**: two position sensors are used to measure the compression of transmission part. The two inputs function submodel is set on the basis of equations 3.5 and 3.6. Its output is a stiffness value which is sent to the mechanical spring submodel.

- **Mechanical part**: a variable linear spring and a constant linear damper are involved. The spring is set on the mode of using external input compliance value. On the basis of setting in the above function submodel, it can reproduce the backlash effect. The damper is used to reproduce the structure damping effect at contact parts. As there is a high uncertainty on this parameter, it is very difficult to measure an accurate value. Meanwhile, it does not have a significant influence on performance. So, it is valued during validating the virtual test bench to get a much closer response as experiment, its typical value is $11000\text{N} \cdot \text{s/m}$.

#### 3.1.8 Compliance of Anchorage

The anchorage between actuator and test bench, as shown in Fig 3-18, it has been neglected in chapter 2. It is involved in virtual test bench in order to reproduce the experimental results.
done on test bench. The anchorage can be set to two different stiffness values:

- **In locked mode**: the linkage between the actuator body and the test bench frame can be considered having an infinite stiffness in comparison with other parts. The typical value is $1.0 \times 10^9 \text{N/m}$, which is several times stiffer than that on a real aircraft (typically 3mm at 50KN making a stiffness of $1.7 \times 10^7 \text{N/m}$).

- **In free mode**: a compliance is inserted in series between the actuator body and the test bench frame to reproduce a realistic condition on aircraft.

The anchorage stiffness in free mode is measured. The experiment setting is similar as in Fig 3-14, EMA is force controlled to load the anchorage. The compression is measured with a sensitive position sensor which is attached to actuator body and test bench frame. The results are shown in Fig 3-18.

![Fig 3-18 Compliance of anchorage and its realization in AMESim](image)

The anchorage stiffness in free mode is a constant value of $1.06 \times 10^7 \text{N/m}$. The experimental results show a backlash effect which is found coming from the mechanical connectors of position sensor.

The reproduction of anchorage in virtual test bench is also displayed in Fig 3-18. An existing mechanical spring and damper submodel is involved. The spring is set to reproduce the effect of compliance. The damper is set to reproduce the damping effect. As the similar reason of the transmission part, the damping value is typically set as $10000\cdot \text{s/m}$.
3.2 Validation of Virtual Test Bench

In this section, all the above mentioned effects are combined to make the virtual test bench of
the hybrid actuation system in Fig 1-10. The virtual SHA and virtual EMA shall be individually
validated through comparison with experimental results. Before this, the virtual surface load
is introduced, as displayed in Fig 3-19.

A force summing node is used to reproduce the force effect acting on load. A one-dimensional
motion mass is used to reproduce the surface inertia effect. The air force acting on surface is
simulated by a signal generator.

3.2.1 Validation of Virtual SHA

The virtual SHA is associated as in Fig 3-20.

where the numbers refer to the following effects:

1) The virtual servovalve built in Fig 3-1 is associated as a supercomponent.
2) A 2 position 2 port hydraulic control valve submodel is used to reproduce the effect of the bypass valve fixed within hydraulic manifold.

3) A variable hydraulic orifice submodel is used to reproduce the SHA inner leakage.

4) The virtual jack is composed by hydraulic part and mechanical part. The hydraulic part includes the cylinder and volume effect models to reproduce the fluid characteristics in jack. The mechanical part includes moving rod mass and moving body mass to reproduce the realistic motion between rod and housing.

5) Jack friction between rod and housing, as built in Fig 3-9.

6) Anchorage between actuator body and test bench frame, as built in Fig 3-18.

7) Transmission compliance between SHA and load, as built in Fig 3-17.

8) Joint friction shared by SHA, as built in Fig 3-12, the weight factor is 0.4.

9) Sensor dynamics, ADC dynamics, sampling and quantization, as summarized in Tab 3-1.

10) SHA P + Notch filter position controller built on signal view.

11) Hydraulic fluid properties setting, such as density, bulk modulus, viscosity etc.

12) Mechanical gravity setting.

13) Position demand input \( x_r \).

14) SHA output force to load \( F_s \) and load velocity reflection \( \dot{x}_r \).

For validating the virtual SHA, experiment and simulation are done in same conditions, the results are shown in Fig 3-21. The EMA is firstly disconnected to avoid its load effect. The SHA is position controlled. Its position demand is a square wave of 1mm magnitude and 0.2Hz
frequency. The hydraulic supply pressure and fluid temperature are kept at 85bars/50°C.

As shown in Fig 3-21, the virtual SHA sufficiently well reproduces the real performances. It improves very significantly the prediction of the output force. However, the virtual SHA seems a little over damped with a slightly lower first resonance. This is found coming from the backlash domain where it remains difficult to get a realistic model of contact damping.

In addition, it must be emphasized that force is a very dynamic state compared with position (twice integration of acceleration that comes from force balance). Therefore, it is much more difficult to get accurate simulated forces.

### 3.2.2 Validation of Virtual EMA

The virtual EMA is associated as in Fig 3-22.

![Fig 3-22 Virtual EMA built in AMESim](image)

where the numbers refers to the following effects:

1) Dynamics of motor torque control loop, it is set to a 600Hz frequency and 0.707 damping. The limit of motor output torque is integrated in the D/A transfer saturation section.

2) A torque summing node is used to reproduce the torque effect on motor shaft. It involves the motor output torque, roller-screw friction torque and rotary inertia dynamics torque.

3) Roller-screw friction acting on motor shaft, as built in Fig 3-10.

4) A speed summing box is used to reproduce the realistic effect of actuator body motion on rod extension.
5) Roller-screw, reproduced by an existing rack pinion submodel.

6) The EMA roller-screw compliance, set according to [41], see Appendix C.

7) Rod mass and actuator body mass.

8) Anchorage between EMA body and test bench frame, as built in Fig 3-18.

9) Transmission compliance between EMA rod and load, as built in Fig 3-17.

10) Joint friction shared by EMA, as built in Fig 3-12, the weight factor is 0.6.

11) Sensor dynamics, ADC filter, sampling and quantization, as summarized in Tab 3-1.

12) EMA Cascade position controller involving inner velocity loop.

13) Position demand $X_r$.

15) EMA output force to load $F_e$ and load velocity reflection $\dot{X}_l$.

Then, same as for virtual SHA, one experiment and a simulation are done in same conditions to validate the virtual EMA. The SHA is disconnected to avoid its load effect. EMA is position controlled and demanded to pursuit a square wave of 0.5mm magnitude and 0.2Hz frequency. The comparison of experimental and simulation results are displayed in Fig 3-23.

It is observed that the virtual EMA reproduces well the experimental responses, in particular regarding the effects of backlash and roller-screw friction.

In summary, the virtual SHA and EMA show a very good representation of the real test bench behaviors. Following this, they are associated together to build a virtual hybrid redundant actuation system, which shall be used to validate the proposed force equalization strategies and perform the robustness analysis.
3.3 Simulation of Force Fighting on Virtual Test Bench

As concluded in chapter 2, the force fighting is caused by the characteristics differences between SHA and EMA. The static force fighting is mainly driven by the static position errors and static rejection stiffness. The dynamic force fighting is mainly driven by the actuators position pursuit/rejection dynamics. These results were got using linear approach. Now with the validated virtual hybrid actuation system, the conclusions will be confirmed and detailed through simulations. The influence of introduced nonlinearities will be assessed.

The simulation is run with the following setting: the virtual test bench is set totally identical to the real one; a 1mm position step (2% of full stroke) is demanded at t=0.1s followed by a 10KN external force step (20% of stall force) at t=1s. The simulation responses are displayed in Fig 3-24.

![Fig 3-24 Simulation response of virtual test bench under pursuit and rejection conditions](image)

Meanwhile, in order to compare with the performance before introducing nonlinearities, a similar simulation is run on linear model built in chapter 2. The transmission stiffness of SHA and EMA are set to $S_{st}=2.9\times10^7$N/m and $S_{et}=2.5\times10^8$N/m individually for keeping consistence with the virtual test bench. The comparison between linear model response and virtual test
bench response are summarized in Tab 3-6.

<table>
<thead>
<tr>
<th>Input</th>
<th>Items</th>
<th>Linear Model</th>
<th>Virtual Test Bench</th>
</tr>
</thead>
</table>
| Position    | $X_s$: fast and **small oscillation**
$X_e$: fast and **smooth**, no static error
$X_t$: fast and **small oscillation**
$X_s - X_e$: maximum 0.18mm
Transient duration: 0.08s
Oscillation frequency: **110Hz** | $X_s$: fast and **smooth**, no observable error
$X_e$: slow and **vibrated**, static error 0.004mm
$X_t$: slow and **vibrated**, static error 0.003mm
$X_s - X_e$: maximum 0.372mm
Transient duration: 0.15s
Oscillation frequency: **45Hz** |}

**Pursuit Condition**

- **Position**
  - Static FF: **oscillation** mean 0N
  - Dynamic FF: maximum 10KN
  - $F_s$: Max: 5KN  $F_e$: Max: -5.12KN

- **Force Fighting**
  - SHA static error: 0.005mm
  - EMA static error: 0.053mm
  - Load static error: **0.085mm**
  - Transient duration: oscillation
  - Oscillation frequency: **110Hz**

**Rejection Condition**

- **Position**
  - SHA static error: 0.005mm
  - EMA static error: 0.053mm
  - Load static error: 0.085mm
  - Transient duration: oscillation
  - Oscillation frequency: **110Hz**

- **Force Fighting**
  - Static FF: **oscillation** mean -6KN
  - Dynamic FF: maximum -16.51KN
  - $F_s$: Max: 1.67KN  $F_e$: Max: 18.2KN

With comparison, it is shown that the nonlinearities involved within the virtual test bench have significant influence on performance. The system statics and dynamics are all altered.

- **Pursuit performance**: due to the backlash existing in the transmission part between the actuators and load, the load position pursuit dynamics becomes poor. The response speed to step position demand is slow and some vibrations can be observed. Meanwhile, some static errors are found on EMA and load position output, which are due to the backlash and the Coulomb effect in roller-screw friction.

- **Rejection performance**: in comparison with the linear model, the rejection stiffness of SHA and EMA in virtual test bench does not change a lot. However, the load rejection behaviors is greatly altered, which is also mainly due to the backlash.

- **Force Fighting**: due to the nonlinearities, generally, the force fighting becomes graver. At first, the static force fighting under pursuit condition is almost null, which confirms the conclusion got using linear approach, see equation 2.42. This is because the nonlinearities such as the position sensor offset, manufacturing tolerance, servovalve opening offset etc. remain not considered in virtual test bench for the time being. While these effects are
introduced, the static force fighting will become worse. Moreover, as the transmission stiffness of SHA and EMA are not equaled, they do not contribute identical force on load even their pursuit and rejection dynamics are identical, as shown in the right lower graph of Fig 3-24.

In addition, the vibration of frequency 45Hz in pursuit case and vibration of frequency 70Hz in rejection case are all driven by the equivalent overall stiffness of SHA and EMA channels, the calculation rule is similar to Fig 2-19. Their difference from the linear model is due to the low stiffness within the backlash domain of transmission parts. Meanwhile, as the operation points are different in pursuit condition and rejection condition (with or without external load), the transmission stiffness is also different. The closer to the centre of backlash area, the lower the stiffness is (see Fig 3-15). This explains the difference between 45Hz and 70Hz.

The above analysis shows that the backlash and unequalled transmission stiffness greatly alter the performance. In order to confirm their influence, a similar simulation is run setting equaled transmission stiffness and removing the backlash effect. The stiffness value is set as $S_{st}=S_{et}=4.0 \times 10^8$ N/m to be more realistic. The simulation responses are shown in Fig 3-25.

**Fig 3-25** Simulation response with equaled transmission stiffness and no backlash

It is observed that with equaled transmission stiffness and no backlash effect, the system
performance becomes better. The load pursuit dynamics is faster and no vibration can be observed. The load rejection stiffness is greatly increased. The static and dynamic force fighting under rejection condition are all reduced. Only the dynamic force fighting under pursuit condition is worsened due to the high transmission stiffness, which is also the main issue we want to address through the proposed force equalization strategy.

In the end, through the simulations in Fig 3-24 and 3-25, the conclusions about force fighting got using linear approach are confirmed on virtual test bench:

- The static force fighting is mainly driven by the actuator position offset, actuator closed loop stiffness and transmission stiffness.
- The dynamic force fighting is mainly driven by the actuator pursuit/rejection dynamics and the overall stiffness in both channels.

As concluded, the transmission stiffness plays a very important role on force equalization. It will be paid more attention on in following work of designing force equalization strategy.

In addition, it must be noticed that the response of a nonlinear system depends on the input magnitude, opposite to a linear system where they are rigorously proportional. Therefore, the values got from the virtual prototype are for indication only with the chosen input steps.

### 3.4 Conclusion of Chapter 3

In present chapter, in order to more accurately represent the performance of the redundant actuation system, a virtual test bench involving most of the key nonlinearities was built in the AMESim environment. The virtual test bench well reproduced the electric signaling (such as filters, sensor dynamics, sampling and quantization) and the nonlinear physical behaviors (such as the servovalve pilot stage and power stage, roller-screw friction, jack friction, joint friction, backlash, compliance etc).

Then, the virtual test bench was validated by comparing its simulation responses with the experimental results. The validation was performed for each actuator individually. As shown in Fig 3-21 and 3-23, the virtual test bench provided a good representation of real behaviors.
After that, the virtual test bench was used to represent the force fighting between SHA and EMA. It was found that the introduced nonlinearities had a great influence on performance, especially the transmission compliance and backlash. Some conclusions about force fighting were confirmed through the simulations on virtual test bench:

- The static force fighting is mainly driven by the actuator position offset (caused by sensor offset, manufacturing tolerance etc.), actuator closed loop stiffness and the transmission stiffness.
- The dynamic force fighting is mainly driven by the actuator pursuit/rejection dynamics and the overall stiffness (including the hydraulic stiffness, transmission stiffness and the roller-screw stiffness).

In the end, due to the large backlash and no mean for generating external load, the real test bench is considered as not competent for performing all the validations of force equalization strategies. As a substituted solution, the virtual test bench which is built totally regarding to the behaviors of real one will be used to in the following work to validate the proposed force equalization strategy. In order to place the research on a much worse condition, the validation work will be performed as:

- **For static force equalization:** the test bench is used to validate the force equalization strategies under null external load condition because it involves more uncertainties, such as the sensor offset, manufacturing and setting tolerance, etc. It has been confirmed that these uncertainties place the static force fighting on a much graver condition. The virtual test bench will be used for pre-validation.
- **For dynamic force equalization:** as indicated in Fig 3-24 and 3-25, a high transmission stiffness aggravates the dynamic force fighting. Therefore, while performing the validation of dynamic force equalization strategies on virtual test bench, the transmission stiffness of SHA and EMA are set to a very stiff value of $1.0 \times 10^9 \text{N/m}$ that makes the dynamics force equalization very challenging.

**Important Conclusions of Present Chapter:**

- **Conclusion 1:** The conclusions about the origins of force fighting obtained from the linear
approach are globally confirmed on the virtual test bench. The nonlinear effects increase significantly the amount of force fighting.

- **Conclusion 2:** The actual real test bench does not allow applying external load on the hybrid SHA/EMA actuators and it presents unrealistic excessive backlash.
- **Conclusion 3:** The virtual test bench got with effort in modeling nonlinearities is able to reproduce in a satisfactory manner the real behaviors. It will be used as a substitute to the real test bench for the validation of force equalization strategies.
Chapter 4
Static Force Equalization Study

As concluded in chapter 3, the sources of static force fighting and dynamic force fighting are different: the static one is mainly driven by actuator position offset, closed loop stiffness and transmission stiffness; the dynamic one is mainly driven by the actuator pursuit/rejection dynamics and the overall stiffness in both channels. Different origins lead to the different solutions. Therefore, the force equalization is addressed in two steps: the static performance is easier to achieve, so the static force equalization is considered in a first step; the dynamic force equalization will be studied in a second step.

So, the present chapter deals with the static force equalization. At first, the conclusions of static force fighting obtained in former parts are more deeply studied and validated on virtual test bench. According to these conclusions, two static force equalization strategies are proposed and studied. The first one introduces integral force fighting signal to compensate the position control; the second one sets the actuators as master/slave mode, the slave actuator is force controlled to track the output force of the master one that is position controlled. In the end, these strategies are pre-validated on virtual test bench and then experimentally validated on real test bench under null external load condition.

In the following parts, for simplifying the statement, SFF will stand for static force fighting; DFF will stand for dynamic force fighting; SFE stands for static force equalization; DFE stands for dynamic force equalization.

4.1 Sources of Static Force Fighting

In chapter 2 and 3, through simulations on linear model and virtual test bench, it has been shown that the SFF is mainly driven by the actuator static position error and the transmission stiffness between actuator and load. In case the setting and manufacturing tolerances are not
considered, the position static error is mainly influenced by the actuator closed loop stiffness. In this condition, the system static behaviors can be presented as:

\[
X_t = X_{sp} + X_{st} = X_{sp} + X_{et}
\]  

(4.1)

where \(X_{sp}\) and \(X_{ep}\) are the actuator length from frame to rod end, \(X_{st}\) and \(X_{et}\) are length of the transmission part.

While the load moves from position 0 to position 1, leading to:

\[
X_{t1} - X_{t0} = \left(X_{sp1} + X_{st1}\right) - \left(X_{sp0} + X_{st0}\right)
= \left(X_{sp1} + X_{st1}\right) - \left(X_{sp0} + X_{st0}\right)
\]  

(4.2)

where \(X_{t0}\) is the initial position under null external force (all the system states can be considered starting from this condition), \(X_{t1}\) is the destination position. This leads to:

\[
\left(X_{sp1} - X_{sp0}\right) + \left(X_{st1} - X_{st0}\right) = \left(X_{sp1} - X_{sp0}\right) + \left(X_{et1} - X_{et0}\right)
\]  

(4.3)

As the actuator body is also considered as stiff, the actuator length change comes from the rod extension, \(X_s\) for SHA and \(X_e\) for EMA, leading to:

\[
\begin{align*}
X_{sp1} - X_{sp0} &= X_s \\
X_{ep1} - X_{ep0} &= X_e
\end{align*}
\]  

(4.4)

As the SHA and EMA are operating under common position demand signal \(X_r\), the static rod extension can be calculated as:

\[
\begin{align*}
X_r - X_s &= S_{sp}^{-1}F_s \\
X_r - X_e &= S_{sp}^{-1}F_e
\end{align*}
\]  

\[
\begin{align*}
X_s &= X_r - S_{sp}^{-1}F_s \\
X_e &= X_r - S_{sp}^{-1}F_e
\end{align*}
\]  

(4.5)
Meanwhile, the compression of transmission part can be calculated as:

\[
\begin{align*}
X_{st1} - X_{st0} &= -S_{st}^{-1}F_s \\
X_{et1} - X_{et0} &= -S_{et}^{-1}F_e
\end{align*}
\]  

(4.6)

Substituting equations 4.4, 4.5 and 4.6 to equation 4.3, leading to:

\[
S_{sp}^{-1}F_s + S_{st}^{-1}F_s = S_{sp}^{-1}F_e + S_{st}^{-1}F_e
\]  

(4.7)

Until now, it is assumed that the actuator position static errors \(X_r-X_s\) and \(X_r-X_e\) are only fixed by the closed loop stiffness, the static error caused by setting and manufacturing tolerances are not considered. But in fact, these uncertainties are impossible to avoid. Therefore, in order to improve the robustness of force equalization strategies, these factors are necessary to be considered. Two general position offsets \(E_s\) and \(E_e\) are introduced in equation 4.7 to represent the influence of these uncertainties:

\[
\begin{align*}
S_{sp}^{-1}F_s + S_{st}^{-1}F_s + E_s &= S_{sp}^{-1}F_e + S_{st}^{-1}F_e + E_e \\
F_s + F_e &= F_L
\end{align*}
\]  

(4.8)

where:

<table>
<thead>
<tr>
<th>(E_s)</th>
<th>Position offset caused by uncertainties in SHA channel ([m])</th>
</tr>
</thead>
<tbody>
<tr>
<td>(E_e)</td>
<td>Position offset caused by uncertainties in EMA channel ([m])</td>
</tr>
</tbody>
</table>

As indicated in equation 4.8, the SHA and EMA static output forces \(F_s\) and \(F_e\) are fixed by three factors: closed loop stiffness, transmission stiffness and position offset. It means while both SHA and EMA are position controlled, in order to equal \(F_s\) and \(F_e\), the only solution is adjusting at least one of these three items. The candidate methods include:

- **Adjusting closed loop rejection stiffness**: this stiffness is mainly fixed by the position controller. For the controllers designed on the basis of classical control theory [62], as the control parameters are constant in a long period, while they are valued, the rejection stiffness is also determined and nonadjustable. So, in order to frequently adjust rejection stiffness, one solution is using intelligent controllers. A example is adaptive control which can adjust the control parameters according to the designed control laws in each control cycle [63]. However, for safety critical applications in aviation, the intelligent controllers are generally not accepted because their uncertain robustness that makes certification impossible. Another solution is pole placement with independent pursuit and regulation.
objectives [73], it could be used but it requires state observer and/or multiple sensors (we are demanded using less and mature sensors). So finally this method is not selected in the present work.

- **Adjusting transmission stiffness**: in fact, once the device is designed and produced, its natural stiffness is determined and almost not affected in service time. So, this stiffness is uncontrollable and this method is also unfeasible. But minimizing the structural stiffness difference between two actuation channels can greatly benefit the force equalization, as already mentioned.

