INVESTIGATION IN THE DYNAMIC FORCE EQUALIZATION OF DISSIMILAR REDUNDANT ACTUATION SYSTEMS OPERATING IN ACTIVE/ACTIVE MODE

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Abstract

With the extensive application of power by wire (PBW) actuators in aircraft industry, a lot of challenges have arisen. One important subject is the force equalization of redundant actuation system composed of dissimilar technologies for position control. In the presented work, the research is focused on one hybrid actuation system combining one servo hydraulic actuator (SHA) and one electro-mechanical actuator (EMA). A virtual test bench is built where the key nonlinear effects are modeled with care. Once the model accuracy is evaluated, the virtual test bench is used for studying the force equalization strategies. The static force equalization has been already presented in a former communication [1], in view of that, the reported research is focused on dynamic force equalization. According to the studied origin of force fighting, three dynamic force equalization strategies are proposed, analyzed, simulated and compared. The first one uses one trajectory generator and two pre-compensators to balance the actuators position loop dynamics; the second one introduces force fighting feedback to compensate the position loop; the third one sets SHA to force control to track the output force of EMA.

1 Introduction

With the recent evolution towards more electric aircraft (MEA) in aircraft industry, the hydraulic power networks are being progressively replaced by the electric ones. For example, the flight control actuation system of Airbus A380 is powered by 2 hydraulic and 2 electric networks, in comparison with the triplex hydraulic networks of previous Airbus products A320/A330/A340. Under the 2 electric power networks, 16 PBW actuators which include 8 electro-hydrostatic actuators (EHA) (for rudders and spoilers) and 8 electrical back-up hydraulic actuators (EBHA) (for ailerons and elevators) are employed [2]. More recently, on Boeing B787, 5 EMAs are involved in the flight control system, 4 simplex EMAs for the spoilers and one redundant EMA for the horizontal stabilizer. However, due to the safety issues and the lack of maturity of PBW actuators for application in normal mode, the use of EHAs and EMAs is still limited to secondary flight control systems and/or for backup in primary flight control systems. For the time being, the SHAs still predominate in the primary flight control actuation systems.

The efforts put on the development of PBW actuators will certainly enable them to be employed in active mode for the primary actuation systems. In a transition period, EMAs and EHAs can be combined with SHAs to make a hybrid redundant actuation system driving the load in active/active mode. One major issue of this working mode is the force fighting between actuators. Because of the dissimilar technology and setting/manufacturing tolerances, their static and dynamic behaviors are so different that they do not share the load equally and often fight one against another to position the load. Therefore the force equalization must be addressed with special attention in order to improve the system energy efficiency, reliability and service life.

Because EMA is considered as the ultimate form of PBW actuator, the hybrid configuration involving one SHA and one EMA is studied in this communication. For this work, a hybrid redundant actuation test bench was designed, manufactured and installed. It was sized for the roll control of single aisle aircrafts, as shown in Fig.1. The test bench includes one industrial SHA which is employed to emulate the aircraft one and one prototype of EMA combining one BLDC motor driving an inverted roller-screw.

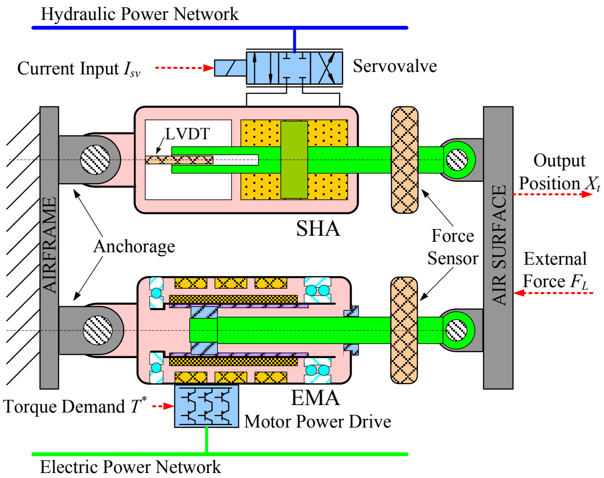

Fig.1. Schematic of Redundant Actuation System

One strategy for the force equalization of SHA/EMA hybrid actuation system is proposed in [3], in which one pre-filter is added to SHA to balance the dynamics difference between SHA and EMA. Meanwhile actuators position difference, velocity difference and force fighting are introduced to compensate the position controller. In [4], the pre-compensators for both actuators are used to introduce velocity and acceleration compensations to help balancing the actuators dynamics. In an author's former communication [1], the static force equalization was studied as the first part of this subject. The force fighting feedback strategy as well as EMA force control strategy proves to significantly reduce the static force fighting in active/active mode.