- **Adjusting position offset**: the position offsets come from mechanical offset (due to the manufacturing tolerances and the integration of sensors, actuators) and electrical offset (quantization, biases, etc). Obviously, the mechanical one is not adjustable in real time. Oppositely, the electrical offset can be easily adjusted. Therefore, the solution is designing a force fighting controller to compensate for the actuator’s position control by modifying the electrical offset at the FCC level.

So finally, it appears that only the third candidate solution adjusting electrically the position offset is easily feasible for SFE. This modifies the equation 4.8 to:

\[
\begin{align*}
S_{mp}^{-1}F_s + S_{s}^{-1}F_s + E_s &= S_{ep}^{-1}F_e + S_{e}^{-1}F_e + E_e + E_0 \\
F_s + F_e &= F_{es}
\end{align*}
\]  

(4.9)

where \(E_0\) is the compensated position offset, unit [m].

For validating this principle of operation, a simulation is run on virtual test bench to evaluate the influence of \(E_0\) on adjusting actuator output force \(F_s\) and \(F_e\). A position offset signal \(E_0\) is progressively added on the SHA position sensor’s output. The position demand, external force, and \(E_s\) and \(E_e\) are assumed as zero. The SHA and EMA output force \(F_s\) and \(F_e\) are observed, as displayed in Fig 4-2.

In the tested range of position offset (-2~2mm, 8% of full stroke), the SHA and EMA output force \(F_s\) and \(F_e\) already change in the whole operation domain. They are in bijection relation with the position offset, which means it is feasible to change the \(F_s\) and \(F_e\) and find out a balance point through adjusting the offset of position sensor. Even if the \(E_s\) and \(E_e\) is not null
as assumed, the $F_s$ and $F_e$ curves only make a horizontal move in x axis, the final conclusion is similar.

The nonlinearity of $F_s$ and $F_e$ around zero offset area is mainly due to the backlash within the transmission joints (as already mentioned in Fig 3-15) and the servovalve pressure gain. The saturation of output force in larger offset area comes first from the reduction in servovalve flow gain then from the hydraulic supply pressure. In addition, for ensuring a reasonable level of segregation, the $E_0$ is saturated in a range of ±1mm in following parts. As the $E_s$ and $E_e$ are normally a small value, this saturation range of $E_0$ is already large enough to ensure correct static force equalization.

In fact, this solution has been widely used in redundant actuation systems composed by SHAs or even HMAs. The common strategy is getting a voted load pressure signal from all the channels and then introducing it to compensate the position controller of each channel, as illustrated in Fig 4-3, where the $\Delta P_1$, $\Delta P_2$ and $\Delta P_3$ are the load pressures of each channel, $\Delta P_m$ is the voted pressure, $H_1(s)$, $H_2(s)$ and $H_3(s)$ are the transfer functions for SFE, $E_{01}$, $E_{02}$ and $E_{03}$ are the calculated position offsets for each channel.
It is known that the hydraulic actuators have minor force losses (rod inertia and seal/bearing friction, as already mentioned in Tab 3-4). For this reason, measuring the chambers pressure is an efficient and cheap solution to get an estimate of the actuator output force, as given by equation 4.10 (symmetrical jack is assumed):

\[ F_H = P_L A - F_f - M_s X_s \]

\[ \approx P_L A \quad (4.10) \]

where \( F_H \) is hydraulic actuator output force, \( P_L \) is the load pressure, \( A \) is the hydrostatic area, \( F_f \) is the jack friction, \( M_s \) is the rod mass and \( X_s \) is the rod displacement.

In this strategy, while all the \( \Delta P_x \) equal to \( \Delta P_m \) it is considered that all the actuators develop the output forces. About the voting laws, there are two candidate solutions: middle value or average value. In practice, the average value rule is more widely used [22]. The saturations after the force equalization transfer function \( H_x(s) \) are set for segregation reason, as already mentioned.

In summary, for SFE, a potential solution is introducing a force fighting compensator to adjust the position offset added on actuator position feedback signal.

### 4.2 SFE-1 Integral Force Fighting Feedback

According to the above analysis, the first SFE strategy is proposed, as displayed in Fig 4-4.
In this SFE strategy, the integral signal of force fighting $\gamma$ is used to generate a position offset in each position control loop of the SHA and EMA through an integral gain $k \ [m/Ns]$. I control is selected because of its long term, low frequency effects, a high gain in statics and very low influence on the dynamics which decreases at a rate of -20dB/dec. While the $F_s$ and $F_e$ are different, the integral will continually adjust the position offset $E_{0s}$ and $E_{0e}$ until $F_s$ and $F_e$ are equaled, this is done only on low frequency domain for SFE.

In this strategy, the position controllers designed for SHA and EMA in chapter 2 are involved, so the only parameter to be valued is the integral gain $k$ for the SFE. As already mentioned in chapter 1, the proposed SFE strategies should not alter the system performance on pursuit, rejection and stability. Therefore, the gain $k$ should be valued to meet all these requirements. A feasible solution is getting an approximate value for $k$ through linear approach, which can theoretically ensure the system stability, pursuit and rejection performance. Then, the value is tuned and pre-validated on virtual test bench.

At first, the system transfer function under this SFE strategy is calculated. The SHA servovalve current input can be written as:

$$I_{sv} = (X_r - X_s - E_s - E_{0s}) K_{sb} \quad (4.11)$$

where:

$$E_{0s} = \frac{k}{s} \gamma$$

The EMA motor torque demand can be presented as:
Chapter 4 Static Force Equalization Study

\[ T_d = \left( X_r - X_e - E_e + E_{0e} \right) K_{oh} - \dot{X}_e \right) K_{ec} \quad (4.12) \]

where:

\[ E_{0e} = \frac{k}{s^\gamma} \]

As the \( E_e \) and \( E_e \) have little influence on dynamics and stability, and their effects on statics can be also presented by the \( E_{0e} \), they are assumed as null in following calculation.

Then, the equations 4.11 and 4.12 are substituted in equations 2.38, 2.39 and 2.40 to get the transfer function of load displacement:

\[ X_f = \frac{U_s X_r - U_2 F_L}{U_2 M_s s^2 + U_1} \quad (4.13) \]

where:

\[ U_1 = \left( G_1 S_{et} + H_1 S_{st} \right) s + S_{et} S_{et} \left( G_2 + H_2 \right) s + 4 S_{et} S_{et} k \]

\[ U_2 = \left( G_1 + G_2 S_{et} \right) \left( H_1 + H_2 S_{st} \right) s^2 + \left( G_1 S_{et} + H_1 S_{st} \right) k s + S_{et} S_{et} \left( G_2 + H_2 \right) k s \]

\( 1/G_1 \): SHA pursuit transfer function, \( G_2/G_1 \): SHA rejection transfer function, \( 1/H_1 \): EMA pursuit transfer function, \( H_2/H_1 \): EMA rejection transfer function, definition see Chapter 2.3.1.

As shown in equation 4.13, the integral gain \( k \) influences both statics and dynamics. According to *Routh Stability Criterion*, the \( k \) value is found playing an important role on system stability. The critical value is \( k=2.3 \times 10^{-6}[m/\text{Ns}] \). During calculating this critical value, the transmission stiffnesses are linearized as \( S_{st}=2.9 \times 10^7[N/m] \) and \( S_{et}=2.5 \times 10^8[N/m] \), which are the stiffness values out of backlash domain, as mentioned in chapter 3.1.7. Meanwhile, it is observed that the \( k \) is always appearing with the \( S_{st} \) and \( S_{et} \) in the form of multiplying. It shows that the \( S_{st} \) and \( S_{et} \) have great influence on deciding the critical value of \( k \). The influence can be observed in Fig 4-5, where it is assumed that the SHA and EMA have identical transmission stiffnesses \( S_{st}=S_{et} \).
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Fig 4-5 Influence of transmission stiffness on critical $k$ value in case $S_{st}=S_{set}$

As indicated in Fig 4-5, the influence of $S_{st}$ and $S_{set}$ on critical value of $k$ is mainly focused on the low stiffness area up to $2\times10^6$ N/m. In high stiffness area, their influence is small.

As mentioned in chapter 2, on linear prototype, the SFF is only due to the external load $F_L$ (while $E_s$ and $E_e$ are neglected). The SFE effect under this condition is assessed by:

$$
\gamma = \left[ \left( H_1 S_x - G_1 S_x \right) + S_x S_x \left( H_2 - G_2 \right) \right] M_x s^4 X_x + \left[ \left( H_1 S_x - G_1 S_x \right) + S_x S_x \left( H_2 - G_2 \right) \right] s F_L \left( U_2 M_x s^2 + U_1 \right)
$$

According to equation 4.14, on linear model, while the system is stable with the selected $k$, the SFF will be totally equaled whatever the position demand or external force. Meanwhile, about the SFF caused by nonlinearities $E_s$ and $E_e$, according to the analysis in above section, it is also ensured could be removed.

Theoretically, the larger the $k$ value, the faster the SFF being equaled. But as mentioned above, the larger $k$ will worsen the system stability. So finally, a balance value should be selected. For choosing this value, several simulations are run on both linear model and virtual test bench.

The first group of simulations is run to check the performance under pursuit condition. The input is a 1mm position demand step at $t=0.1$s with null external force input. Several different $k$ values are set. The responses are displayed in Fig 4-6.
The second group is run to check the performance under rejection condition with an input of 10KN external force step occurring at t=0.1s and null position demand, as shown in Fig 4-7.

As shown in Fig 4-6 and Fig 4-7, the theoretical result (critical $k=2.3 \times 10^{-6}$ m/Ns) is verified in the linear simulation and in the virtual test bench as well. Meanwhile, while system is stable, such as the second one $k=1.0 \times 10^{-6}$ m/Ns, the SFF is well equaled whatever the type of input signal (position demand or external load) even in the virtual test bench that involves many nonlinearities.

However, the system pursuit and rejection performance become worse after introducing the integral force fighting compensation. The pursuit dynamics is less influenced, but the stability
is altered. The load rejection stiffness is decreased about twice, but it is only mattered by the
integral compensation path and has less relationship with the value of $k$. The poor rejection is
mainly due to the strong unbalance of transmission stiffness $S_{st}$ and $S_{et}$. The actuator with high
transmission stiffness will shrink to output less force in order to equal the low stiffness one
that does extension. This increases the load displacement leading to lower rejection stiffness.

One proposal for fixing this problem is unequally distributing the position offset $E_0$ in two
channels. The high stiffness channel is distributed less position offset and the low stiffness
one shares more position offset and does more extension. The schematic of this strategy is
given on Fig 4-8.

![Fig 4-8 Schematic of unequally distribution position offset $E_0$](image)

For this strategy, under rejection condition with only external force and null position demand,
in case the SFF is well equaled, the load displacement can be calculated as:

$$
\begin{align}
\Delta X_t &= \frac{F_s}{S_{sp}} + \frac{F_e}{S_{st}} - E_{0s} = \frac{F_s}{S_{sp}} + \frac{F_e}{S_{et}} + E_{0e} \\
F_s &= F_e = \frac{1}{2} F_L
\end{align}
$$

(4.15)

This leads the load rejection stiffness to be:

$$
\begin{align}
S_L = \frac{F_L}{\Delta X_t} &= \frac{1}{2\frac{S_{sp}}{S_{et}} + \frac{1}{2\frac{S_{st}}{S_{et}}} - E_{0s}} = \frac{1}{\frac{S_{sp}}{S_{et}} + \frac{1}{2\frac{S_{st}}{S_{et}}} + E_{0e}} \\
E_{0s} &= E_0 k_1 \\
E_{0e} &= E_0 k_2 \\
k_1 + k_2 &= 1
\end{align}
$$

(4.16)

where $S_L$ is the load rejection stiffness, unit [N/m].

As the $S_{sp}$, $S_{st}$, $S_{ep}$ and $S_{et}$ are all constant, the general position offset $E_0$ is also constant under
common external load force $F_L$, the load rejection stiffness is fixed by how the $E_{0s}$ and $E_{0e}$ are
distributed. For example, in our case, the EMA has higher structural stiffness, so the less the
$E_{0e}$ is distributed, the larger the load static rejection stiffness. Generally, although this method
increases the SFE transient duration, it is an attractive solution to obtain a high load rejection
stiffness.

The large difference between linear prototype and virtual test bench while $k=5 \times 10^{-6}[m/Ns]$ is caused by the power supply limitation in virtual test bench.

So finally, according to the above analysis and simulations, the balance value of $k$ is selected as $k=1 \times 10^{-6}[m/Ns]$. With this value, the SFE, stability, load pursuit and rejection requirement can all be met.

In addition, during simulations in virtual test bench, the dynamic sections are found having influences on the critical value of $k$, like the SHA servovalve dynamics, EMA motor dynamics and the ADC sampling rate. In order to further study these influences, some simulations are run on virtual test bench. The dynamics of the above mentioned or components are changed individually to illustrate how they influence the system stability in the presence of the integral SFE strategy, as shown in Fig 4-9.

![Fig 4-9 Influence of main components dynamics on critical k value](image)

It is observed in Fig 4-9 that up to $k=1.4 \times 10^{-6}[m/Ns]$, the system stability is driven by the SHA servovalve dynamics. For the selected value of $k=1 \times 10^{-6}[m/Ns]$, the dynamics of motor torque loop could be reduced to 50Hz or (exclusive) the sampling could be reduced to 170Hz (only on the view of system stability, of course the dynamics become poor with a so low motor torque loop dynamics).

Moreover, in order to accurately present the force fighting, two force sensors are installed on
Chapter 4 Static Force Equalization Study

the rod ends of SHA and EMA. Using the motor current signals to estimate the EMA force is very weak due to the huge torque loss caused by roller-screw friction and motor rotor inertia in dynamics, as indicated in Fig 3-13. In Fig 4-10, the SFE effects of the present strategy with different force fighting estimation way are evaluated on the virtual test bench. The excitation is a position offset added on the SHA position sensor and it changes as a slow triangular wave in the range of ±1mm.

A first-order low pass filter (time constant 0.1s) is used on the SHA pressure sensor and EMA motor current sensor to filter the high frequency noise. It is found that the two estimation solutions involving motor current sensor do not work well. Oppositely, the second solution using pressure signal to estimate SHA force has a good performance, although not as smooth as the first solution which involves two force sensors.

As the roller-screw friction involves many uncertainties, using friction model to compensate the motor current signal for estimating EMA force is not a robust solution. So, an easier and acceptable solution is introducing a force sensor to estimate EMA output force. Meanwhile, a force sensor is installed on the rod of SHA to keep consistency with EMA.

In summary, the integral force fighting compensation strategy is proved having a good overall
performance on SFE. The force fighting integral gain $k$ is set as a middle value of $1 \times 10^{-6}[m/Ns]$ to balance stability and SFE dynamics. The transmission stiffness and servovalve dynamics are found having big influence on the admissible value of $k$. And using a force sensor for EMA is mandatory.

### 4.3 SFE-2 One Position/One Force Control

In the above proposed SFE strategy, both SHA and EMA are position controlled. As another candidate strategy, one actuator could be position controlled while the other one could follow the first one as a slave in force control. The idea comes from the MIMO (multi-input multi-output) decoupled system, in which each output is only controlled by one input. The common MIMO decoupling strategy is considering the system as a whole and designing an integrated controller for all the outputs. This causes many cross links between channels leading to worse segregation, as illustrated in Fig 4-11. For safety critical applications in aviation, as our case, the segregation between SHA controller and EMA controller should be kept to ensure the immunity against failure propagation between channels. Therefore, the integrated decoupling control is not selected although it could be an option in another context [63, 64].

![Fig 4-11 Schematic of redundant actuation system](image)

Our solution is dividing the decoupling control task into two parts: one actuator as position control to be in charge of load displacement performance and one actuator as force control following the other actuator force to meet the force equalization requirements. This suggests two candidate configurations: either the EMA or the SHA is force controlled. In this section, these two configurations will be studied and assessed individually.

#### 4.3.1 SFE-2.1 SHA Force/EMA Position Control

The first studied candidate configuration is SHA force controlled and EMA position controlled.
Chapter 4 Static Force Equalization Study

The cascade position controller designed for EMA in chapter 2 is involved. A force controller is proposed for SHA, which operates as the slave of EMA force. In case the SHA output force \( F_s \) can well track the EMA output force \( F_e \), the force fighting will be decreased. The schematic of this strategy is shown in Fig 4-12.

![Fig 4-12 Schematic of SHA force controlled and EMA position controlled](image)

The SHA force controller is designed in following steps: at first set using the linear approach and then adjusted and validated on the virtual test bench.

As a preliminary approach and to make it simple, the SHA is force controlled through a pure P action. The P gain is \( K_{sf} \frac{[A/N]}{} \). In this case, the servovalve current can be written as:

\[
I_{sv} = (F_e - F_s)K_{sf}
\] (4.17)

Then substituting equation 4.17 to the relative equations in chapter 2, the load displacement can be expressed as:

\[
X_i = \left[ \frac{V_s}{4E_sK_c}s + 1 + 2G_s \right]S_{ep}X_r - \left[ \frac{V_s}{4E_sK_c}s + 1 + G_s \right]\left(1 + \frac{S_{ep}}{S_{et}}\right)F_L
\] (4.18)

where:

\[
G_s = \frac{A_sK_{sg}K_{sf}}{K_c}D_s, \quad \text{and } D_s \text{ is the servovalve dynamics: second order, 76Hz natural frequency and 1.283 damping factor:}
\]

\[
S_{ep} = \frac{2\pi K_{eb}K_{ec}}{I}
\]

\[
a_3 = \frac{V_s\pi^2J_m}{E_sK_cI^2} + \frac{V_sM_f}{4E_sK_c}\left(1 + \frac{S_{ep}}{S_{et}}\right)
\]
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\[ a_2 = \frac{4\pi^2 J_m}{l^2} (1+2G_s) + \frac{V_s B_s}{4E_s K_c} \left( 1 + \frac{S_{ep}}{S_{et}} \right) + \frac{V_s \pi^2 B_s}{2E_s K_c l^2} + \frac{V_s \pi K_{ec}}{2E_s K_c l} + (1+G_s) \left( 1 + \frac{S_{ep}}{S_{et}} \right) M_t \]

\[ a_1 = (1+2G_s) \left( \frac{4\pi^2 B_s}{l^2} + \frac{2\pi K_{ec}}{l} \right) + \left( 1 + \frac{S_{ep}}{S_{et}} \right) \left( \frac{A_t^2}{K_c} + B_s \right) + \frac{V_s S_{ep}}{4E_s K_c} \]

At first, the influence of \( K_{sf} \) on system stability is studied according to Routh Stability Criterion with the system characteristic polynomial. The critical value is \( K_{sf}=1.69\times10^{-6} [A/N] \). As a result, \( K_{sf}\leq1.69\times10^{-6} [A/N] \) for ensuring system stability.

Then, according to equation 4.18, the static error of load displacement is null under pursuit condition (with position demand \( X_r \)) and only exists under rejection condition (with external force \( F_L \)). With this, the load rejection stiffness \( S_L \) can be expressed as:

\[ S_L = \frac{(1+2G_s) S_{ep}}{(1+G_s) \left( 1 + \frac{S_{ep}}{S_{et}} \right)} = \frac{S_{ep} S_{et}}{S_{et} + S_{et}} \left( 1 + \frac{A_t K_{sg} K_{sf}}{K_c} \right) \left( 1 + \frac{A_t K_{sg} K_{sf}}{K_c} \right) \]  

(4.19)

According to equation 4.19, the load rejection stiffness \( S_L \) is driven by two items:

- **Item 1**: equivalent stiffness of EMA channel, which comes from the series combination of EMA transmission stiffness \( S_{et} \) and EMA closed loop stiffness \( S_{ep} \). With linearized values \( S_{et}=2.5\times10^8 \text{N/m} \) (Tab 3-5), \( S_{ep}=2\times10^8 \text{N/m} \) (Fig 2-13), this item equals to \( 1.11\times10^8 \text{N/m} \).
- **Item 2**: SHA parameters involves servovalve flow/current gain \( K_{sg} \), flow/pressure gain \( K_c \), hydrostatic area \( A_t \) and force control gain \( K_{sf} \). Theoretically, the larger the \( K_{sf} \), the larger this item.

The influence of \( K_{sf} \) on load rejection stiffness is illustrated in Fig 4-13.

Accordingly, in order to meet both the stability and load rejection, the \( K_{sf} \) should be set in the range of \( K_{sf} \in [1.3\times10^{-7}, 1.69\times10^{-6} \text{A/N}] \).
Then, moving to SFE, the force fighting $\gamma$ under this strategy can be written as:

$$\gamma = \frac{a_x X_r + a_s F_L}{a_x s + a_y s^2 + a_y s + (1 + 2G_r) S_{ep}} \quad (4.20)$$

where:

$$a_x = -S_{ep} \left[ \frac{V_m M}{4E_c K} s^3 + \left( \frac{V_B}{2E_c K} + M \right) s^2 + 2 \left( \frac{A^2}{K} + B_y \right) s \right]$$

$$a_y = -\frac{V_m \pi^2 J_m}{E_c K l^2} s^3 - \left[ \frac{4\pi^2 J_m}{E_c K l^2} + \frac{V_B \pi^2 B_y}{E_c K l} + \frac{V_B \pi K_{ec}}{2E_c K l} - \frac{V_B}{4E_c K l} \left( 1 + \frac{S_{ep}}{S_{et}} \right) \right] s^2 \ldots$$

$$\ldots - \left[ \frac{V_{ep} S_{ep}}{4E_c K c} + \frac{4\pi^2 B_y}{E_c K l} + \frac{2\pi K_{ec}}{l} - \left( 1 + \frac{S_{ep}}{S_{et}} \right) \left( \frac{A^2}{K} + B_y \right) \right] s - S_{ep}$$

According to equation 4.20, with linear approach, the SFF is null in pursuit condition and only exists in rejection condition (not consider the influence of position offset $E_s$ and $E_c$). The SFF due to external force $F_L$ can be expressed as:

$$\gamma = \frac{-1}{1 + \frac{2A^2 K_{sg} K_{sf}}{K_c}} F_L \quad (4.21)$$

It is observed that the SFF is totally fixed by the SHA force controller $K_{sg}$. While the system is stable, the larger the $K_{sg}$, the smaller the SFF is, as shown in Fig 4-14.

It should be noticed that the $K_{sg}$ and $K_c$ are the servovalve flow/current and flow/pressure
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Influence of \( K_{sf} \) on Static Force Fighting gains got from linearization around the operating point. In practice, their range of variation with operating conditions is not so large [53].

In order to meet the stability, rejection and SFE requirements, the \( K_{sf} \) value should be set in the range \( [3.7 \times 10^{-7}, 1.69 \times 10^{-6}] \text{A/N} \). According to this, simulations are run on linear prototype and virtual test bench to verify the above results and to define an acceptable value of \( K_{sf} \).

The first group of simulations sets the input as a 1mm position demand step at \( t=0.1 \text{s} \) with null external force, several different \( K_{sf} \) are tested, as displayed in Fig 4-15.

![Fig 4-15 Influence of valuing \( K_{sf} \) on system stability](image-url)
The second group of simulations is run with an input of 10KN external force step occurring at 
t=0.1s, as displayed in Fig 4-16.

![Graphs showing load displacement and force fighting](image)

**Fig 4-16** Influence of valuing $K_{sf}$ on force fighting

It is observed that setting $K_{sf}=5\times10^{-7}[A/N]$ can meet all the requirements with a satisfactory 
margin: the stability, pursuit dynamics, load rejection and SFE. The static force is well equaled 
between SHA and EMA whatever the type of excitation (position demand or external load).

While $K_{sf}$ becomes greater than $K_{sf}=5\times10^{-6}[A/N]$, the oscillation comes from the pumping effect 
between the two channels. As the SHA transmission stiffness is very low, it is mainly driven by 
the equivalent stiffness of EMA channel. The difference of oscillation frequency between the 
linear model and virtual test bench is caused by the decreasing transmission stiffness within 
the backlash domain that is only involved in virtual prototype.

In addition, according to the simulations in virtual test bench, the SHA transmission stiffness 
$S_{st}$ and the servovalve dynamics are also found having influence on setting the value of $K_{sf}$ for 
keeping system stability. Their influence is demonstrated in Fig 4-17.
In summary, in this section, a pure proportional force controller is designed for SHA because mainly a static view of the force equalization is addressed and not too much complex to study. The force control gain is set as $K_{sf}=5\times10^{-6}$ A/N, which can meet all the requirements. However, it is shown that the strong unbalance of the SHA and EMA transmission stiffness has very bad influence on the performance got with this SFE strategy.

### 4.3.2 SFE-2.2 SHA Position/EMA Force Control

In this part, the second candidate configuration with SHA position controlled and EMA force controlled is studied. The P plus Notch filter position controller designed for SHA in chapter 2...
is involved. A force controller is proposed for the EMA, which is demanded to slave the SHA output force. The schematic is displayed in Fig 4-18.

![Schematic of SHA position controlled and EMA force controlled](image)

**Fig 4-18** Schematic of SHA position controlled and EMA force controlled

Similarly as the first candidate strategy, a pure proportional force controller is designed for EMA as the preliminary approach with the gain as \( K_{ef} [1/m] \). The value of \( K_{ef} \) will be set on the basis of linear approach at first and then adjusted on virtual test bench.

Now, the EMA motor torque demand \( T_d \) should be expressed as:

\[
T_d = (F_s - F_e) K_{ef}
\]  
(4.22)

Then substituting equation 4.22 to the relative equations in chapter 2, the load displacement becomes:

\[
X_l = \frac{S_{sp} (1 + 2G_e) X_r - (1 + G_e) \left( \frac{V_s}{4E_s K_c} - s + \frac{S_{sp}}{S_m} \right) F_l}{b_3 s^3 + b_2 s^2 + b_1 s + S_{sp} (1 + 2G_e)}
\]  
(4.23)

where:

\[
S_{sp} = \frac{A K_{sa} K_{ab}}{K_c}
\]

\[
G_e = \frac{2\pi}{l} K_{ef}
\]

\[
b_3 = \frac{V_s M_r}{4E_s K_c} (1 + G_e) + \frac{V_s \pi^2 J_m}{E_s K_c l^2}
\]

\[
b_2 = \frac{V_s B_r}{4E_s K_c} (1 + 2G_e) + \frac{V_s \pi^2 B_e}{E_s K_c l^2} + M_r (1 + G_e) \left( 1 + \frac{S_{sp}}{S_m} \right) + \frac{4\pi^2 J_m}{l^2} \left( 1 + \frac{S_{sp}}{S_m} \right)
\]

\[
b_1 = \left( \frac{A^2}{K_c} + B_r \right) (1 + 2G_e) + \frac{4\pi^2 B_e}{l^2} \left( 1 + \frac{S_{sp}}{S_m} \right)
\]
At first, according to Routh Stability Criterion, the influence of valuing $K_{ef}$ on system stability is studied. Through calculation, while $K_{ef}$ is positive, the system is always asymptotically stable. The SHA transmission stiffness $S_{st}$ is found having great influence on the acceptable value of $K_{ef}$ for practical stability.

- While $S_{st}$ is lower than $1 \times 10^8 [N/m]$, the Routh Stability Criterion is always satisfied with a positive $K_{ef}$.
- While $S_{st}$ is larger than $1.4 \times 10^8 [N/m]$, the Routh Stability Criterion is always fault no matter the value of $K_{ef}$.
- While $S_{st}$ is between these two value, the stability is fixed by the value of $K_{ef}$.

In order to clearly present the relationship between $S_{st}$ and $K_{ef}$, several root locus graphs are drawn to display the poles of the transfer function equation 4.23 as a function of $K_{ef}$ for 4 values of $S_{st}$ as illustrated in Fig 4-19.

As the linearized value of SHA transmission stiffness $S_{st}=2.9 \times 10^7 N/m$ is smaller than the critical value, the actuation system is stable as far as $K_{ef}$ is positive.

Moreover, as got from equation 4.23, the load static error is zero while operating under null external load. The load rejection stiffness can be presented as:
Similar to the above strategy, the load rejection stiffness under this strategy is also driven by two items:

- **Item 1**: equivalent stiffness of SHA channel as a combination of series SHA transmission stiffness $S_{st}$ and SHA closed loop stiffness $S_{sp}$. Due to the low stiffness of $S_{st}=2.9\times10^7 \text{N/m}$, even the $S_{sp}=5.0\times10^8 \text{N/m}$ is very high, the equivalent value is only $2.74\times10^7 \text{N/m}$.

- **Item 2**: this item is mainly fixed by the $K_{ef}$. Its varying range is $[1, 2]$ with increasing $K_{ef}$.

So, due to the low equivalent stiffness of SHA channel, even with a very large value of $K_{ef}$, the final load rejection stiffness is still far lower than required. However, on real aircrafts, with a higher $S_{st}$, this situation will be improved.

After that, the force fighting $\gamma$ is written as:

$$\gamma = \frac{b_x X_e + b_y F_L}{b_3 s^3 + b_4 s^2 + b_5 s + S_{sp} \left( 1 + 2G_e \right)}$$

(4.25)

where:

$$b_3 = S_{sp} \left[ \left( M_i + \frac{8\pi^2 J_m}{l^2} \right) s^2 + \frac{8\pi^2 B_e}{l^2} s \right]$$

$$b_4 = -\frac{V_e \pi^2 J_m}{E_s K_e l^2} s^3 - \left[ \frac{V_e \pi^2 B_e}{E_s K_e l^2} + \frac{4\pi^2 J_m}{l^2} \left( 1 + \frac{S_{sp}}{S_{st}} \right) - \frac{V_e B_s}{4E_s K_e} \right] s^2 \ldots$$

$$\ldots - \left[ \frac{4\pi^2 B_e}{l^2} \left( 1 + \frac{S_{sp}}{S_{st}} \right) - \left( \frac{A_e^2}{K_e} + B_e \right) \right] s + S_{sp}$$

According to equation 4.25, if the position offset $E_e$ and $E_s$ are not considered, the SFF caused by the position demand is null. The SFF only exists under external load, it can be written as:

$$\gamma = \frac{1}{1 + 2G_e} F_L = \frac{1}{1 + \frac{4\pi}{l} K_{ef}} F_L$$

(4.26)
As indicated in equation 4.26, the SFF caused by $F_l$ is driven by the EMA force controller $K_{ef}$. The value of $K_{ef}$ should be $K_{ef} \geq 5.7 \times 10^{-3} [1/m]$ to meet the SFE requirement.

As analyzed above, on the basis of linear approach, the actuation system is stable with any positive value of $K_{ef}$. But the simulation results on virtual test bench do not confirm this non-realistic conclusion, where the system becomes unstable when $K_{ef}$ increases. Through analysis, the instability is caused by the dynamic sections in EMA channel which are not considered in linear approach, such as the motor torque control dynamics and signal processing dynamics. After introducing these dynamics into the linear prototype in Matlab/Simulink environment, the system also becomes unstable when $K_{ef}$ increases.

The influence of EMA motor torque control dynamics on setting the value of $K_{ef}$ is studied on the basis of virtual test bench, as shown in Fig 4-20.

![Critical Value of $K_{ef}$](image)

**Fig 4-20** Influence of motor torque loop dynamics on setting value of $K_{ef}$ for stability

It is observed that the better EMA motor torque control dynamics leads to a better stability whatever the value of transmission stiffness. But a higher transmission stiffness will decrease the acceptable range of $K_{ef}$.

However, within the stability range of $K_{ef}$ indicated in Fig 4-20, it is impossible to define a value for $K_{ef}$ to meet both the load rejection and the SFE requirements. In order to verify this conclusion, several simulations are run on virtual test bench with different value of $K_{ef}$.

The first group of simulations sets an input of 1mm position demand step at $t=0.1$s with null...
external load, the responses are shown in Fig 4-21.

![Fig 4-21](image)

**Fig 4-21** System pursuit performance under different value of $K_{ef}$

The second group of simulations sets an input of 10KN external force step occurring at $t=0.1s$, the responses are shown in Fig 4-22.

![Fig 4-22](image)

**Fig 4-22** System rejection performance under different value of $K_{ef}$

As shown in Fig 4-21 and Fig 4-22, the above mentioned conclusions about the stability, load rejection and SFE are verified both on linear prototype and virtual test bench. Meanwhile, the
load pursuit performance is also observed. A big overshoot appears on the response of load displacement. This is mainly due to the huge inertia of EMA. With this pure P force controller, this overshoot is very difficult to be removed. The oscillation with a frequency of 14Hz while $K_{ef}=4\times 10^{-3}[1/m]$ is mainly driven by the EMA force control loop and its huge rotary inertia.

In summary, in this section, a pure proportional force controller is proposed for EMA which is the slave of the SHA force. However, through analysis, this force controller cannot meet all the requirements at the same time involving the stability, pursuit, load rejection and SFE.

Then combining the above simulation results, the performances of the two master/slave candidate configurations are summarized and compared in the following Tab 4-1.

<table>
<thead>
<tr>
<th>Tab 4-1 Summary of two configurations with one force controlled and one position controlled</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Conf</strong></td>
</tr>
<tr>
<td>SHA Force/EMA Position</td>
</tr>
<tr>
<td>SHA Position/EMA Force</td>
</tr>
</tbody>
</table>

As summarized in Tab 4-1, the first candidate configuration (SHA slave/EMA master) has much better performance than the second one (SHA master/EMA slave). In order to improve the performance of the second one, the following could be done:

- **Load rejection**: could be improved by increasing the SHA transmission stiffness $S_{st}$.
- **Static force equalization**: could be met by introducing an integral control.
- **Load pursuit**: could be improved by using a complex controller. But generally, this is very difficult to achieve. As the huge inertia and roller-screw friction acting on the EMA motor shaft, while reflected on rod level, they are magnified by $2\pi/l$ (2094) times and greatly alter the EMA dynamics. While EMA is position controlled, this effect is offset by its position controller; but while EMA is force controlled, this effect will be reflected on the SHA rod level whose position controller is only designed for a relative smaller inertia. Opposite to this, the inertia of SHA is very small. While it is reflected on EMA rod level, its influence can be neglected.
So generally, considering the difficulty on obtaining a good pursuit dynamics for the second candidate configuration, the first one is more attractive for this force equalization strategy.

### 4.3.3 SHA Position/EMA No Load Control

In the above mentioned strategies, the SHA and EMA operate in active/active mode. Although it is not in straight line with the force equalization issues, it is interesting to make "electrical" declutching of the EMA in case of active/standby design.

In hybrid actuation, the back-driving friction and the rotor inertia of EMA oppose high loads when the power bridge of the EMA electronic power drive is simply opened. This situation may be not acceptable. To fix this issue, it is proposed to investigate the feasibility of "electrical" declutching on the EMA by forcing it to oppose null load (as far as the EMA is not failed).

At first, the schematic of actuation system under this operation mode is shown in Fig 4-23.

![Figure 4-23: Schematic of EMA no-load control strategy](image)

It is observed that the only difference between this strategy and the above mentioned SHA master/EMA slave one is the force demand of EMA. According to the calculations on linear approach, the properties of these two strategies are similar. So, as a preliminary investigation, the force controller designed for EMA in above part is involved. The P force control gain is set to $K_{fe}=4 \times 10^{-4}[1/m]$.

Then, simulations are run on virtual test bench to compare the performance of EMA no power mode and EMA "electrical" declutching mode. The SHA is inputted a position demand step of 1mm at $t=0.1s$ and following a 10KN external force step acting on load occurring at $t=1s$. The responses are shown in Fig 4-24.
Chapter 4 Static Force Equalization Study

It is observed that the main improvement comes from the pursuit dynamics and decreasing of EMA back-driving load in static which leads to a lower static error under pursuit condition. The worse load rejection is due to the strong unbalance of transmission stiffness between two channels. In case the SHA transmission stiffness could be made equal to the EMA one $S_r=S_{st}=2.5\times10^8\text{N/m}$, this situation will be improved. A simulation with this assumption is run, and the inputs are same as the above one. The responses are displayed in Fig 4-25.

Fig 4-24 Performance of EMA no power mode and electrical declutching mode

Fig 4-25 Performance of EMA electrical declutching control with high transmission stiffness
As shown in Fig 4-25, after improving the SHA transmission stiffness $S_{st}$, the load rejection stiffness increases and meets the requirement. In this case, the reducing of EMA back-driving load mainly presents on rejection condition, where the EMA outputs almost null force.

In summary, even with a simple P force controller, the EMA no load control strategy already shows a good performance on reducing the EMA opposed load while it operates on standby mode. So generally, this EMA “electrical” declutching strategy is a potential solution for addressing the EMA huge opposed load issue, as far as the EMA is functional.

4.4 Experimental Validation of SFE Strategies

Once validated on virtual test bench, the proposed SFE strategies are to be evaluated through experiments on real test bench. The integral force fighting feedback strategy (SFE-1, IFFF), the SHA force control/EMA position control strategy (SFE-2.1, SF/EP) and EMA no-load control strategy (SP/ENL) are selected to be experimented. The experiment input is a slowly changing position offset added on SHA position sensor signal. It varies from 0 to 5mm in 20s, then from 5 mm to -5mm in 40s, finally returns to null in 20s. As displayed in Fig 4-26, the amount of force fighting $\gamma$ is plotted as a function of the parasitic offset.
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The SFE requirement is 4% of the rated force 50KN making 2KN. As indicated in Fig 4-26, all these SFE strategies have good performance. The SFF is reduced from 40KN to lower than 1KN, well meeting the SFE requirement. Among these strategies, the SFE-1 IFFF is the most stable one without low frequency vibration in force fighting.

The time delay effect appearing on SFE-1 (location of A) is caused by the input dynamics, a much slower input shows that the magnitude of force fighting is proportionally decreased. Oppositely, the time delay effects appearing on SFE-2 and SP/ENL are mainly driven by the force controller, a slower input does not change a lot this "hysteresis like" effect.

In addition, as shown in Fig 4-26, by comparing with the experimental results, the virtual test bench built in chapter 3 is proved again having very good representation of real behaviors.

4.5 Conclusion of Chapter 4

In present chapter, the static force equalization of the hybrid redundant actuation system was addressed carefully: although the SFE strategy has to be stable, no attention has been paid to the amount of DFF during transients. The sources of static force fighting were firstly studied. The actuators closed loop stiffness, transmission stiffness and position offset have been found playing a major role on SFF. It has been shown that the SFF could be significantly reduced by equaling the transmission stiffness (by design) and adjusting the position offset.

Then, on the basis of this, 3 SFE strategies were proposed. The first one introduced an integral force fighting feedback (SFE-1, IFFF) to adjust the actuator position signals for achieving force equalization; the second one set SHA as force controlled to slave the output force of position controlled EMA (SFE-2.1, SF/EP); the third one was opposite to the second one and set SHA as position controlled (SFE-2.2, SP/EF). Through analysis, the SFE-2.2 was found unsatisfied due to the poor performance on load pursuit and SFE which was mainly influenced by the huge EMA rotary inertia.

In addition, the interest of making the EMA operating in flight control was also addressed for the active/standby operation mode. In this case, the EMA was force controlled to output null force, where the SHA was position controlled to energize the load (SP/ENL).
In the end, the SFE-1, SFE-2.1 and SP/ENL strategies were validated on test bench after being pre-validated on virtual test bench. According to the experimental results, they were proved all having good performance on meeting the SFF requirement.

The properties of proposed control strategies are summarized in Tab 4-2.

<table>
<thead>
<tr>
<th>Item</th>
<th>SFE-1 IFFF</th>
<th>SFE-2.1 SF/EP</th>
<th>SFE-2.2 SP/EF</th>
<th>SP/ENL</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stability</td>
<td>++</td>
<td>+</td>
<td>+</td>
<td>0</td>
</tr>
<tr>
<td>Pursuit</td>
<td>++</td>
<td>++</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Rejection</td>
<td>+</td>
<td>++</td>
<td>-</td>
<td>--</td>
</tr>
<tr>
<td>SFE</td>
<td>++</td>
<td>++</td>
<td>-</td>
<td>+</td>
</tr>
<tr>
<td>General</td>
<td>++</td>
<td>++</td>
<td>0</td>
<td>+</td>
</tr>
</tbody>
</table>

As they show good performance, the SFE-1 and SFE-2.1 will be involved in following design of the dynamic force equalization strategies.

**Important Conclusions of Present Chapter:**

- **Conclusion 1:** Adding a position offset on the position control loops is an effective way to ensure SFE. When implemented with a bounded integral action from the force unbalance (SFE-1), the SFE and the closed loop performance can be simultaneously ensured.

- **Conclusion 2:** The master/slave force control proposed in SFE-2 works well when SHA is the slave actuator.
Chapter 5
Dynamic Force Equalization Study

In chapter 4, the static force equalization issue of hybrid actuation system was addressed and validated by simulations and experiments. In present chapter, the dynamic force equalization shall be studied. Three DFE control strategies are proposed and if necessary, combined with the SFE. Then, these strategies are evaluated on virtual test bench. In the end, the robustness of these control strategies against parameter uncertainties is studied by design exploration.

As concluded in chapters 2 and 3, the DFF is mainly driven by the actuator pursuit/rejection dynamics and the transmission stiffnesses. Their effects have been confirmed by simulations, as shown in Fig 2-18, Fig 3-24 and Fig 3-25. However, their influence on DFF is still not clearly indicated. As introducing in details the source of DFF is very important for designing the force equalization strategies, it shall be studied at first.

To start with, the study is engaged based on the linear approach. As indicated in equation 2.42, while both SHA and EMA are position controlled, the force fighting $\gamma$ can be written as:

$$\gamma = \left[ \frac{S_{st} (H_1 + H_2 S_{st}) - S_{et} (G_1 + G_2 S_{et})}{(G_1 + G_2 S_{et}) (H_1 + H_2 S_{et})} \right] M_s s^2 + 2H_1 S_{st} S_{et} - 2G_1 S_{et} S_{et} X_r + \frac{G_1 S_{st} (H_1 + H_2 S_{et}) - H_1 S_{et} (G_1 + G_2 S_{et})}{(G_1 + G_2 S_{et}) (H_1 + H_2 S_{et})} F_L$$

(5.1)

In order to remove the DFF, the items before $X_r$ and $F_L$ should be made as small as possible. In an ideal case, while these items are null, the force fighting will be totally eliminated. It leads to:

$$\left[ \frac{S_{st} (H_1 + H_2 S_{st}) - S_{et} (G_1 + G_2 S_{et})}{(G_1 + G_2 S_{et}) (H_1 + H_2 S_{et})} \right] M_s s^2 + 2H_1 S_{st} S_{et} - 2G_1 S_{et} S_{et} = 0$$
$$G_1 S_{st} (H_1 + H_2 S_{et}) - H_1 S_{et} (G_1 + G_2 S_{et}) = 0$$

(5.2)

giving:
In equation 5.3, the first line indicates that the pursuit dynamics of SHA and EMA are to be made similar; the second line indicates that the rejection dynamics of SHA and EMA should be made according to the values of their transmission stiffness.