In the present communication, the focal point is the dynamic force equalization. The virtual redundant actuation test bench employed in former work is improved and adapted to support evaluating the proposed dynamic force equalization strategies. The reasons causing dynamic force fighting are studied and presented on basis a simplified system model. According to that, three control strategies are proposed and simulated. In the first one, the pre-compensator strategy is improved, and a high level segregation is kept by limiting the cross coupling between the two actuation channels. In the second one, the PID force fighting feedback signal is introduced to compensate the actuators position demand. In the third one, the SHA is set to force control to track the output force of EMA which is under position control to pursuit the position demand. Finally the advantages and drawbacks of these three strategies are assessed, compared and concluded.

2 Virtual Test Bench

The virtual test bench, as shown in Fig.2, is built in the AMESim simulation environment with respect to system physics. All the key elements on test bench are accurately modeled and inserted. A special attention is paid to the most significant nonlinear effects, like the servovalve pressure/opening gain, flow/opening gain, spool dynamics, jack friction, EMA roller-screw friction, and so on.

By comparing simulation and experimental results, the virtual test bench has been proven to reproduce accurately the operation of real test bench. It will play an important role on evaluating the performances of the candidate force equalization strategies.

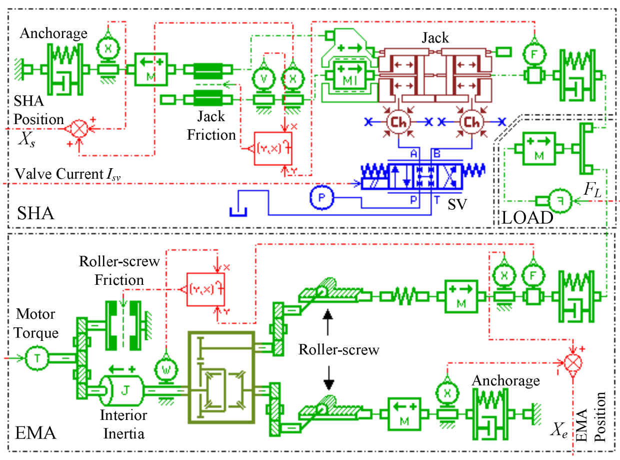

Fig.2. Virtual Test Bench in AMESim

3 Dynamic Force Fighting Reasons

For a redundant actuation system operating on force summing principle, the force fighting is generally caused by the inconsistence between actuators which are demanded to drive a same load together. Because of statics and dynamics difference, the actuators cannot output at any time the same position even under the same position demand. The position difference leads to actuators output force difference. For the hybrid actuation system under study, because of the definitely different behaviors of SHA and EMA, the force fighting becomes larger.

In order to help developing dynamic force equalization control strategies, the force fighting is formulated on basis the simplified model as shown in Fig.3.

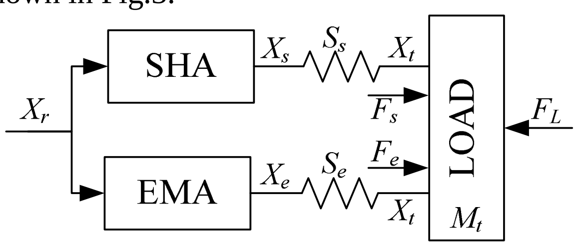

Fig.3. Schematic of Simplified System

where:
X_r: position demand
X_e: EMA output position
F_e: EMA output force
S_e: EMA transmission stiffness
X_s: SHA output position
F_s: SHA output force
S_s: SHA transmission stiffness
X_t: load position
M_t: load mass
F_L: external load force

The force fighting γ is defined as the difference between F_s and F_e.