In addition, as already mentioned, in order to place the DFE in a very challenging case, it is decided to set the transmission stiffnesses to a very high level of $S_{st}=S_{et}=1 \times 10^9$ N/m. This leads the right hand of second line in equation 5.3 to disappear. Thus, both the pursuit dynamics and the rejection dynamics are to be made similar as far as possible to remove any DFF.

As indicated in Fig 2-18 and Fig 3-24, the DFF in pursuit condition is much graver than that in rejection condition. So, while the achievement of identical pursuit dynamics and identical rejection are in conflict, the priority is given to pursuit dynamics. Accordingly, the first DFE control strategy is proposed making the pursuit dynamics of SHA and EMA identical.

### 5.1 DFE-1 Feed-Forward Compensation for Position Control

The key idea of this control strategy is to force SHA and EMA having same pursuit dynamics. With the position controllers designed for the SHA and EMA in chapter 2, only the actuator position is accurately controlled, the velocity and acceleration are not directly concerned, but these two items have great influence on the pursuit dynamics. For getting a desired pursuit performance, it becomes interesting to better control the velocity and acceleration. So, in this DFE strategy, the velocity and acceleration compensations for position control are introduced. Meanwhile, as the feedback of velocity and acceleration will decrease the actuator rejection stiffness, the feed-forward is employed. The reference velocity and acceleration signals are generated by a trajectory generator. The schematic of this DFE strategy is shown in Fig 5-1.

As far as both SHA and EMA can well pursuit the position, velocity and acceleration reference, they can be considered as having similar pursuit dynamics.
5.1.1 Trajectory Generator

The desired pursuit dynamics is set by the trajectory generator. From the position demand $X_r$, it generates three outputs as the reference position $X_{tr}$, velocity $\dot{X}_{tr}$ and acceleration $\ddot{X}_{tr}$. The common solution to make a trajectory generator is using a second-order filter (2 parameters) or a third-order filter (3 parameters). Until now, it is still not easy to assess the interest of increasing the order of trajectory generator. So, the simple second-order is selected at first.

The schematic of the second-order trajectory generator is shown in Fig 5-2.

$$
\begin{align*}
X_r &\quad \begin{cases}
\frac{1}{\omega_i^2} s^3 + \frac{2 \xi}{\omega_i} s^2 + 1 & \text{Ideal} \\
\frac{1}{\omega_i^2} s^3 + \frac{2 \xi}{\omega_i} s^2 + \frac{1}{s} & \text{Realistic}
\end{cases}
\end{align*}
$$

where:

- $\omega_i$ Reference natural frequency [rad/s]
- $\xi$ Reference damping factor [null]
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The Fig 5-2 left is the ideal schematic. The Fig 5-2 right shows one kind of practical and realistic realization. Two saturations are introduced to limit the velocity and acceleration magnitudes. According to the actuator capability, the velocity limit is set to ±0.16m/s and the acceleration limit is set to ±2.0m/s².

The two parameters $\omega_i$ and $\xi_i$ of trajectory generator are set to meet the pursuit dynamics requirement: -3dB/-45° for 1mm/3Hz sine position demand. Considering the tracking phase lag of closed loop actuators, the trajectory generator is set 10% faster than the required actuator output. Therefore, the typical values are $\omega_i=40\text{rad/s}$ and $\xi_i=0.707$.

This trajectory generator will be used during designing pre-compensators for SHA and EMA.

5.1.2 SHA Pre-compensator Design

As concluded in chapter 2, although the linear model does not provide a good representation of the statics, it represents quite well the dynamics. Thus, the design of pre-compensator is started from linear approach. Once validated on linear model, it is tuned more accurately on virtual test bench.

In SHA, the compensation signal is input as a servovalve current $I_{sv}^*$. According to equation 2.9, the flow delivered by the servovalve is consumed by the functional need (hydrostatic flow due to rod velocity) and the parasitic effects (hydraulic compression and leakage). Therefore, the compensation can be calculated as:

\[
\begin{align*}
M_i \ddot{X}_s &= A_i P_f - F_{ij} - F_L \\
Q_{sv} &= K_{sg} I_{sv} - K_{sc} P_f \\
&= A_i \dot{X}_s + \frac{V_i}{4E_y} \dot{P}_f + K_{wc} P_f
\end{align*}
\]

\[
P_f^* = \frac{M_i^* \ddot{X}_{sr} + F_{ij}^* + F_L^*}{A_i^*}
\]

\[
I_{sv}^* = \frac{1}{K_{sg}^*} \left( A_i^* \dot{X}_{sr} + \frac{V_i^*}{4E_y} \dot{P}_f^* + K_{sc}^* P_f^* \right)
\] (5.4)

where * means the estimated value of effective parameters.

In equation 5.4, the item 1 indicates the functional flow compensation through the reference velocity signal $\dot{X}_{tr}$; the item 2 indicates the parasitic compensation through the load pressure $P_f^*$ which is got from the reference acceleration signal $\ddot{X}_{tr}$.
While achieving the parasitic compensations, the influences of jack friction $F_{jf}$ and external load $F_L$ are neglected. The jack friction $F_{jf}$ is neglected because its magnitude is very small in comparison with the jack output force, as indicated in Fig 3-13. The influence of external force $F_L$ is neglected in order to avoid introducing a positive feedback loop that could decrease the stability of actuator.

According to the above analysis, the SHA pre-compensator can be designed as in Fig 5-3:

![Fig 5-3 Schematic of pre-compensator for SHA](image)

Then, the performance of this compensator is assessed on the linear prototype. The EMA is disconnected to remove its load effect. The position demand is a 1mm step at $t=0.1s$ and the external force is null. The simulation response is displayed in Fig 5-4.

![Fig 5-4 Performance of pre-compensator for SHA on linear model](image)

As shown in Fig 5-4, after introducing the pre-compensator, the SHA pursuit dynamics is close to the desired one. The magnitudes of velocity and acceleration are significantly reduced in comparison with the original controller. This will benefit the force equalization. Moreover, as
indicated in the right graph, the SHA acceleration does not perfectly pursue the reference one because it is a very dynamic demand not consistent with the dynamics of SHA (the servovalve dynamics, hydro-mechanical mode, etc). In addition, the pre-compensator avoids exciting the hydro-mechanical mode and therefore reduces the oscillations in the pursuit mode.

Meanwhile, through comparing the compensation values, the functional flow compensation is found playing a major role in final response. The parasitic compensation only generates a very small current that is lower than 5% of the total compensated servovalve current. So, the modeling error in parasitic channel caused by linearization can be neglected. In addition, while the whole DFE strategy is validated on the virtual test bench, the effect of this parasitic compensation will be evaluated and if possible, it will be eliminated as its uncertainties will significantly reduce the robustness.

Once validated on linear model, the pre-compensator is tuned more accurately on virtual test bench. According to above analysis, the compensation of functional and parasitic flow can still be calculated as in equation 5.4. The only part needs to be improved is the servovalve current/flow gain.

According to equation 2.2, the compensation current can be calculated as:

\[
I_{sv}^* = \frac{Q_{sv}^*}{D_{sv}(s)K_{sv}^* \sqrt{1 - \frac{P_f}{P_s} \text{sgn}(X_{sv})}}
\]  

(5.5)

where \(D_{sv}(s)\) is the servovalve dynamics, as it is much faster in comparison with the position loop, it is considered as a pure proportional part with a gain of 1.

As shown in equation 5.5, in order to calculate the compensation current, the load pressure \(P_f\) is needed, which will cause various issues. This forms a positive pressure feedback, although it can reduce the static error, it will alter the position loop stability. If involved, it can be used for very low frequency domain only. Meanwhile, due to the sgn effect, a strong discontinuity will appear around null opening area, this also makes the stability worse. So, the influence of load pressure is neglected for stability reasons. The substitute solution is using the constant linearized current/flow gain as in Fig 5-3.
So the pre-compensator designed on linear approach is totally kept for the virtual test bench. Simulations are run to check its performance in nonlinear environment. The position demand is similar as in Fig 5-4, but one simulation is under null external force and the other one is under a 10KN external force applying from the beginning.

As displayed in Fig 5-5, the pre-compensator works well on virtual test bench whatever the external load. Similar as on linear model, except for the time delay in acceleration pursuit, the position and velocity well track the reference value. Generally, although the compensator uses estimated value of the real parameter, its performance is satisfied.

5.1.3 EMA Pre-compensator Design

In this section, the pre-compensator for EMA will be designed. Same process as SHA, at first validated on linear model then validated on virtual test bench.

In EMA, the feed-forward compensation signal is the motor torque demand \( T_m' \). Except for the surface load, the motor torque is consumed by two parasitic needs: rotor inertia to accelerate and decelerate and the roller-screw friction. According to this, the force effects acting on the motor shaft can be presented as:

\[
T_m = T_f + J_m \dot{\omega}_c + \frac{l^2}{4\pi^2} M_i \dot{\omega}_c + \frac{l}{2\pi} F_L
\]  

(5.6)

With the cascade position controller designed in chapter 2, the EMA motor torque demand can be written as:
As the motor torque control loop has a natural frequency of 600Hz which is much higher than the position loop, the motor electromagnetic torque $T_m$ is considered equaling to the torque demand $T_d$.

Accordingly, in order to make the position error $X_r-X_e$ to be null, the torque consumption in equation 5.6 and the electrical viscous effect caused by velocity feedback $K_{ec}$ in equation 5.7 should be both compensated. This leads the compensator to be written as:

$$T_d^* = T_f^* + J_m^* \dot{\omega}_e^* + \frac{I_m^*}{4\pi^2} M_i^* \dot{\omega}_i^* + \frac{I^*}{2\pi} F_L^* + \frac{I^*}{2\pi} \omega_e^* K_{ec}$$

$$= \left( T_f^* + \frac{I^*}{2\pi} \omega_e^* K_{ec} \right) + \left( J_m^* + \frac{I_m^*}{4\pi^2} M_i^* \right) \dot{\omega}_e^* + \frac{I^*}{2\pi} F_L^* \tag{5.8}$$

For the same reason as in SHA, the influence of external load $F_L$ is neglected in order to avoid the positive feedback. The surface inertia $M_i$ is also neglected as it generates negligible inertia effect in comparison with that of the rotor inertia $J_m$. Meanwhile, the roller-screw friction is linearized as a pure viscous effect on linear model. So, the compensator can be expressed as:

$$T_d^* = T_f^* + K_{ec} \dot{X}_n + \frac{2\pi}{I^*} J_m^* \dot{X}_n \tag{5.9}$$

And the realization of this pre-compensator is:

After that, the pre-compensator is validated on linear model. The SHA is disconnected. The position demand step of 1mm is applied at 0.1s under null external force.
As shown in Fig 5-7, the pre-compensator works well on setting the pursuit dynamics. The time delay of acceleration is much smaller than the one in SHA. This is because the dynamics of motor torque control loop is very high at 600Hz. The torque demand can be immediately transferred to torque output to meet the dynamics demand.

Then the pre-compensator is tuned in details and validated on virtual test bench. From linear model to nonlinear model, the roller-screw friction changes from pure viscous to the model as indicated in equation 3.3 including Coulomb, Stribeck and viscous. The influence of external force $F_L$ is neglected for the same reason as explained above.

$$T_f = \left( T_{cl} + (T_{sl} - T_{cl}) e^{\frac{P_e}{\mu}} \right) \text{sgn}(\omega_e) + B_e \omega_e$$

(5.10)

With this more realistic compensation, two simulations are run on virtual test bench. The position demand is common as on linear model, but one under null external force and one under 10KN external force applying from the beginning. The responses are shown in Fig 5-8.
It can be found that the performance of the EMA pre-compensator is not very good. The load position is not as smooth as the one of SHA due to the strong nonlinearity of sgn function in roller-screw friction. In addition, many oscillations appear on the velocity and acceleration. This is mainly due to the huge rotary inertia of EMA, the load mass is so small in comparison with it that it becomes very difficult to accurately control the load acceleration while many nonlinearities are involved between the load and the motor rotor.

The frequency of the oscillation on velocity and acceleration is driven by the EMA equivalent stiffness (series combination of EMA transmission stiffness and roller-screw stiffness) and load mass. Due to the lost motion in the roller-screw, the overall equivalent stiffness is lower while within the lost motion domain (under null or small external load, see appendix C). This explains the frequency difference under two conditions.

Generally, after introducing the compensator, the load dynamics is much closer to the desired one than that before introducing compensator.

Following that, the contribution of each part within the compensator is studied in details. As we know, the mechanical friction is changing with the environment and service time. It is very difficult to accurately represent it with realism in all conditions. For improving the robustness of compensator, it is necessary to make clear the contribution of friction force compensation on performance in the presence of modeling uncertainties.

The EMA feed-forward compensation includes three parts: inertia dynamic, electrical viscous effect and roller-screw friction. They are combined to different groups and evaluated through simulations. The results are shown in Fig 5-9.

Through comparison between graphs, it is shown that:

- **Inertia dynamic compensation**: by comparing graphs 2 and 6, this compensation plays a major role on improving the system response speed at the beginning of step pursuit, the responses after t=0.1s.

- **Electrical viscous compensation**: by comparing graphs 1 and 2, this compensation is observed contributing the most part of final performance.
**5.1.4 Validation of DFE-1 Strategy**

After validated on single actuator, the trajectory generator and compensators are introduced into the hybrid actuation system to evaluate their performance on dynamic force equalization.

To start with, the surface displacement pursuit dynamics is simulated to check if it can meet the requirement after including the DFE strategy. The input is a 1mm position demand step at $t=0.1s$ with null external force. The results are shown in Fig 5-10.

It is observed that the load dynamics after compensated is very close to the desired one and meets the requirement. Meanwhile, this validates the choice made for the trajectory generator.

- **Roller-screw friction compensation**: by comparing graphs 4 and 6, it is shown that introducing the friction compensation does not significantly improve the performance.

Now, the friction compensation is found having less influence on load dynamic performance. In final part, while the whole DFE strategy is validated, its effect on force equalization will be assessed. If the performance does not alter a lot in the absence of this compensation, it will be eliminated for improving robustness.
Then, the dynamic force equalization performance is evaluated. A simulation is run where the inputs are a 1mm position demand step at $t=0.1s$, a -10KN external force step at $t=0.5s$ and a +10KN external force step at $t=1s$. The amount of force fighting $y$ is plotted in Fig 5-11.

It is observed that this DFE control strategy works well on reducing the force fighting:

- **In pursuit condition**: the DFF is decreased by 90% from 36KN to 4KN while the SFF is also well removed ($t=0.4s$).
- **In rejection condition**: although the DFF does not decrease a lot, it can still meet the requirement (20% of rated force 50KN). However, the performance of SFE is poor ($t=1.4s$). The SFF remains 30% of the external force input. From this point of view, a SFE needs to be better addressed.
According to the architecture of this DFE strategy, the first SFE strategy (IFFF) proposed in chapter 4 is introduced. As shown in Fig 5-12, the integral signal of force fighting is used to adjust the actuator position feedback for SFE.

As the setting of effective transmission stiffness is changed in comparison with chapter 4, the integral gain $k$ is recalculated. According to the indication of Fig 4-5, the gain $k=1\times10^{-8}[m/\text{Ns}]$ is defined. The saturation limit is still set to $\pm 1\text{mm}$.

After that, a simulation is run to compare the performance of this hybrid force equalization control strategy with the single DFE one. The result is displayed in Fig 5-13 (where the range of $y$ axis force fighting $y$ is reduced by 5 times from Fig 5.11, from 55KN to 10KN).

**Fig 5-12 Combination of DFE-1 and SFE-1**

**Fig 5-13 Performance of the hybrid force equalization control strategy**
As shown in Fig 5-13, the combination of DFE-1 and SFE-1 displays good results that makes
the hybrid actuation system meet the force equalization requirement in any case. Meanwhile,
the load pursuit and rejection are also well met.

In the following parts, for simplifying the statement, this hybrid control strategy is still called
as strategy DFE-1.

**5.1.5 Third-Order Trajectory Generator Study**

Now, the DFE-1 has been validated on virtual test bench. Following that, in order to enhance
the system performance, the trajectory generator and two pre-compensators are improved.
To start with, in this section, the trajectory generator is to be improved.

In above part, a second-order trajectory generator has been proposed and validated. But, it is
known that the SHA is at least a 5\textsuperscript{th}-order system (3 orders for the position dynamics and 2
orders for the servovalve dynamics, see equation 2.12); EMA is at least a 4\textsuperscript{th}-order system (2
orders for position dynamics and 2 orders for motor torque control dynamics, see equation
2.27). This motivates increasing the trajectory generator order to be more realistic and to less
excite the high frequency modes of the actuators.

Therefore, on the basis of the validated DFE-1, a third-order trajectory generator is proposed,
simulated and compared with the second-order one.

The canonical form of a third-order trajectory generator can be written as:

\[
G_r = \frac{1}{(\tau_s + 1) \left( \frac{1}{\omega_i^2} s^2 + \frac{2\zeta_i}{\omega_i} s + 1 \right)}
\]  

(5.11)

where \( \tau_i \) is the time constant of first-order part.

The parameter setting process is assuming the damping factor of second-order as the critical
value \( \zeta_i = 0.707 \) and then defining the filter effect by fixing the \( \omega_i \) and \( \tau_i \). It has been found that
\( \omega_i = 60 \text{ rad/s} \) and \( \tau_i = 1/60 \text{s} \) minimize the speed and acceleration demand while still meeting the
dynamics requirement (10\% faster than the actuator requirement).
The realization of the third-order trajectory generator is shown in Fig 5-14.

\[ X_1 \xrightarrow{1/s} X_2 \xrightarrow{\omega_i^2} X_3 \xrightarrow{1/s} X_4 \xrightarrow{2\xi\omega} \]

**Fig 5-14** Realization of third-order trajectory generator

Then, with this third-order trajectory generator, a simulation is run where the input is a 1mm position demand step applied at t=0.1s with null external force. The simulated responses are displayed in Fig 5-15.

As shown in Fig 5-15, the surface pursuit response under the third-order trajectory generator is much smoother than that of the second-order. The high frequency vibrations on velocity and acceleration are greatly damped. This naturally benefits the dynamic force equalization as illustrated by the right graph. In addition, as the trajectory generator only produces a filter effect (the static gain is constant 1) on the position demand, it has no influence on the system rejection performance (see Fig 5-23).

Generally, the 3\textsuperscript{rd}-order trajectory generator has a better performance, so it will be employed to replace the former 2\textsuperscript{nd}-order one.
5.1.6 Improvement of the Feed-Forward Robustness

After improving the trajectory generator, the robustness of pre-compensators is addressed. At first, the parameters involved in compensators are summarized in Tab 5-1.

<table>
<thead>
<tr>
<th>SHA Pre-Compensator</th>
<th>Functional Flow (major role)</th>
<th>( A_i^* \dot{X}_r )</th>
<th>( A_i ): set by design, constant and known.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Parasitic Effect</td>
<td>( V_i^* \dot{P}_f^* + K_i^* P_f^* )</td>
<td>( V_i ): changes with position, known</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>( E_Y^* ): changes with mean pressure and actuation, difficult to get an accurate value</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>( K_s^* ): including servovalve flow/pressure gain, changes with valve opening (could be compensated by behaviors around null opening)</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>( P_f^* ): estimated by acceleration reference signal, the influence of jack friction and external force is neglected</td>
<td></td>
</tr>
</tbody>
</table>

| EMA Pre-Compensator | Inertia Dynamic (major role for response speed) | \( \frac{2\pi}{l} J_m^* \dot{X}_r \) | \( J_m^* \): set by design, constant and known |
|                     | Electrical Viscous (major role in the whole duration) | \( K_{ec} \dot{X}_r \) | \( K_{ec} \): set by control parameter, known |
| Roller-Screw Friction Compensation (no significant influence) | \( T_f^* \) | \( T_{cl}^* \): Coulomb friction, changes with temperature and external load |
|                     |                                           | \( T_{st}^* \): Stribeck friction, changes with temperature and external load |
|                     |                                           | \( B_f^* \): viscous effect, changes with temperature and external load |
|                     |                                           | \( F_f^* \): not considered in the model |

As indicated in Tab 5-1, many uncertainties are involved in the parasitic compensation of SHA and roller-screw friction compensation of EMA. In order to improve robustness, it is intended to remove these uncertain compensations if the DFE is not significantly altered in the absence of them.

For validating this, the DFE-1 is simulated with and without these uncertain compensations (parasitic part for SHA and friction part for EMA). The input is a 1mm position demand step occurring at \( t=0.5s \) followed by a 10KN external force step applied at \( t=3s \).
As shown in Fig 5-16, after removing these uncertain compensations, the DFE-1 (with third-order trajectory generator) performance is not altered (max 4KN, requirement 10KN) while the high frequency oscillations are removed. This proves the interest of eliminating the feed-forward compensations that involve high uncertainties from DFE-1.

In summary of this section, the DFE-1 strategy is proposed and validated on virtual test bench. It introduces velocity and acceleration compensations to force both SHA and EMA to identical position pursuit dynamics. Meanwhile, the SFE-1 is also applied to be in charge of the static force equalization.

5.2 DFE-2 PID Force Fighting Feedback

The DFE-2 comes from the extension of the SFE-1 by using the PID action instead of a pure I control. The schematic is shown in Fig 5-17.

![Fig 5-17 Schematic of strategy DFE-2](image-url)
Chapter 5 Dynamic Force Equalization Study

The study of this strategy is divided into two steps:

- **First step:** using a linear approach to define the best setting of the SHA and EMA control parameters that minimize the DFF in pursuit condition (major effect).
- **Second step:** applying the control parameters in the virtual prototype and explore the influence of the PID gains to define the best DFF equalizer.

### 5.2.1 Optimization of Position Control Parameters

As mentioned in the beginning of this chapter, identical actuator pursuit dynamics can benefit the DFE. But while designing position controllers for SHA and EMA in chapter 2, this is not highlighted. Although the SHA and EMA can meet all the requirements, their dynamics are not similar. In view of this, optimizing the control parameters to reduce their dynamics difference will naturally benefit the DFE.

The exploration is on the basis of linear approach. The object is looking for a group of values for control parameters $K_{sb}$ of SHA, $K_{eb}$ and $K_{ec}$ of EMA that minimizes the DFF in pursuit case while ensuring that the system pursuit dynamics can meet the requirement. The control parameters are limited within a range, $K_{sb}$: 7~15[A/m] (see Fig 2-6), $K_{eb}$: 20~40[1/s], $K_{ec}$: 1000~3000[Ns] (see Fig 2-12). The exploration results are displayed in Fig 5-18.

\[
\begin{align*}
  K_{sb} &= 7 \text{A/m} \\
  K_{sb} &= 8 \text{A/m} \\
  K_{sb} &= 9 \text{A/m} \\
  K_{sb} &= 10 \text{A/m} \\
  K_{sb} &= 11 \text{A/m} \\
  K_{sb} &= 12 \text{A/m} \\
  K_{eb} &= 1 \text{A/m} \\
  K_{eb} &= 2 \text{A/m} \\
  K_{eb} &= 3 \text{A/m} \\
  K_{eb} &= 4 \text{A/m} \\
  K_{eb} &= 5 \text{A/m} \\
  K_{eb} &= 6 \text{A/m} \\
  K_{eb} &= 7 \text{A/m} \\
  K_{eb} &= 8 \text{A/m} \\
  K_{eb} &= 9 \text{A/m} \\
  K_{eb} &= 1 \text{A/m} \\
  K_{eb} &= 2 \text{A/m} \\
  K_{eb} &= 3 \text{A/m} \\
  K_{eb} &= 4 \text{A/m} \\
  K_{eb} &= 5 \text{A/m} \\
  K_{eb} &= 6 \text{A/m} \\
  K_{eb} &= 7 \text{A/m} \\
  K_{eb} &= 8 \text{A/m} \\
  K_{eb} &= 9 \text{A/m} \\
\end{align*}
\]

**Fig 5-18** Exploration of SHA and EMA control parameters on the influence of DFF
While the dynamics of SHA is higher, the SHA pulls the EMA, the maximum DFF is positive; oppositely, while the dynamics of EMA is higher, the EMA pulls the SHA, the DFF is negative. So, the balance point should be on the corner line between positive and negative DFF.

Accordingly, the “improved control parameters” are defined as: $K_{sb}=8[A/m]$, $K_{se}=30[1/s]$ and $K_{ec}=2000[Ns]$. Then, this group of control parameters is evaluated on virtual test bench, as shown in Fig 5-19. The DFF under pursuit condition is decreased by 75% from 36KN to 9KN. This shows the good effectiveness of setting the control parameters not only with a pursuit view but also with a DFE view.

5.2.2 PID Force Fighting Compensation

Then on the basis of this improved position controller, the PID force fighting compensation is introduced. Within this PID compensator:

- **I action**: is in charge of the static force equalization, the gain is set according to Fig 4-5 and then explored on virtual test bench to define the best value.
- **D action**: is in charge of the dynamic force equalization, set by exploration.
- **P action**: is used to speed the convergence to static force equalization. While only with the I action, the SFE transient duration will be very long under large force fighting value.

With exploration, the typical setting of PID parameters is $K_p=1.0\times10^{-8}[m/N]$, $K_i=5\times10^{-8}[m/Ns]$ and $K_d=2\times10^{-10}[ms/N]$. The saturation of the force equalization signal is limited to ±1mm to avoid the failure propagation between channels.

Simulations are run to assess the DFE-2 performance under 4 conditions: original controller, original controller + DFE-2, improved controller, improved controller + DFE-2. The input is a 1mm position demand step applied at t=0.5s followed by a 10KN external force step at t=3s. The comparison results are shown in Fig 5-19.
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Fig 5-19 Validation of DFE-2 under different condition

As shown in Fig 5-19, after introducing the PID compensator, the DFF under pursuit condition is decreased by 56% from 9KN to 4KN (-90% from that of original controller which is 36KN). The DFF under rejection condition does not change a lot, but remains within the limit as the load has a much lower influence on DFF than the position demand. The SFF is well equaled in any case. Meanwhile, the position pursuit dynamics and rejection are still met.

However, while the position control parameters are not improved, the performance of this DFE strategy is not good. The maximum DFF in pursuit case is about 15KN which is only 58% reduction from the original 36KN. It is higher than the limit (20% of rated force 50KN) and does not meet the requirement. So generally, whatever the control parameters (improved or not), the DFF is typically reduced by 60% when the DFE-2 is introduced.

5.3 DFE-3 One Position/One Force Control

This DFE is an extension of the SFE-2. As already mentioned in chapter 4.3, the two candidate configurations are:

- **EMA force control/SHA position control**: as concluded in chapter 4, due to the huge rotary inertia of EMA, the dynamics of its force control loop is poor. Even while the inertia dynamics torque is compensated as in DFE-1, its performance is still not good. The reason can be explained through equation 2.32. The rotary inertia part takes 98.54% of the total
equivalent mass, this means only 1.5% of the motor output torque works on the surface dynamics (in case the friction is neglected). Thus, in order to accurately control the force driving surface dynamics, the motor torque control accuracy should be improved hundred times to get a satisfied accuracy. In view of the huge and strong nonlinear roller-screw friction, it is a very challenging work. In comparison with the SHA force controlled which has good servovalve dynamics and smaller inertia, this configuration is not attractive.

- SHA force control/EMA position control: the inertia of SHA is very small that almost has no influence on its force control dynamics. The dynamics of servovalve is also high enough to perform the force control (25 times of the position loop dynamics). The EMA is competent for the pursuit dynamics of surface and its rotary inertia. So, this configuration is selected to construct the DFE-3.

It is a common and efficient practice to use PID controllers for SHA force control [53]. After exploration on PID gains, the typical set is defined as $K_{sp}=1.75 \times 10^{-7}[A/N]$, $K_{si}=5 \times 10^{-8}[A/Ns]$, $K_{sd}=2.5 \times 10^{-9}[As/N]$. However, it is also known that the force error under load velocity remains high even when the I action is set at its max value regarding to stability. This is confirmed by simulations on virtual test bench where the DFE performance is very poor. As indicated in Fig 5-21, the DFF in pursuit condition is close to 18KN that is much higher than the requirement limit (10KN).

The solution of this issue is to compensate the SHA functional flow as in DFE-1. The reference velocity is generated by a second-order trajectory generator that is used to represent the EMA position pursuit dynamics. Using the EMA real velocity as the reference value will introduce a positive feedback that worsens the stability. Oppositely, although a good representation of the velocity is required for the speed generator, it does not introduce positive feedback. Moreover, using this speed generator does not decrease the segregation between channels.

Consequently, the DFE-3 strategy is defined as shown in Fig 5-20.
where $K_{vc}$ is the functional flow compensation gain.

Theoretically, the functional flow compensation gain $K_{vc}$ should be set as:

$$K_{vc} = \frac{A}{K_{sg}} = 0.3735 [\text{As/m}] \quad (5.12)$$

But in practice, in order to avoid over compensation due to the nonlinear servovalve flow gain, it is set to 60% of the full value as $K_{vc} = 0.22 [\text{As/m}]$.

Then, a simulation is run to evaluate the performance of DFE-3. The input is a 1mm position demand step occurring at $t=0.5s$ followed by a 10KN external force step applied at $t=1.6s$. The simulated responses are displayed in Fig 5-21.

As shown in Fig 5-21, after introducing the velocity compensation, the DFF under pursuit...
Chapter 5 Dynamic Force Equalization Study

Condition is reduced by 63% from 17.6 KN to 6.5KN (and -82% from without DFE 36KN). The SFF in pursuit case is also well achieved and close to null. Considering the performance under rejection condition, the DFF is smaller than 3KN and the SFF is around 0.5KN which all meet the requirement. In addition, the pursuit dynamics and rejection are not degraded and still meet the requirement (see Fig 5-22 and Fig 5-23).

In summary, according to the simulations on virtual test bench, the DFE-3 is proved being an attractive solution for DFE.

5.4 Comparison of Dynamic Force Equalization Performance

In this section, the different DFE strategies are compared on virtual test bench. Simulations are run to evaluate the performance under both pursuit condition and rejection condition.

The simulation input is a 1mm position demand step (2% of full stroke) applied at t=0.5s for pursuit case followed by a 10KN external force step (20% of rated force 50KN) occurring at t=3s for rejection case. The simulated responses are studied in two parts:

a) Performance under pursuit condition:

![Fig 5-22 Performance under pursuit condition](image-url)
As illustrated in Fig 5-22:

- **SFE**: all the proposed DFE control strategies can meet the requirements only with a very little difference.

- **DFE**: the DFE-2 has the best performance as the minimum DFF and fastest convergence speed; in second position comes the DFE-1 which also has a small DFF but with slower convergence speed; the DFE-3 is the worst one with the largest DFF.

- **Position pursuit dynamics**: the DFE-1 and DFE-3 have a good performance; the pursuit dynamics of DFE-2 is a little poor but still meets the requirement. As in DFE-2, the SHA control gain $K_{sb}$ is significantly reduced after the improvement of control parameters. This worsens the SHA dynamics and alters the position control performance.

**b) Performance under rejection condition**:

As illustrated in Fig 5-23:

- **SFE**: all the proposed DFE control strategies can meet the requirements. The DFE-2 has the best performance as the fastest convergence speed. The second position is the DFE-1 which needs a long time to arrive SFE. The DFE-3 is the worst one because of the presence
of badly damped oscillations at 75Hz that come from the servovalve dynamics.

- **DFE**: all the DFE control strategies can meet the requirements and the DFE-2 has the best performance. As mentioned in chapters 2 and 3, the DFF in rejection condition is mainly fixed by the actuator overall stiffness (including hydraulic stiffness, roller-screw stiffness and transmission stiffness). In the present chapter, in order to place the DFE in a worse case, the transmission stiffness were identically set to a high value of $S_{st} = S_{et} = 1 \times 10^9 \text{N/m}$. Meanwhile, the hydraulic stiffness ($1 \times 10^8 \text{N/m}$) and roller-screw stiffness ($1.5 \times 10^8 \text{N/m}$ at operation point, see Appendix C) are also similar. Therefore, although the DFF dynamics is out of the position control dynamics domain, the amount of force fighting remains below the DFE requirement (10KN).

- **Rejection performance**: the DFE-1 has the best performance. It provides a closed loop stiffness of $6 \times 10^8 \text{N/m}$ at low frequency that is 3 times of the requirement. The DFE-2 and DFE-3 are a little poor only with $3 \times 10^8 \text{N/m}$. In comparison with the actuation system without force equalization, the DFE-1 produces no change in the position response. The DFE-2 and DFE-3 increase by twice the static error under permanent load. In following part to study the robustness of DFE strategies, this becomes a major issue for DFE-2 and DFE-3. In addition, it is mandatory to remark that introducing the pressure or force inner loop in SHA position control reduce the actuator closed loop static stiffness. But this effect can be reduced by filtering the pressure or force signal with a high pass filter [52].

In addition, the oscillations with a frequency of 110Hz are driven by the equivalent stiffness of two channels and the load mass as a spring/mass effect.

However, it should be noticed that the performance of nonlinear system is driven by the input magnitude. With a different input magnitude, the performance could be significantly changed. For this, the performance of the DFE strategies is assessed under different position demands and external load magnitudes. The maximum DFF is measured to present the performance, as shown in Fig 5-24.
Chapter 5 Dynamic Force Equalization Study

It is found that the performances of all the DFE strategies are significantly degraded under large position demand magnitude. This is mainly due to:

- The nonlinearities involved in system: the control strategies are designed on the basis of linear approach and adjusted through nonlinear simulation. The load dependent or speed dependent nonlinearities are not explicitly taken into consideration in the FE strategies.

- The unbalance of actuator capability: as the EMA needs to drive the huge rotary inertia for dynamics, its acceleration capability is much lower (1.4m/s²) in comparison with the SHA (80m/s²). This makes the EMA output torque saturation in large position demand case. This would enlarge the pursuit dynamics difference between two channels.

- The high value of transmission stiffness: as mentioned in the beginning of present chapter, in order to place the DFE in a worst case, the transmission stiffness were set to very high values of $S_{st}=S_{et}=1 \times 10^9$ N/m. A lower stiffness could reduce the amount of force fighting.

In order to verify the above analysis, two groups of simulations are run. In the first group, the EMA motor torque limit are removed; in the second group, the transmission stiffness are set to a more common values of $S_{st}=S_{et}=1 \times 10^8$ N/m (that is still higher than the practical situation $1.7 \times 10^7$ N/m). The simulation responses are shown in Fig 5-25.

The significant beneficiary of removing the EMA power limit is the DFE-2. Because in DFE-1, the acceleration limit is already involved in the trajectory generator design regarding to the actuator capability; in DFE-3, the EMA is the force master, a higher position dynamics of EMA raises the SHA force dynamics requirement.
Fig 5-25 DFF performance under no EMA power limit and low transmission stiffness cases

Decreasing the transmission stiffness improves the performance of DFE-1 and DFE-2, because both the actuators are position controlled in these strategies. Oppositely, in DFE-3, as the SHA is force controlled, the influence of decreasing transmission stiffness is not so obvious.

Now, the DFF in rejection condition is considered from Fig 5-24 right. Opposite to the pursuit condition, the performance of DFE strategies under large external load is still acceptable. The DFF is generally proportional to external load. This is mainly because the actuator structural stiffness is permanent and does not change with the input magnitude.

In summary, it can be observed that the DFE-1 is a more attractive candidate solution for the force equalization issue of hybrid active/active redundant actuation system.

5.5 Robustness Study

As a real technological system, the hybrid actuation system behaviors are highly influenced by the nonlinearities and uncertainties. In the present work, the force equalization strategies are designed on the basis of linear approach, the influences from nonlinearities and uncertainties are not considered at first with a robust control objective. Although several nonlinearities are involved in the virtual test bench, they are only modeled regarding to the particular operating condition and cannot reproduce the realistic behaviors under all conditions, such as the roller-screw friction, where the influence of temperature is not considered. So, even the proposed force equalization strategies are already validated on virtual test bench, their performance in real system is not strictly ensured. For this reason, studying the robustness of the proposed
force equalization strategies becomes a mandatory work for us.

The two common solutions for addressing the robustness issue are: involving the robustness considerations in controller design, such as the \( H_{\infty} \) methods [65]; or assessing the robustness by exploration of behaviors once the controllers are designed. The second solution is selected in the present work. As for the first kind of solution, a reasonably good mathematical model is necessary to place the work successfully. This is very difficult for a technological system with low level loops involving so many nonlinearities. Therefore, all the nonlinear constraints are not totally well-handled by the robust control theory [22, 66]. Opposite to this, the second solution is easy to achieve and has been proved by practice.

For the second solution, the controller has already been designed, so the following work is exploring the behaviors to evaluate the performance of controller. At first, the nonlinearities and uncertainties are listed in Tab 5-2.

<table>
<thead>
<tr>
<th>Nonlinearities</th>
<th>Uncertainties</th>
</tr>
</thead>
<tbody>
<tr>
<td>Servovalve pilot stage, current/opening gain, saturation effect</td>
<td>Manufacturing tolerances</td>
</tr>
<tr>
<td>Servovalve power stage, flow/pressure gain, pressure drop and flow gain at null opening</td>
<td>Operation settings</td>
</tr>
<tr>
<td>Jack chamber volume, change with position</td>
<td>Supplier dispersion</td>
</tr>
<tr>
<td>Jack friction, Coulomb, Stribeck, external load dependence effect</td>
<td>Environment, such as the temperature, humidity, electromagnetic interference</td>
</tr>
<tr>
<td>Fluid bulk modulus, change with pressure</td>
<td>Power supply, hydraulic supply pressure, electrical supply voltage</td>
</tr>
<tr>
<td>Fluid viscosity, change with temperature</td>
<td>Modeling errors, almost all the parameters are involved, such as the motor rotary inertia, motor torque control dynamics, servovalve dynamics, frictions, sensor dynamics, etc</td>
</tr>
<tr>
<td>EMA roller-screw friction, Coulomb, Stribeck, external load dependence effect</td>
<td>…</td>
</tr>
<tr>
<td>Compliance involving backlash</td>
<td>…</td>
</tr>
<tr>
<td>Electrical signaling, sampling, quantization, saturation</td>
<td>…</td>
</tr>
<tr>
<td>...</td>
<td>…</td>
</tr>
</tbody>
</table>

After summarizing the nonlinearities and uncertainties in system, the coming problem is how to perform the robustness study. For now, it is not feasible to explore it on the real test bench. A substitute solution is performing the study on the basis of the virtual test bench through varying the parameters to simulate the uncertainties.
For this kind of model-based robustness study, the common solution is using the Monte-Carlo method, which is the most straightforward sampling method to identify the sensitivity of the design to small perturbations of parameters. A number of simulations will be performed with randomly selected parameter combinations for the given distribution and then the simulation results are to be analyzed to assess the design robustness [52, 67], as illustrated in Fig 5-26.

Accordingly, the first step of Monte-Carlo method is defining the distribution of considered parameters. The common selected distribution types have the normal, lognormal, exponential, uniform, triangle, gamma, beta, Rayleigh etc [68]. Among these types, the normal distribution is the most common choice for describing the uncertainties in the natural sciences [69]. For this reason, it is selected for our robustness study.

As illustrated in Fig 5-27, 4 parameters (the nominal value, standard deviation, low and high limits) are needed to define a normal distribution for each considered uncertainty. According to the summaries in Tab 5-2, 17 uncertain parameters are selected for the robustness study involving most of the uncertainties, as summarized in Tab 5-3.
In order to assess the performance, the indicators are to be defined as the output of the Monte Carlo method. They shall be collected on the basis of the simulation results having identical position demand and external load, as shown in Fig 5-28, the input is a 1mm position demand step occurring at t=0.5s and followed by a 10KN external force step applied at t=1.5s.

<table>
<thead>
<tr>
<th>Uncertain Parameters</th>
<th>Description</th>
<th>Nominal Value $\mu$</th>
<th>Standard Deviation $\sigma$</th>
<th>Low Limit</th>
<th>High Limit</th>
</tr>
</thead>
<tbody>
<tr>
<td>RSF_Coulomb</td>
<td>EMA Coulomb friction $T_c$ (Nm)</td>
<td>3.62</td>
<td>0.5</td>
<td></td>
<td></td>
</tr>
<tr>
<td>RSF_Stibbeck</td>
<td>EMA Stibbeck friction $T_s$ (Nm)</td>
<td>-2.25</td>
<td>0.5</td>
<td></td>
<td></td>
</tr>
<tr>
<td>RSF_C</td>
<td>EMA friction quadratic coefficient $c$ (m)</td>
<td>$1.1\times10^{-4}$</td>
<td>$2\times10^{-5}$</td>
<td></td>
<td></td>
</tr>
<tr>
<td>RSF_D</td>
<td>EMA friction quadratic coefficient $d$ (m)</td>
<td>$-6.2\times10^{-5}$</td>
<td>$1\times10^{-5}$</td>
<td></td>
<td></td>
</tr>
<tr>
<td>EMA_Transmission_Stiffness</td>
<td>EMA transmission stiffness $S_t$ (N/m)</td>
<td>$1.0\times10^{9}$</td>
<td>$5.0\times10^{8}$</td>
<td>$-9.2\times10^{8}$</td>
<td>1</td>
</tr>
<tr>
<td>EMA_Backlash_Coe</td>
<td>EMA backlash coefficient $1/X_{cr}$ (1/m)</td>
<td>$6\times10^{4}$</td>
<td>$3\times10^{5}$</td>
<td>$-5.5\times10^{4}$</td>
<td>1</td>
</tr>
<tr>
<td>SV_Amplifier_Gain</td>
<td>SHA servovalve amplifier gain (mA/V)</td>
<td>5</td>
<td>0.1</td>
<td></td>
<td></td>
</tr>
<tr>
<td>SV_Dynamic_Dm</td>
<td>SHA servovalve dynamic damping (null)</td>
<td>1.28</td>
<td>0.5</td>
<td>-0.9</td>
<td>0.8</td>
</tr>
<tr>
<td>SV_Corner_Radius</td>
<td>SHA servovalve corner radius $R_c$ (mm)</td>
<td>$7.9\times10^{-3}$</td>
<td>$5\times10^{-4}$</td>
<td></td>
<td></td>
</tr>
<tr>
<td>SV_Clearance</td>
<td>SHA servovalve clearance $D_c$ (mm)</td>
<td>$2.4\times10^{-3}$</td>
<td>$2\times10^{-4}$</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Jack_Leakage</td>
<td>SHA jack leakage (L/min/bar)</td>
<td>$4.4\times10^{-3}$</td>
<td>$5\times10^{-4}$</td>
<td>$3.5\times10^{-3}$</td>
<td>$5.3\times10^{-3}$</td>
</tr>
<tr>
<td>SHA_Transmission_Stiffness</td>
<td>SHA transmission stiffness $S_t$ (N/m)</td>
<td>$1.0\times10^{9}$</td>
<td>$5\times10^{8}$</td>
<td>$-9.2\times10^{8}$</td>
<td>1</td>
</tr>
<tr>
<td>SHA_Backlash_Coe</td>
<td>SHA backlash coefficient $1/X_{cr}$ (1/m)</td>
<td>$6\times10^{4}$</td>
<td>$3\times10^{5}$</td>
<td>$-5.5\times10^{4}$</td>
<td>1</td>
</tr>
<tr>
<td>Jack_Friction_Gain</td>
<td>SHA jack friction gain $F_{sg}$ (null)</td>
<td>1</td>
<td>0.2</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Fluid_Viscosity</td>
<td>Fluid viscosity (cP) (-30°C~70°C, [70])</td>
<td>27</td>
<td>150</td>
<td>-22</td>
<td>360</td>
</tr>
</tbody>
</table>

![Fig 5-28 Performance indicators for robustness analysis](image-url)
We propose to define 14 performance indicators that will be representative of the stability, accuracy, dynamics, SFF, DFF in both pursuit and rejection conditions. These indicators are summarized in Tab 5-4, where the index #X refers to the Fig 5-28.

### Tab 5-4 Summary of performance indicators

<table>
<thead>
<tr>
<th>Num</th>
<th>Indicators</th>
<th>Description</th>
<th>Requirement Low limit</th>
<th>Requirement High Limit</th>
</tr>
</thead>
<tbody>
<tr>
<td>#1</td>
<td>DFF_PD_PP</td>
<td>DFF under position demand, peak-to-peak (N)</td>
<td>-1×10⁴</td>
<td>1×10⁴</td>
</tr>
<tr>
<td>#2</td>
<td>DFF_PD_Max</td>
<td>DFF under position demand, positive max (N)</td>
<td>0</td>
<td>1×10⁴</td>
</tr>
<tr>
<td>#3</td>
<td>DFF_PD_Min</td>
<td>DFF under position demand, negative max (N)</td>
<td>-1×10⁴</td>
<td>0</td>
</tr>
<tr>
<td>#4</td>
<td>DFF_EF_PP</td>
<td>DFF under external force input, peak-to-peak (N)</td>
<td>-1×10⁴</td>
<td>1×10⁴</td>
</tr>
<tr>
<td>#5</td>
<td>DFF_EF_Max</td>
<td>DFF under external force input, positive max (N)</td>
<td>0</td>
<td>1×10⁴</td>
</tr>
<tr>
<td>#6</td>
<td>DFF_EF_Min</td>
<td>DFF under external force input, negative max (N)</td>
<td>-1×10⁴</td>
<td>0</td>
</tr>
<tr>
<td>#7</td>
<td>SFF_PD</td>
<td>SFF under position demand (N)</td>
<td>-2×10³</td>
<td>2×10³</td>
</tr>
<tr>
<td>#8</td>
<td>SFF_EF</td>
<td>SFF under external force input (N)</td>
<td>-2×10³</td>
<td>2×10³</td>
</tr>
<tr>
<td>#9</td>
<td>Load_Overshot</td>
<td>Surface displacement overshoot (m)</td>
<td>9.5×10⁻⁵</td>
<td>1.2×10⁻³</td>
</tr>
<tr>
<td>#10</td>
<td>Load_RespTime</td>
<td>Load displacement response time (s)</td>
<td>0.59</td>
<td>0.65</td>
</tr>
<tr>
<td>#11</td>
<td>Dis_PD_Sta</td>
<td>Static error under pursuit condition (m)</td>
<td>-5×10⁻⁵</td>
<td>5×10⁻⁵</td>
</tr>
<tr>
<td>#12</td>
<td>Dis_EF_Sta</td>
<td>Static error under rejection condition (m)</td>
<td>-5×10⁻⁵</td>
<td>5×10⁻⁵</td>
</tr>
<tr>
<td>#13</td>
<td>Vel_PD_Sta</td>
<td>Static velocity under pursuit condition (m/s)</td>
<td>-5×10⁻⁶</td>
<td>5×10⁻⁶</td>
</tr>
<tr>
<td>#14</td>
<td>Vel_EF_Sta</td>
<td>Static velocity under rejection condition (m/s)</td>
<td>-1×10⁻⁵</td>
<td>1×10⁻⁵</td>
</tr>
</tbody>
</table>

On the basis of the above defined uncertain parameters and performance indicators, while a number of simulations are run, some post-process methods are used to analyze the simulated results. For such a complex system involving so many uncertainties, the histogram plot which gives a general view on the selected performance is an interesting solution, as displayed in Fig 5-29 left. And for indicating the influence of one parameter variation on one performance, the scatter plot can give a direct view, as shown in Fig 5-29 right.

![Fig 5-29 Post-process method used for robustness analysis](image-url)
Chapter 5 Dynamic Force Equalization Study

While everything is prepared, the robustness study will be performed through co-simulation of LMS_AMESim and Noesis_Optimus [51]. The Optimus is a simulation environment aiming at system optimization and robustness analysis. It has an interface with AMESim to make easy the co-simulation. The general configuration is shown in Fig 5-30.

![Configuration of co-simulation for robustness analysis](image)

**Fig 5-30** Configuration of co-simulation for robustness analysis

The co-simulation is performed in four steps:

- First step: the uncertain parameters and performance indicators are defined in AMESim on the basis of the virtual test bench.
- Second step: through the interface between two environments, the definition of uncertain parameters and indicators are sent to Optimus, in which the distribution of the uncertain parameters and the requirement range of indicators are generated.
- Third step: Optimus and AMESim form a simulation loop, in which Optimus generates the parameters’ values, drives AMESim to run simulations and then collects and saves the values of performance indicators after each simulation.
- Fourth step: a number of simulations are run and the results are studied in Optimus.

The robustness study is separately performed for each force equalization strategy. The results are shown and analyzed in the following parts.
5.5.1 Robustness Study of DFE-1

The robustness of DFE-1 has been evaluated through 2000 simulations that are generated by the Monte-Carlo method (spend 48 hours, a balance number between the time consuming and statistics accuracy) and all the uncertain parameters defined in Tab 5-3 are involved. The histogram plot of each performance indicator is shown in Fig 5-31. The #1 and #4 indicators peak-to-peak DFF are not plotted as their information is included in #2, #3, #5 and #6.

As shown in Fig 5-31, the major loss of robustness of DFE-1 is found in the second plot that comes from the DFF under pursuit condition. Totally 25.7% of simulations cannot meet the requirement. However, the other performances, like the DFF under rejection condition, SFF, stability, load pursuit dynamics and rejection are met.

Then, the origins of DFE performance loss under pursuit condition are studied in Fig 5-32 by the scatter plots. (12 uncertain parameters that have obvious influence are studied).
As shown in these figures, the key uncertainty comes from the hydraulic fluid viscosity. In DFE-1, the SHA and EMA are forced to have similar pursuit dynamics. In low temperature condition, the SHA dynamics becomes poor because the flow gain of the servovalve power stage reduces strongly under high fluid viscosity. While the SHA dynamics is lower than that of the EMA, the EMA pulls the SHA in pursuit condition. The larger their dynamics difference, the worse the DFF performance. One solution for this could be to use a scheduled gain on the servovalve in the feed-forward control [53].

The parameter values that do not permit to satisfy the DFE requirement in pursuit case when a parameter limit appears clearly are emphasized by adding a dashed box.

5.5.2 Robustness Study of DFE-2

The same process is applied to DFE-2. The major loss of robustness is studied in Fig 5-33.

As shown in Fig 5-33, for DFE-2, the performance is lost in both pursuit and rejection cases:

- **Pursuit dynamics (#10 load response time):** totally 83.48% simulations cannot meet the requirement and lead to a very bad performance. This is because in DFE-2, the SHA position control gain $K_{sb}$ is reduced for force equalization (see section 5.2.1), that greatly alters the load pursuit dynamics.

- **Rejection (#12 load static error under external load):** 65.83% simulations cannot meet
the requirement. However, the performance is not so bad. Even in the worst case, the performance indicator #12 is only twice greater than the requirement. The poor rejection performance is mainly due to the low position control gain and force fighting feedback.

- **DFE under pursuit condition** (#3): 8.39% simulations do not meet the requirement. However, for this indicator, the DFE-2 is better than the DFE-1. Even in the worst case, the DFF is only 40% greater than the accepted value.

- **Stability** (#13): 31.65% simulations are over the limit but the performance is acceptable.

It can be found that the pursuit dynamics and rejection are most sensitive to the parameters variation. So their origins are studied in Fig 5-34 and Fig 5-35 individually.
As displayed by the boxes added on Fig 5-34 and Fig 5-35, the system performance is bad for low transmission stiffness, low servovalve dynamics damping and high fluid viscosity. The key parameter for pursuit dynamics and rejection is the hydraulic fluid viscosity. While the fluid viscosity is high at low temperature, the servovalve flow/opening gain reduces significantly. This worsens the load pursuit dynamics. Meanwhile, this gain plays an important role on SHA closed loop stiffness, as indicated in equation 2.19.
5.5.3 Robustness Study of DFE-3

A same process is also applied for DFE-3. The performance indicators are analyzed in Fig 5-36.

As shown in Fig 5-36, for DFE-3, the performance is mainly lost for the load rejection (#12). Totally 71.79% simulations cannot meet the selected requirement. While the SHA is force controlled, the load rejection is mainly driven by the equivalent stiffness of EMA channel. As the EMA controller provides a very little margin in performance rejection, the presence of parasitic transmission compliance makes the general rejection stiffness a little poor. This is confirmed by analysis in Fig 5-37, where the origins of bad rejection performance are studied.

In addition, the unmet simulations of indicator #3 DFF in pursuit case is mainly due to the high fluid viscosity (at low temperature), which reduces the servovalve flow gain strongly and worsens the SHA force control dynamics.
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5.5.4 Summary of Robustness Study

The former results enable the robustness of each force equalization strategy to be compared as summarized in Tab 5-5. The % value represents the simulations that do not meet the actual performance requirement. The color of the cell facilitates the comparison of the strategies (the blue prefers not sensitive to parameter variation, red is extremely sensitive). The general indicator shows the all unsatisfied simulations whatever which performance is not met.

**Tab 5-5** Summary of system performance under multi uncertainties

<table>
<thead>
<tr>
<th>System Performance Indicator</th>
<th>DFE-1</th>
<th>DFE-2</th>
<th>DFE-3</th>
</tr>
</thead>
<tbody>
<tr>
<td>#1 Peak-peak DFF in pursuit case</td>
<td>34.5%</td>
<td>45.46%</td>
<td>17.93%</td>
</tr>
<tr>
<td>#2 Positive max DFF in pursuit case</td>
<td>0.55%</td>
<td>0.06%</td>
<td>0.43%</td>
</tr>
<tr>
<td>#3 Negative max DFF in pursuit case</td>
<td>25.7%</td>
<td>8.39%</td>
<td>6.35%</td>
</tr>
<tr>
<td>#4 Peak-peak DFF in rejection case</td>
<td>1.6%</td>
<td>5.11%</td>
<td>8.22%</td>
</tr>
<tr>
<td>#5 Positive max DFF in rejection case</td>
<td>0.2%</td>
<td>0.88%</td>
<td>3.92%</td>
</tr>
<tr>
<td>#6 Negative max DFF in rejection case</td>
<td>1.25%</td>
<td>0.25%</td>
<td>4.3%</td>
</tr>
<tr>
<td>#7 SFF in pursuit case</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>#8 SFF in rejection case</td>
<td>0</td>
<td>0</td>
<td>2.8%</td>
</tr>
<tr>
<td>#9 Load overshoot in pursuit case</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>#10 Load response time in pursuit case</td>
<td>7.35%</td>
<td>83.48%</td>
<td>6.85%</td>
</tr>
<tr>
<td>#11 Load static error in pursuit case</td>
<td>0</td>
<td>0</td>
<td>0.06%</td>
</tr>
<tr>
<td>#12 Load static error in rejection case</td>
<td>3.35%</td>
<td>65.83%</td>
<td>71.79%</td>
</tr>
<tr>
<td>#13 Load static velocity in pursuit case</td>
<td>1.85%</td>
<td>31.65%</td>
<td>4.36%</td>
</tr>
<tr>
<td>#14 Load static velocity in rejection case</td>
<td>0</td>
<td>0</td>
<td>5.85%</td>
</tr>
<tr>
<td><strong>General</strong></td>
<td><strong>40.35%</strong></td>
<td><strong>87.83%</strong></td>
<td><strong>73.16%</strong></td>
</tr>
</tbody>
</table>
Generally speaking, it can be observed that DFE-1 has the best robustness. The least robust performance is the DFF under pursuit condition. The DFE-2 has the worst robustness which is due to the poor pursuit dynamics and rejection. The DFE-3 is the middle one. Its rejection performance is over the maximum accepted value due to the low closed loop stiffness of the EMA (could be improved).

5.6 Analysis of Segregation and Complexity

As the dual channel design comes from a reliability construct, ensuring segregation between each channel is of prime importance. This concerns all the channels from the controller to the connection to the load. Unfortunately, the force equalization strategy may require introducing cross links between channels. One solution to reduce this effect consists in limiting the authority of the force equalization controller, e.g. through a saturation function.

The segregation between the SHA and EMA channels introduced by DFE-1, DFE-2 and DFE-3 is summarized by Fig 5-37.

In strategies DFE-1 and DFE-2, force equalization generates a secondary position demands to each channel. As already mentioned, the loss of segregation is reduced by limiting these signals to a few percent (4%) of the actuator stroke. The DFE-3 only introduces a directional cross link as a force demand to the SHA channel generated from the EMA channel output force.

In addition, the complexity of the force equalization strategy is also an important criterion to evaluate their possibility of realization. It involves the complexity during design and setting (non recurring cost, NRC) and also the complexity coming from hardware needs (recurring
cost, RC). The design process and hardware needs of each strategy are summarized in Tab 5-6.

Tab 5-6 Summary of design process and hardware needs

<table>
<thead>
<tr>
<th>DFE-1</th>
<th>DFE-2</th>
<th>DFE-3</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>NRC Complexity</strong></td>
<td><strong>NRC Complexity</strong></td>
<td><strong>NRC Complexity</strong></td>
</tr>
<tr>
<td>4 steps and 6 parameters</td>
<td>2 steps and 6 parameters</td>
<td>2 steps and 6 parameters</td>
</tr>
<tr>
<td>➢ 3rd-order trajectory generator: 3 (t_e, \omega_i, \text{and } \zeta_i)</td>
<td>➢ Position control parameters improvement: 3 (K_{db} \text{ for SHA, } K_{db} \text{ and } K_{dc} \text{ for EMA})</td>
<td>➢ SHA PID force controller: 3 (K_{sp}, K_{si} \text{ and } K_{sd})</td>
</tr>
<tr>
<td>➢ SHA feed-forward: 1 (A_f^i / K_{dp})</td>
<td>➢ PID force fighting feedback: 3 (K_p, K_i \text{ and } K_d)</td>
<td>➢ Velocity feed-forward gain and 2nd-order speed generator: 3 (K_{vc}, \omega_i \text{ and } \zeta_i)</td>
</tr>
<tr>
<td>➢ EMA feed-forward: 1 (f_m^i)</td>
<td>➢ SFE integral gain: 1 (k)</td>
<td>➢ SHA: 1 force sensor (or 2 pressure sensors)</td>
</tr>
<tr>
<td>➢ EMA: 1 force sensor</td>
<td>➢ EMA: 1 force sensor</td>
<td>➢ EMA: 1 force sensor</td>
</tr>
</tbody>
</table>

Generally speaking, although the DFE-1 is the most complex one, its parameters are easily set (according to system model); the DFE-2 has simpler process but the PID parameters of the force fighting feedback need to be defined through parameter exploration (however, if the similar pursuit dynamics is involved from the position controller design, the first step could be removed); the DFE-3 has the simplest configuration only with a little difficulty on setting the PID force controller for SHA, the velocity feed-forward is set according to the model.

5.7 Conclusion of Chapter 5

In present chapter, three DFE control strategies were proposed and assessed on virtual test bench:

- **DFE-1**: both SHA and EMA were position controlled and forced to similar pursuit dynamics through velocity and acceleration feed-forwards. The functional flow compensation for the SHA, the velocity feed-forward and the rotary inertia dynamics compensation for the EMA played a major role in the final performance.
- **DFE-2**: both SHA and EMA were position controlled, the force fighting feedback signal was used to compensate the actuator’s position feedback signal through a PID controller.
- **DFE-3**: the EMA was position controlled, the SHA was force controlled and demanded to track the EMA output force. A PID force controller was combined with a velocity feed-
forward to get high dynamics and to reject the influence of the load velocity feedback.

Then, the load pursuit dynamics, rejection and force equalization performance of these DFE control strategies were compared. The DFE-1 had good force equalization and the best pursuit dynamics and rejection; the DFE-2 had best force equalization but with the worst load pursuit dynamics and rejection; the DFE-3 had a poor force equalization and rejection but with a good pursuit dynamics.

Following that, the robustness of these DFE control strategies was studied by the parameters exploration according to Monte-Carlo method. The DFE-1 had the best robustness except for the DFF under pursuit condition; the DFE-2 had the worst robustness mainly due to the poor pursuit dynamics and rejection; the DFE-3 had a bad robustness as its rejection performance is unmet. The hydraulic fluid viscosity was found playing a major role on system performance because it greatly influenced the SHA dynamics for any strategy.

The segregation and complexity of these DFE control strategies have been studied to provide additional criteria for the general comparison.

Finally, the general performance of these DFE strategies is summarized in Tab 5-7.

**Tab 5-7 General comparison of the DFE strategies**

<table>
<thead>
<tr>
<th>Items</th>
<th>Without FE</th>
<th>DFE-1</th>
<th>DFE-2</th>
<th>DFE-3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Load Pursuit Dynamics</td>
<td>++</td>
<td>++</td>
<td>0</td>
<td>++</td>
</tr>
<tr>
<td>Load Rejection</td>
<td>++</td>
<td>++</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Dynamic Force Equalization</td>
<td>−−</td>
<td>+</td>
<td>++</td>
<td>0</td>
</tr>
<tr>
<td>Static Force Equalization</td>
<td>−−</td>
<td>+</td>
<td>++</td>
<td>+</td>
</tr>
<tr>
<td>Robustness</td>
<td>+</td>
<td>+</td>
<td>−</td>
<td>0</td>
</tr>
<tr>
<td>Segregation</td>
<td>++</td>
<td>+</td>
<td>0</td>
<td>−</td>
</tr>
<tr>
<td>Complexity</td>
<td>++</td>
<td>0</td>
<td>+</td>
<td>+</td>
</tr>
<tr>
<td>General</td>
<td>−</td>
<td>++</td>
<td>+</td>
<td>0</td>
</tr>
</tbody>
</table>

- **Conclusion 1:** For safety critical applications such as the primary flight control actuation or landing gear systems in aerospace, the DFE-1 looks the most attractive solution as it combines good segregation and good robustness.
• **Conclusion 2:** For applications more highlighting force equalization and less regarding segregation, the DFE-2 is a good candidate choice. The DFE-3 could be used to investigate the hybrid control of actuation system as a whole with MIMO view for the controller design.
Conclusion

As mentioned by chapter 1, the motivation of aviation is calling for safer, cheaper and greener aircraft. For this objective, more and more electrical power is involved on aircraft to replace the other types of power networks and power transformers. One important contributor to the more electrical aircraft (MEA) comes from the flight control actuation system, where the power-by-wire actuators (PbW) are progressively introduced as a substitute to the conventional hydraulically powered actuators. The electro-hydrostatic actuator (EHA) and electro-mechanical actuator (EMA) are the two main types of PbW actuators. But due to the lack of maturity, the EHAs and EMAs are still operated as the backup of the servo-hydraulic actuators (SHAs) in primary flight control system. On the way towards totally replacing the hydraulic actuators with the electrical ones, one necessary transition is that the hydraulically powered actuators and electrically powered actuators can work together, e.g. on active/active mode to drive a common flight control surface.

For the hybrid active/active redundant actuation system, a major issue is the force fighting between different channels. Due to the dissimilar technologies and the manufacturing/setting tolerances, the actuators do not try to drive the surface at an exactly identical position. As a convergence, they share the load unequally and force one against the other (force fighting). Considering the grave effect of force fighting on material fatigue and energy consumption, this issue must be addressed with particular attention.

In the present thesis, the combination of SHA and EMA was selected, as using EHAs is often considered as a through point on the way to EMAs. A hybrid actuation test bench of this type was used to help performing the research work.

Addressing actuation for commercial aerospace places the force fighting problem in a very specific context. The force equalization strategies must comply with the severe reliability requirement. This induces four major constraints that have driven the research work:
Conclusion

- The force equalization strategy shall not degrade the segregation between channels in unacceptable amount in order to not propagate a failure from one to the other path.
- The force equalization strategy shall use a reduced number of mature sensors.
- The controller shall be certified easily.
- The real time workload added by the force equalization strategy shall remain low.

According to the above mentioned constraints:

In the first step (chapter 2), two position controllers were designed for the SHA and EMA individually on the basis of the linear approach. A proportional plus notch filter controller was selected for the SHA and a cascade controller (velocity and position loops) for EMA. They were experimentally validated on the real test bench. Then, the force fighting was primarily pointed out on linear model which was built for designing position controllers at this step. It was shown that:

- The dynamic force fighting under pilot demand is much severer than that under external load.

Following that, in the second step (chapter 3), a virtual prototype of the hybrid actuation system was developed in the AMESim simulation environment. This was due to the limitation of the real test bench that presented too much transmission backlash and was not designed to apply any external load. The virtual test bench also provided a fast way to assess and compare force equalization strategies without risk. However, it was required that the virtual test bench was sufficiently accurate to reproduce the phenomenon that driven the final performance. On this attempt, the key nonlinearities were modeled with care, such as the servovalve flow gain, pressure gain, jack friction, roller-screw friction, backlash, structural compliance and electric signaling. The virtual test bench was validated by comparing simulation responses with the experimental results and proved giving a very good representation of the realistic behaviors. Then, on the basis of this virtual test bench, the force fighting was further studied. It was shown that:

- The static force fighting is mainly driven by the position offset of each actuation channel, actuator closed loop compliance and the rod/load transmission compliance.
The dynamic force fighting is mainly driven by the actuator pursuit/rejection dynamics and the equivalent overall stiffness of each channel.

The roller-screw friction of the EMA plays an important role on the EMA performance.

On the basis of the above built virtual test bench, in the third step (chapter 4), the static force fighting (SFF) issue was addressed to focus on static force equalization (SFE) strategies. The sources of static force fighting were studied analytically in detail. As an outcome, the SFF was found to be significantly reduced by adding an adequate position offset on the actuator position feedback. According to this result, the first SFE strategy (SFE-1) was proposed in which the integral force difference between channels was used to determine the position offset. Following this, another SFE strategy was studied, where one actuator was force controlled and behaved as the force slave of the other position controlled actuator (force master). Two candidate configurations were analyzed where either the SHA (SFE-2.1) or EMA (SFE-2.2) was force controlled. The second one shown bad performance due to the huge rotary inertia and roller-screw friction. Oppositely, the first one gave satisfactory results meeting the requirements. In the end, the proposed SFE strategies were experimentally validated on real test bench.

Once the static force equalization was addressed, the fourth step (chapter 5) was dedicated to the dynamic force equalization (DFE). In order to place the DFE problem in a worst case, the actuator’s transmission stiffness were set to a value of 1×10⁹N/m that is very constraining for the force equalization. Through analysis, it was shown that forcing both actuators to similar dynamics naturally reduces the dynamic force fighting (DFF). Following this idea, three DFE control strategies were proposed. In DFE-1, the velocity and acceleration feed-forwards were introduced (the velocity and acceleration reference signals were generated by a third-order trajectory generator) to force the SHA and EMA having similar pursuit dynamics. The DFE-2 extended the principle of SFE-1 by replacing the I controller with a PID one. The DFE-3 was an extension of SFE-2.1, where the force control of SHA was improved from the pure P action to the PID and the velocity compensation was involved to reduce the dynamic error. Each DFE strategy was assessed in details for the whole operation domain on virtual test bench. The robustness of these DFE control strategies was studied through evaluating the sensitivity of
performance indicators to parameters variation. It was shown that:

- The temperature plays a very important role on force equalization performance as it has strong influence on the hydraulic fluid viscosity.

The DFE-1 was observed having the best robustness. In the end, the different properties of the proposed force equalization strategies were summarized to enable a concluding globally.

According to the research in thesis, in order to achieve a good force equalization performance, the followings recommendations can be:

Reduce the amount of force fighting at mechanical design level:

- Making the overall stiffness of each channel similar: the equivalent overall stiffness of SHA channel is the series combination of hydraulic stiffness and transmission stiffness; that of EMA channel is the series combination of roller-screw stiffness and transmission stiffness. Equaling as far as possible the equivalent overall stiffness greatly benefits the DFE under rejection condition.

- Making the power capability of each channel similar: while the actuation capabilities for inertia acceleration and deceleration are of the same level, the DFE performance under different position demand magnitude can be improved.

Reduce the amount of force fighting at position control design level:

- Making actuators’ pursuit dynamics similar: forcing the two channels to similar pursuit dynamics from position controller design will naturally improve the DFE performance.

- Making the static rejection stiffness similar: as a function of the transmission stiffness and the actuator closed loop static stiffness, the SFF under rejection condition is significantly reduced if the similar equivalent general rejection stiffness of each channel is achieved.

Reduce the amount of force fighting at force equalization control strategy design level:

- Introducing SFE control strategy: the SFE-1 that uses integral force fighting to compensate position control is an effective candidate solution for SFE.

- Introducing DFE control strategy: the DFE-1 that introduces the velocity and acceleration
feed-forwards to compensate the position control in such a way to force the SHA and EMA to similar pursuit dynamics is a good candidate solution for DFE. It has a globally satisfied segregation and robustness that are required to achieve.

Regarding the remaining issues in present research, the further investigation can be listed as follows:

- Now the proposed force equalization control strategies cannot be completely validated on the real test bench. For this issue, a quadruple redundant actuation test bench involving 2 SHAs and 2 EMAs are going to be built in ICA, Toulouse. It could be used to validate the proposed control strategies in a more realistic way, where one SHA (or one EMA) could be force controlled to load the SHA/EMA combination redundant actuation system.
- It was shown on virtual test bench that the dynamic force equalization performance was unsatisfactory under large position demand magnitude. For addressing this problem, the nonlinearities should be involved while designing the controllers by using the theory of nonlinear system control and robust control.
- According to the robustness study performed on virtual test bench, the force equalization performance was found unsatisfied under the low temperature condition. For addressing this, the temperature influence on the actuator performance (hydraulic fluid viscosity and mechanical friction) should be considered in controller design.
- Our present research is on the TRL4 level, theory validation in a laboratory environment. The ongoing work should progressively move to a higher level. On the way to final level TRL9, where the force equalization strategy is to be implemented in commercial aircraft through successfully mission operation, there is still a lot of work to do.
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[60] Moog Servovalve


List of Publications


Appendix

Appendix A Electric Signals Involved on Test Bench

This appendix introduces the main signals involved on test bench as all the control task and experimental measurement are on the basis of them.

The main signals involved on test bench are summarized in Tab A-1.

<table>
<thead>
<tr>
<th>Description</th>
<th>Sensor Model</th>
<th>Electrical Range</th>
<th>Physical Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>SHA Rod Extension</td>
<td>LVDT-SX12W100</td>
<td>-10V~+10V</td>
<td>-50mm~50mm</td>
</tr>
<tr>
<td>EMA Rod Extension</td>
<td>RC20-100</td>
<td>-10V~+10V</td>
<td>-40.083mm~+40.083mm</td>
</tr>
<tr>
<td>EMA Rod Velocity</td>
<td>Parvex Analog Output</td>
<td>-10V~+10V</td>
<td>-100mm/s~+100mm/s</td>
</tr>
<tr>
<td>Anchorage Compression</td>
<td>RC13-25</td>
<td>-10V~+10V</td>
<td>0~+25mm</td>
</tr>
<tr>
<td>SHA Output Force</td>
<td>U10M/50</td>
<td>-10V~+10V</td>
<td>-50KN~+50KN</td>
</tr>
<tr>
<td>EMA Output Force</td>
<td>U10M/50</td>
<td>-10V~+10V</td>
<td>-50KN~+50KN</td>
</tr>
<tr>
<td>Jack Pressure 1</td>
<td>P901-0002</td>
<td>-10V~+10V</td>
<td>0~200bar</td>
</tr>
<tr>
<td>Jack Pressure 2</td>
<td>P901-0002</td>
<td>-10V~+10V</td>
<td>0~200bar</td>
</tr>
<tr>
<td>EMA Torque Demand</td>
<td>XPC Out1</td>
<td>-10V~+10V</td>
<td>-40Nm~+40Nm</td>
</tr>
<tr>
<td>Servovalve Amplifier Input</td>
<td>XPC Out2</td>
<td>-10V~+10V</td>
<td>-50mA~+50mA</td>
</tr>
</tbody>
</table>
Appendix B Static Loading Experiment for Stiffness Measurement

The transmission compliance on test bench is very complex and driven by many mechanical components. As indicated in Fig B-1, in the SHA channel, so many compliances are involved, such as the compliance between load and actuator rod, the compliance of rod, the hydraulic compliance, the actuator body compliance and the compliance between actuator and frame. Except for the hydraulic compliance, all the other compliances form the overall transmission compliance in a series combination.

Therefore, in order to accurately reproduce the transmission compliance on virtual test bench, all the above mentioned compliance should be measured. In fact, the stiffness of actuator rod and body is very high close to $5 \times 10^{10}$ N/m that their influence can be neglected. So, the main contributor comes from the connecting parts between actuator, load and frame. Accordingly, two experiments are done to measure their stiffness.

First: stiffness between actuator and load

Two sensitive position sensors are placed as shown in Fig B-2 to measure the displacement of the attached components. The SHA is powered to load the structural part. The deformation of this part can be calculated as the difference between two position sensors.
While the rod extends, the two position sensors do rotary motion around the load connecting axis. So, their distances from the axis in vertical direction are measured to remove the rotary influence. The measurement result of position sensor 2 should be corrected as:

\[ \Delta x_2 = \Delta x_{2o} \times \frac{82}{109} \]

where \( \Delta x_{2o} \) is the original measurement results.

The measurement result is displayed in Fig B-3.

The linearized stiffness between the load and the actuator rod is closed to \(8.0 \times 10^7\) N/m. Meanwhile, a lost motion about 0.1mm is found around the null transmitted load area.

**Second:** stiffness between actuator and frame
Appendix

The two position sensors are placed as displayed in Fig B-4 to measure the deformation of the part between actuator and frame. The anchorage is set to locked mode.

![Fig B-4 Stiffness measurements between actuator and frame](image)

Same process as the first one, the measurement result is shown in Fig B-5.

![Fig B-5 Stiffness between actuator and frame](image)

The linearized stiffness between actuator and frame is close to $2.5 \times 10^8$ N/m. A lost motion is also found around the null external force area caused by the backlash.

However, due to the limitation of test bench, only the positive force is added, as shown in Fig B-3 and Fig B-5. This makes impossible to observe the compliance effect under negative load. But in normal condition, the component stiffness under negative load is similar to that under positive load. Therefore, they are assumed as symmetrical on virtual test bench.
In addition, the connection mechanization between EMA and frame is same as that of SHA, so its stiffness is considered also equal to $2.5 \times 10^8 \text{N/m}$.

While reproducing the actuator transmission compliance in the virtual test bench, the overall compliance involving both SHA and EMA channels is experimentally obtained, as shown in Fig 3-15. But, accurately distributing this compliance into the SHA and EMA channels is very difficult as the transmission compliance changes under different connection mode. Therefore, we choose a simple solution, in case both SHA and EMA are connected: the EMA transmission stiffness is only driven by the compliance between actuator and frame, as shown in Fig B-5, with a value of $2.5 \times 10^8 \text{N/m}$; the SHA is distributed the rest part, including the compliance of load, compliance between SHA and frame and the compliance of test bench. With this setting, the pre-validation of the static force equalization strategies on the basis of virtual test bench in chapter 4 is not influenced because the series combination of SHA and EMA transmission stiffness is similar to the real condition. This can be confirmed by the comparison between simulation responses and experimental results as shown in Fig 4-26.

While using the virtual test bench in chapter 5, the transmission stiffness is identically set to a very high value in order to place the research in a worst case. So, the above mentioned how to distribute transmission compliance in each channel has also no influence on the final result.
Appendix C EMA Roller-Screw Compliance

As mentioned in chapter 2, the EMA channel overall stiffness is the series combination of the EMA roller-screw stiffness and the transmission stiffness between EMA and load. This overall stiffness plays an important role on the dynamic force equalization (in rejection condition), meanwhile the frequency of the oscillation appeared in outputs is also driven by it. Therefore, accurately modeling of this part is very important.

The EMA roller-screw stiffness was measured on passive mode by Doc. Wissam [41]. During the experiment, the EMA was powered off and set to break mode. The SHA was used to load the EMA. The experiment results are shown in Fig C-1.

![Fig C-1 EMA roller-screw compliance under passive mode](image)

The experiment result shows a lost motion around the null transmitted force area due to the backlash. In order to presents this nonlinear effect, a model is proposed as:

\[
\Delta x = \frac{1}{S_0} F_r + \frac{F_s}{S_1} \tanh\left(\frac{F_r}{F_r}\right)
\]

where:

- \(S_0\) Reference force [N]
- \(F_s\) Force applied on the screw [N]
- \(S_1\) High load stiffness [N/m]
- \(S_1\) Low load stiffness [N/m]
- \(\Delta x\) Roller-screw deformation [m]

The idea identification results are: \(S_0=3.79\times10^8\text{N/m}\) (the high stiffness out of the backlash domain), \(S_1=1.34\times10^8\text{N/m}\) (the low stiffness within the backlash domain) and \(F_r=3829\text{N}\). It will produces a lost motion of \(S_1/F_r=28\mu\text{m}\).
Université de Toulouse
Institut National des Sciences Appliquée – Université Paul Sabatier
Institut Clément Ader

Stratégie d'égalisation d'effort dans les systèmes d'actionnement actif-actif impliquant les technologies servo-hydraulique et électro-mécanique

Présentée et soutenue par

WANG Lijian

Directeur de Thèse: Prof. Jean-Charles Maré
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Autres Membres du Jury: Prof. Bideaux Eric
                        Prof. Yongling Fu

Toulouse
18 Dec 2012
Introduction

Pendant ces dernières années, avec la croissance rapide du voyage aérien, l'industrie aéronautique contribue sensiblement aux émissions de carbone. L'IPCC prédit que le pourcentage d'émissions de CO$_2$ issues de l'aviation commerciale sera porté à 3 % en 2050. Cependant, malgré une croissance du nombre de passagers qui est en moyenne de 5 % par an, l'aviation a réussi à réduire sa croissance d'émissions par autour de 3 % (ou environ 20 millions de tonnes annuellement). Ceci est réalisé par l'investissement massif dans les nouvelles technologies et les procédures d'exploitation. Comme les remplaçants des "best-sellers" d'Airbus et de Boeing, A320Neo et B737Max, les prochains aéronefs commerciaux seront très significativement plus économiques et plus respectueux de l'environnement tout en offrant un niveau de sécurité encore accru.

En parallèle avec le développement de nouveaux moteurs, l'usage généralisé des composites pour les structures, l'amélioration des qualités aérodynamiques, l'amélioration technologique de systèmes secondaires de puissance jouent aussi un rôle important dans cette évolution. Parmi ces systèmes de bord, les réseaux de distribution de puissance, les systèmes de commande de vol et de train d'atterrissage ont un fort potentiel d'amélioration. Sur les avions actuels, l'hydraulique est le réseau secondaire principal de puissance. La puissance hydraulique produite par prélèvement mécanique sur les réacteurs alimente les actionneurs de commande de vol et des trains d’atterrissage par un réseau de distribution difficile à intégrer, lourd et requérant des actions de maintenance spécifiques. Il est donc avantageux de le remplacer par un réseau de puissance électrique.

Sur les avions commerciaux récemment développés, comme l'Airbus A380 et le Boeing B787, des réseaux d'énergie électrique ont été utilisés plus largement pour remplacer partiellement un réseau hydraulique. En raison de leur manque de maturité, les actionneurs électriques sont seulement utilisés comme le soutien ou secours des actionneurs hydrauliques. Cependant, avec les efforts constants dans l'aviation, les actionneurs électriques seront certainement de plus en plus impliqués dans la prochaine génération d'avions. Pour relever les challenges de fiabilité des fonctions critiques, une solution intéressante pourrait consister à utiliser des solutions d'actionnement mixtes en combinant en mode actif/actif des
actionneurs à source de puissance électrique et hydraulique, en particulier pour les commandes de vol primaires.

Comme indiqué sur la figure 1, pour un tel système d’actionnement hybride qui fonctionne en mode actif/actif, un problème important réside dans l’équilibrage des efforts développés par chacun des actionneurs. À cause de leurs technologies dissemblables qui produisent des comportements absolument différents, les actionneurs ne peuvent pas produire exactement la même position à chaque instant et ils luttent l’un contre l’autre. Le combat d’effort peut accélérer la fatigue, réduire la durée de vie et augmenter la consommation d’énergie. Donc, les efforts des actionneurs doivent être égalisés pour reduire le combat d’effort à un niveau admissible.

Cette thèse se propose d’aborder la question d'égalisation d'effort d'un système d'actionnement hybride qui est composé d'un actionneur servo-hydraulique (SHA, à source de puissance hydraulique) et d'un actionneur électromécanique (EMA, à source de puissance électrique), les deux actionneurs fonctionant en mode actif/actif. Les valeurs numériques adoptées correspondent à l'actionnement d'un aileron d'avion monocouloir. La thèse est organisée selon la figure 2:
Chapitre 1 État de l'Art et Banc d’essai

Dans ce chapitre, l'état de l'art en actionnement pour les commandes de vol est présenté. Pour commencer, les fonctions d'opération et l'histoire de développement des actionneurs sont présentées. Ensuite, l'aspect fiabilité, notamment à travers la redondance est abordé avant de terminer par les commandes de vol électriques (fly-by-wire ou FBW).

Pour valider expérimentalement nos travaux de recherche, un banc d'essai d'actionnement hybride qui est composé d'un SHA et d'un EMA est employé. Le banc d'essai a été construit pour l'évaluation d'un prototype d'EMA utilisé en générateur d'effort antagoniste, figure 1-1 face à un SHA émulant un actionneur de commande de vol. Le banc d'essai sera utilisé pour valider les stratégies d'égalisation d'effort dans un environnement de laboratoire, au niveau de maturité TRL 4.

Le banc d'essai inclut un SHA industriel et un prototype d'EMA qui combine un moteur BLDC conduisant une vis à rouleaux inversée. Un bras équipé de deux masselottes est utilisé pour reproduire l'inertie de la charge avion. Il est connecté à l'extrémité de tige des actionneurs par un levier. Les actionneurs sont fixés sur le banc par deux ancrages, qui peuvent simuler l'élasticité de la cellule d'avion et qui peuvent être ajustés en fonction des conditions de test. En outre, un capteur d'effort est installé à l'extrémité de tige de chacun des actionneurs pour mesurer leurs efforts et donc la valeur instantanée du combat d'effort.

Le système électrique de banc d'essai est présenté sur la figure 1-2. La commande du système d'actionnement redondant est implémentée dans un ordinateur XPC Target, qui est utilisé pour simuler les deux calculateurs de commande de vols (FCC). Deux sorties analogiques sont

Fig 1-1 Schéma du banc d'essai
élaborées et envoyées séparément à l'amplificateur tension-courant du SHA servovalve et comme consigne de couple pour le variateur Parvex associé au moteur de l'EMA.

Lorsque le système d'actionnement redondant fonctionne en mode actif/actif, l'objectif de la thèse est de concevoir les contrôleurs de position avec des caractéristiques d'égalisation d'effort pour le SHA et l'EMA afin d'assurer à la fois les performances de poursuite de position de consigne, de rejection de l'effort de charge et d'égalisation d'effort tout en assurant ségrégation et robustesse avec un faible niveau de complexité.

Chapitre 2 Conception de la Commande en de Position pour les Actionneurs

Le principal défi de présent chapitre est de présenter le combat d'effort avec une modélisation très simple. Avec l'approche linéaire, la commande en position des SHA et EMA est conçue de façon préliminaire : une commande proportionnelle plus un filtre réjecteur pour le SHA et une commande cascade (position/vitesse/couple pour l'EMA, comme indiqué sur la figure 2-1 et la figure 2-2.

Fig 1-2 Schéma du système de commande du banc d'essai

Fig 2-1 Contrôleur P+ filtre réjecteur pour le SHA
Ensuite, les commandes en position sont validées sur le banc d’essai et montrent une bonne performance qui satisfait les exigences. Les résultats expérimentaux sont montrés sur la figure 2-3 et la figure 2-4.
sont globalement atteintes lorsqu'elles sont appliquées au modèle linéaire de l'actionnement. Cependant, le combat d'effort entre SHA et EMA reste important en l'absence de fonction d'égalisation. A cause de leur différence de comportement statique et dynamique, le SHA et l'EMA ne cherchent pas à produire à chaque instant la même position, partageant les efforts de charge inéquitablement. Même avec des modèles linéaires des actionneurs, ce chapitre montre à quel point il est important de traiter avec soin l'égalisation d'effort statique et dynamique. Pour récapituler l'analyse, l'origine du combat d'effort est présentée dans le tableau 2-1.

| Combat d'e effort causé par la consigne pilote (condition de poursuite) | Statique (effet majeur) | Principalement conduit par la offset de position de actionneur, tandis que les erreurs statiques des positions de SHA et EMA sont identiques ou semblables, ce combat d'effort pourrait être diminué. |
| Dynamique (effet très important) | Principalement engendré par la dynamique de poursuite des actionneurs, tandis que le SHA et EMA ont la dynamique de poursuite identique ou semblable, ce combat d'effort pourrait être diminué. |

| Combat de effort causé par charge externe (condition de rejection) | Statique (effet important) | Principalement conduit par la raideur du boucle fermée et la raideur de transmission des actionneurs, tandis que le SHA et EMA ont la raideur de rejet identique ou semblable, ce combat d'effort pourrait être diminué. |
| Dynamique (effet mineur) | Principalement conduit par la dynamique de rejet et la raideur équivalente globale des actionneurs, comme dans ce cas la fréquence d'oscillation est beaucoup plus haute que la bande passante de la commande en position, il est difficile de la réduire par une action d'égalisation. |

Selon le tableau 2-1, l'égalisation d'effort se concentre principalement sur la statique dans toutes les conditions et la dynamique dans la condition de poursuite. Dans les parties suivantes, on proposera les stratégies d'égalisation d'effort basées sur la commande en position qui est conçue pour le SHA et EMA dans le présent chapitre. L'égalisation d'effort statique sera adressée comme la première étape et ensuite l'égalisation d'effort dynamique sera traitée.

**Les conclusions importantes du présent chapitre :**

- **Conclusion 1**: le modèle linéaire ne peut pas montrer le combat d'effort statique sous la charge nulle tant que l'on ne considère aucune compensation.
- **Conclusion 2**: reproduire le combat d'effort avec le prototype virtuel exige une
modélisation fine avec beaucoup de non-linéarités et une attention aux élasticités structurales.

- **Conclusion 3**: le combat d’effort dynamique est beaucoup plus critique dans la condition de poursuite que dans la condition de rejet.

**Chapitre 3 Construction de Banc d’essai Virtuel**

Pour plus précisément représenter le comportement du système d’actionnement redondant et permettre l’analyse de robustesse, un banc d’essai virtuel qui implique la plupart des non-linéarités a été construit dans l’environnement LMS AMESIM. Le banc d’essai virtuel a bien reproduit le traitement du signal (comme les filtres, la dynamique de capteur, l’échantillonnage et la quantification) et les comportements physiques non-linéaires (comme pour l’étage pilote de servovalve et l’étage de puissance, le frottement de vis à rouleaux, etc). Le SHA et EMA virtuel sont montrés sur la figure 3-1 et la figure 3-3.

Ensuite, le banc d’essai virtuel a été validé en comparant les réponses simulées et expérimentales. La validation a été exécutée actionneur par actionneurs. Comme indiqué sur la figure 3-2 et la figure 3-4, le banc d’essai virtuel a fourni une bonne représentation des comportements réels.
Après cela, le banc d’essai virtuel a été utilisé pour représenter le combat d’effort entre SHA et EMA. Il a été trouvé que les non-linéarités présentées avaient une grande influence sur la
performance, particulièrement l’élasticité de transmission et le jeu. Quelques conclusions du combat d’effort ont été confirmées par les simulations du banc d’essai virtuel:

- Le combat d’effort statique est principalement conduit par les offsets des positions des actionneurs (causé par l’offset de capteur, la tolérance industrielle etc), la raideur du boucle fermée et la raideur de transmission.
- Le combat d’effort dynamique est principalement conduit par la dynamique de poursuite /rejet des actionneurs et la raideur globale (y compris la raideur hydraulique, la raideur de transmission et la raideur de vis à rouleaux).

Malheureusement, le banc d’essai réel n'a pas été prévu pour produire une charge extérieure sur les deux actionneurs. Il a donc été utilisé pour valider le prototype virtuel sur lequel seront simulées les stratégies d’égalisation en présence de charges extérieures permanentes.

Pour placer le travail de recherche dans des conditions défavorables, les simulations seront réalisées dans les conditions suivantes :

- **Pour l’égalisation d’effort statique**: le banc d’essai est utilisé pour valider les stratégies d’égalisation d’effort sans charge externe, parce qu'il implique plus d’incertitudes, comme l’offset de capteur, la tolérance de fabrication et de réglage, etc. Il a été confirmé que ces incertitudes placent le combat d’effort statique sur une condition beaucoup plus pénalisante. Le banc d’essai virtuel sera utilisé pour la pré-validation et le banc réel pour la validation.

- **Pour l’égalisation d’effort dynamique**: comme indiqué pendant des simulations, une raideur de transmission plus haute aggrave le combat d’effort dynamique. Donc, en exécutant la validation de stratégies d’égalisation de effort dynamiques dans le banc d’essai virtuel, les raideurs de transmission de SHA et EMA sont mises à une valeur très élevée de $1.0 \times 10^9$ N/m.

**Les conclusions importantes du présent chapitre :**

- **Conclusion 1** : les conclusions des origines de combat d’effort obtenues à partir de l’approche linéaire sont globalement confirmées dans le banc d’essai virtuel. Les effets non-linéaires augmentent significativement le montant de combat d’effort.
- **Conclusion 2** : le banc d’essai réel ne permet pas d’appliquer la charge externe sur les
actionneurs hybride SHA/EMA et il présente un jeu excessif qui n'est pas réaliste.

- **Conclusion 3** : le banc d'essai virtuel soigné pour bien modéliser les effets non-linéaires peut reproduire de façon satisfaisante les comportements réels. Il sera utilisé comme un remplaçant au banc d'essai réel pour la validation de stratégies d'égalisation d'effort.

Chapitre 4 Étude de l'Égalisation d'Effort Statique

Dans ce présent chapitre, l'égalisation d'effort statique (SFE) du système d'actionnement hybride a été soigneusement adressée: bien que la stratégie SFE doive être stable, aucune attention n'a été accordée au combat d'effort dynamique (DFF). Les sources de combat d'effort statique ont été premièrement étudiées. La raideur du boucle fermée, la raideur de transmission et l’offset de position des actionneurs ont été identifiés comme des sources majeures de SFF. Nous avons montré que le SFF pourrait être significativement réduit en égalant la raideur de transmission (par conception) et en ajustant la position offset $E_s$ et $E_e$ comme indiqué sur la figure 4-1 et l'équation suivante.

\[ S_s^{-1} F_s + S_e^{-1} F_e + E_s = S_s^{-1} F_s + S_e^{-1} F_e + E_e \]

\[ F_s + F_e = F_L \]

Ensuite, sur cette base, nous avons proposé 3 stratégies SFE :

- La première utilise une information en retour de combat d'effort intégrale (SFE-1, IFFF) pour ajuster les signaux de position des actionneurs, comme indiqué sur la figure 4-2
- La deuxième asservit le SHA effort comme esclave de l’effort produit par l’EMA qui est contrôlé en position (SFE-2.1, SF/EP), comme indiqué sur la figure 4-3
La troisième était l’opposée de la deuxième (SFE-2.2, SP/EF), comme indiqué sur la figure 4-4. La mauvaise performance de l’EMA en asservissement d’effort, principalement due à la forte inertie réfléchie de son moteur nous a conduit à éliminer cette solution.

Par ailleurs, a été évalué l’intérêt de rendre l’EMA transparent pour l’SHA, en l’asservissant à effort nul (SP/ENL).

À la fin, les stratégies de SFE-1, SFE-2.1 et SP/ENL ont été validées avec le banc d’essai réel après pré-validation avec le banc d’essai virtuel. La figure 4-5 montre le SFF expérimental pour les différentes solutions testées. Les stratégies SFE-1 et SFE-2.1 seront conservées lors de la conception des stratégies d’égalisation d’effort dynamiques.
Les conclusions importantes de présent chapitre :

- **Conclusion 1** : l’ajout d’un offset de position sur les boucles de position est une façon efficace de réaliser le SFE. Quand elle est mise en œuvre avec une action intégrale limitée (SFE-1), la SFE et la performance du boucle fermée peuvent être simultanément assurées.

- **Conclusion 2** : la stratégie de commande en effort maître/esclave donne des résultats satisfaisants lorsque le SHA est l’actionneur esclave.

**Chapitre 5 Étude de l’Égalisation d’Effort Dynamique**

Ce chapitre est organisé selon la même logique du chapitre 4. Pour commencer, les sources de combat d’effort dynamique ont été étudiées par une approche linéaire par l’étude la fonction de transfert de combat d’effort $\gamma$ lorsque le SHA et l’EMA sont commandés en position sans stratégie d’égalisation d’effort. Deux conditions importantes sont nécessaires pour réaliser efficacement la DFE :

- les dynamiques de poursuite du SHA et de l’EMA doivent être rendues semblables.
• les dynamiques de rejection d’effort du SHA et de l’EMA doivent idéalement être adaptées aux valeurs des raideur de transmission.

A partir de ces résultats, nous avons proposé et évalué sur le banc virtuel trois stratégies de contrôle de DFE :

• **DFE-1** : le SHA et l’EMA sont commandés en position et forcés à une même dynamique de poursuite par action d’anticipation en accélération et en vitesse. Ceci est réalisé par compensation du débit fonctionnel du SHA, de la force contre électromotrice et du couple d’accélération du rotor pour l’EMA. La consigne de position a été filtrée par un générateur de trajectoire du 3ème ordre. Une action intégrale de retour d’effort du type SFE-1 a été ajoutée pour assurer l’égalisation statique d’effort, comme montré sur la figure 5-1.

• **DFE-2** : cette stratégie était une extension du SFE-1 en utilisant un filtre PID sur le retour de combat d’effort, à la place de la seule action I testée pour l’égalisation statique. Le SHA et l’EMA étaient commandés en position. Le combat d’effort a été utilisée pour compenser chaque boucle de position, comme montré sur la figure 5-2.

• **DFE-3** : cette stratégie était aussi une extension de SFE2.1 dans laquelle l’EMA était commandé en position et le SHA en suiveur d’effort d’EMA. Une commande en effort de type PID a été combinée avec une action d’anticipation vitesse pour obtenir à la fois une bonne dynamique et un bon rejet de l’influence de la charge aérodynamique, comme montré sur la figure 5-3.

![Fig 5-1 Schéma de DFE-1](image-url)
Ensuite, la dynamique de poursuite, le rejet et l'égalisation d'effort de ces DFE stratégies ont été comparés par simulation. Nous présentons les résultats sur la figure 5-4 et la figure 5-5. Le DFE-1 a montré une bonne égalisation d'effort et la meilleure dynamique de poursuite et de rejet. La solution DFE-2 offrait la meilleure égalisation d'effort, mais avec la pire dynamique de poursuite et de rejet. La solution DFE-3 présentait une égalisation d'effort et un rejet pauvre, mais avec une bonne dynamique de poursuite.
Par la suite, la robustesse de ces stratégies DFE a été étudiée par l’exploration de paramètres selon la méthode de Monte-Carlo. Au final, nous avons considéré 17 paramètres de système et 14 indicateurs de performance qui ont été utilisés pour évaluer la robustesse des solutions.
proposées. Pour chaque étude, 2000 simulations ont été exécutés avec une variation aléatoire des paramètres, selon une distribution normale. Les résultats d'analyse sont montrés dans le tableau 5-1, où la valeur de % représente les simulations qui ne satisfont pas à l'exigence de performance. L'indicateur général met en évidence toutes les simulations qui ne satisfont pas au moins un des critères de performance.

**Tab 5-1** Résumé de performance de système sous incertitudes diverses

<table>
<thead>
<tr>
<th>Indicateur de performance</th>
<th>DFE-1</th>
<th>DFE-2</th>
<th>DFE-3</th>
</tr>
</thead>
<tbody>
<tr>
<td>#1 Valeur maximale DFF en poursuite</td>
<td>34.5%</td>
<td>45.46%</td>
<td>17.93%</td>
</tr>
<tr>
<td>#2 Valeur positive maximale de DFF en poursuite</td>
<td>0.55%</td>
<td>0.06%</td>
<td>0.43%</td>
</tr>
<tr>
<td>#3 Valeur négative maximale de DFF en poursuite</td>
<td>25.7%</td>
<td>8.39%</td>
<td>6.35%</td>
</tr>
<tr>
<td>#4 Valeur maximale de DFF en rejet</td>
<td>1.6%</td>
<td>5.11%</td>
<td>8.22%</td>
</tr>
<tr>
<td>#5 Valeur positive maximale de DFF en rejet</td>
<td>0.2%</td>
<td>0.88%</td>
<td>3.92%</td>
</tr>
<tr>
<td>#6 Valeur négative maximale de DFF en rejet</td>
<td>1.25%</td>
<td>0.25%</td>
<td>4.3%</td>
</tr>
<tr>
<td>#7 SFF en poursuite</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>#8 SFF en rejet</td>
<td>0</td>
<td>0</td>
<td>2.8%</td>
</tr>
<tr>
<td>#9 Premier dépassement en poursuite</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>#10 Temps de réponse en poursuite</td>
<td>7.35%</td>
<td>83.48%</td>
<td>6.85%</td>
</tr>
<tr>
<td>#11 Erreur statique en poursuite</td>
<td>0</td>
<td>0</td>
<td>0.06%</td>
</tr>
<tr>
<td>#12 Erreur statique en rejet</td>
<td>3.35%</td>
<td>65.83%</td>
<td>71.79%</td>
</tr>
<tr>
<td>#13 La vitesse de charge en cas de poursuite</td>
<td>1.85%</td>
<td>31.65%</td>
<td>4.36%</td>
</tr>
<tr>
<td>#14 La vitesse de charge en cas de rejet</td>
<td>0</td>
<td>0</td>
<td>5.85%</td>
</tr>
<tr>
<td>Général</td>
<td><strong>40.35%</strong></td>
<td><strong>87.83%</strong></td>
<td><strong>73.16%</strong></td>
</tr>
</tbody>
</table>

La solution DFE-1 a montré la meilleure robustesse, mis à part le DFF en poursuite. La solution DFE-2 a présenté la pire robustesse, principalement concernant la dynamique de poursuite et la pauvre capacité de rejet. La solution DFE-3 a montré une mauvaise robustesse, sa performance de rejet étant insuffisante.

L'analyse des causes de dégradation de la performance a montré que la viscosité fluide hydraulique jouait un rôle majeur sur la performance de système, à travers son effet sur la dynamique du SHA à très faible température, quelque soit la stratégie d'équilibrage d'effort adoptée.

La ségrégation et la complexité de ces stratégies DFE ont été étudiées pour fournir des critères supplémentaires de comparaison.

Finalement, la performance générale de ces stratégies DFE est résumée dans le tableau 5-2.
Résumé Long de la Thèse – WANG Lijian

Tab 5-2 Comparaison générale des stratégies DFE

<table>
<thead>
<tr>
<th>Articles</th>
<th>Sans FE</th>
<th>DFE-1</th>
<th>DFE-2</th>
<th>DFE-3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Dynamique de Poursuite de Charge</td>
<td>++</td>
<td>++</td>
<td>0</td>
<td>++</td>
</tr>
<tr>
<td>Rejet de Charge</td>
<td>++</td>
<td>++</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>Égalisation d'Effort Dynamique</td>
<td>− −</td>
<td>+</td>
<td>++</td>
<td>0</td>
</tr>
<tr>
<td>Égalisation d'Effort Statique</td>
<td>− −</td>
<td>+</td>
<td>++</td>
<td>+</td>
</tr>
<tr>
<td>Robustesse</td>
<td>+</td>
<td>+</td>
<td>−</td>
<td>0</td>
</tr>
<tr>
<td>Ségrégation</td>
<td>++</td>
<td>+</td>
<td>0</td>
<td>−</td>
</tr>
<tr>
<td>Complexité</td>
<td>++</td>
<td>0</td>
<td>+</td>
<td>+</td>
</tr>
<tr>
<td>Général</td>
<td>−</td>
<td>++</td>
<td>+</td>
<td>0</td>
</tr>
</tbody>
</table>

- **Conclusion 1** : pour des applications critiques de sécurité comme l’actionnement de contrôle de vol principal ou les systèmes de train d’atterrissage dans l’aéronautique, le DFE-1 est la solution plus attractive car il combine bonne ségrégation et bonne robustesse.

- **Conclusion 2** : pour des applications qui exigent plus d’attention sur l’égalisation de force et moins d’attention sur la ségrégation, le DFE-2 est une bonne solution candidate.

- **Conclusion 3** : le DFE-3 pourrait être utilisé pour examiner le contrôle hybride de système d’actionnement dans son ensemble avec une vue entrée et sorties multiples (MIMO) pour la conception de contrôleur sans contrainte forte de ségrégation.

**Conclusion Générale**

Selon les travaux conduits dans cette thèse, pour réaliser une bonne performance d’égalisation d’effort, les recommandations ci-après peuvent être émises :

Réduire le montant de combat d’effort au niveau de design mécanique:

- Par conception, faire en sorte que les raideurs globales de chaque voie, EMA et SHA, soient identiques. Égalant autant que possible ces raideurs par conception bénéficie énormément au DFE en conditions de rejet.
- Égaler les capacités de puissance des deux voies EMA et SHA. En particulier, la capacité d’accélération permet d’améliorer la performance DFE en poursuite.

Réduire le niveau de combat d’effort par la boucle de position :
Egaler les dynamiques de poursuite de chaque actionneur. Ceci améliorera naturellement la performance DFE.

Egaler la raideur de rejet statique : le SFF en rejet est significativement réduit si la raideur de rejet générale équivalente de chaque voie est semblable.

Réduire le niveau de combat d'effort par une commande d'égalisation d'effort :

- Pour l'égalisation d'effort statique, la solution SFE-1 qui exploite l'intégrale du combat d'effort pour compenser le contrôle de position est une solution efficace.
- Pour l'égalisation d'effort dynamique, la solution DFE-1 qui introduit des actions d'anticipation vitesse et accélération et force des dynamiques de poursuite semblables pour les deux voies est très efficace. Elle offre un bon équilibre entre ségrégation et robustesse.

La poursuite de nos travaux pourrait s'orienter dans plusieurs directions :

- Valider expérimentalement les stratégies d'égalisation d'effort proposées. Avant des essais d'environnement, un banc d'essai d'actionnement redondant quadruple qui implique 2 SHAs et 2 EMAs va être construit au laboratoire, Toulouse. Il pourrait être utilisé pour valider les stratégies proposées d'une façon plus réaliste, où un SHA (ou un EMA) pourrait reproduire dynamiquement la charge aérodynamique sur l'ensemble SHA/EMA du système d'actionnement redondant.
- Développer des lois de commande non-linéaires. Nous avons montré sur le banc virtuel que la performance d'égalisation d'effort dynamique était insatisfaisante pour les fortes demandes de position qui engendrent des vitesses de charge élevées. Pour traiter ce problème, la connaissance des non-linéarités devrait être mieux exploitée en exploitant la théorie de la commande non-linéaire et/ou robuste.
- Prendre en compte l'effet de la température dans la stratégie d'égalisation d'effort. Selon l'étude de robustesse exécutée dans le banc d'essai virtuel, la performance d'égalisation d'effort a été trouvée insatisfaisante en conditions de température basse. Il serait intéressant de considérer l'influence de température sur la performance de l'actionneur (la viscosité fluide hydraulique et le frottement) dans la synthèse de la commande.
- Notre présente recherche est sur le niveau de maturité TRL4 qui correspond à la
validation des solutions proposées dans un environnement de laboratoire. Le travail en cours devrait progressivement se placer au plus haut niveau. Sur la route vers le niveau TRL9 final, où la stratégie d'égalisation d'effort doit être mise en compagnie aérienne, il y a encore beaucoup de travail à faire...
Titre de la thèse en français

Stratégie d'égalisation d'effort dans les systèmes d'actionnement actif-actif impliquant les technologies servo-hydraulique et électro-mécanique

Résumé

L'évolution vers les avions plus électriques engendre des efforts importants pour développer des actionneurs à source de puissance électrique pour les commandes de vol. Pour de telles applications critiques, il est peut-être intéressant dans le futur d'associer à une même surface de contrôle un actionneur conventionnel à source de puissance hydraulique et un actionneur à source de puissance électrique mais ceci pose un problème important lorsque les deux actionneurs sont actifs simultanément : comme chacun essaie d'imposer sa position à l'autre, les deux actionneurs luttent l'un contre l'autre en développant des efforts néfastes qui ne sont pas utilisés par la charge. L'objet du présent travail est de proposer des stratégies d'égalisation d'effort pour un système d'actionnement impliquant ces deux types d'actionneurs qui opèrent en mode actif-actif. La première étape est de concevoir leur commande en position et de la valider sur banc d'essai. Puis un banc d'essai virtuel fidèle à la réalité est élaboré dans l'environnement de simulation AMESim pour pouvoir évaluer facilement les différentes stratégies d'égalisation d'effort entre les deux actionneurs. Un effort particulier est porté sur la modélisation fine des phénomènes non linéaires majeurs comme les gains en pression de débit de la servovalve ou encore la sensibilité du frottement dans le système vis écou à la vitesse d'opération et à l'effort transmis à la charge. Ensuite, diverses stratégies d'égalisation d'effort sont proposées et évaluées virtuellement. Pour l'égalisation en statique, il apparaît que l'introduction à autorité limitée d'un offset de position fonction de l'intégrale de la différence d'effort entre les deux actionneurs est une solution efficace. Pour l'égalisation en dynamique, les meilleurs résultats sont obtenus lorsque une même trajectoire d'état est imposée aux deux actionneurs en combinaison avec des actions d'anticipation vitesse et accélération qui présente l'avantage de ne pas requérir de capteur supplémentaire. Pour finir, une étude de robustesse est réalisée a posteriori pour évaluer la sensibilité des indicateurs de performance d'asservissement de position et d'égalisation d'effort aux incertitudes sur les modèles de simulation et sur les points et les conditions de fonctionnement.

Mots Clés

Actif/actif, actionneur, aéronautique, commandes de vol, égalisation d'effort, électrohydraulique, électromécanique, hybride, redondant.