In former research, the required dynamic performance of SHA is obtained with a pure P controller when it is solely connected to the load. Meanwhile a PD controller is designed for the single EMA. This enables writing the transfer functions of actuators output position as:

SHA position transfer function:

$$X_s = \frac{X_r - G_2(s)F_s}{G_1(s)} \tag{1}$$

EMA position transfer function:

$$X_e = \frac{X_r - H_2(s)F_e}{H_1(s)} \tag{2}$$

In upper equations, the first parts $1/G_1(s)$ and $1/H_1(s)$ present the position pursuit function of actuators. The second parts $G_2(s)/G_1(s)$ and $H_2(s)/H_1(s)$ present the rejection function of actuators to external force.

In addition each actuator's output force can be calculated on basis mechanic deformations between rods and load:

$$F_s = (X_s - X_t)S_s = \frac{S_s X_r - G_1 S_s X_t}{G_1 + G_2 S_s} \tag{3}$$

$$F_e = (X_e - X_t)S_e = \frac{S_e X_r - H_1 S_e X_t}{H_1 + H_2 S_e} \tag{4}$$

Then with the Newton's second law, the load position can be expressed as:

$$X_t = \frac{F_s + F_e - F_L}{M_t s^2} \tag{5}$$

Finally combining equations (1) to (5) gives the force fighting as equation (6):

$$\gamma = F_s - F_e = \frac{M_2(s)X_r + M_3(s)F_L}{M_1(s)} \tag{6}$$

where:

$$M_1 = (G_1 + G_2 S_s)(H_1 + H_2 S_e)M_t s^2 + G_1 S_s (H_1 + H_2 S_e) + H_1 S_e (G_1 + G_2 S_s)$$

$$M_2 = \left[S_s(H_1 + H_2 S_e) - S_e(G_1 + G_2 S_s)\right]M_t s^2 + 2S_s S_e (H_1 - G_1)$$

$$M_3 = G_1 S_s (H_1 + H_2 S_e) - H_1 S_e (G_1 + G_2 S_s)$$

It is clear that the force fighting is driven by two factors: the position demand X_r and the external force F_L. The M_1 is fixed by the system dynamics and directly related to the actuators position dynamics. So in order to weaken the influences of these two factors, M_2 and M_3 should be made as small as possible. When both M_2 and M_3 are equal to zero, the force fighting will be totally removed. This leads to:

$$\begin{cases} H_1 = G_1 \\ H_2 = G_2 - \dfrac{S_s - S_e}{S_s S_e} G_1 \end{cases} \tag{7}$$

According to equation (7), the null force fighting requires that the SHA and EMA have identical pursuit dynamics (the first equation). If the transmission stiffness S_s and S_e are same, the identical rejection transfer functions are also required (the second equation).

Consequently the only way to equalize exactly SHA and EMA output forces without signal cross coupling is to set them with the same position dynamics.

4 Dynamic Force Equalization Strategies

4.1 Requirements of Controller

The force equalization strategy design consists in generating the control inputs servovalve current I_{sv} for SHA and motor torque demand T^* for EMA so that the following performances can be obtained:

- Pursuit performance: the load position magnitude ration is greater than -3dB and the phase lag is lower than 45° for a 3Hz/1mm magnitude sine input;
- Rejection performance: the load position must be not sensitive to the air load and provide a closed loop static stiffness of 2×10^8N/m;
- Force equalization performance: the static force fighting should be fewer than 5% of the rated force 50KN, and 15% for dynamic force fighting.

The transmission stiffness S_s and S_e are assumed to be identical in the following parts.

4.2 Trajectory Pre-compensator Strategy

In [3], it is proposed to feed forward filter the SHA position demand signal and force the SHA to have the same dynamics of EMA. As the pre-filter is calculated on basis linear model which cannot accurately represent the real system, the robustness of this controller is not very good. In addition the essence of this strategy is lowering the SHA dynamics to make it equal the EMA's one. Obviously this decreases the dynamics of whole system. According to this, one strategy is proposed to combine one trajectory generator and two dynamic compensators for balancing the actuators dynamics as well as keeping high robustness and dynamics.

This strategy aims at forcing both actuators to the same dynamics without cross signals as a model reference approach, as shown in Fig.4.

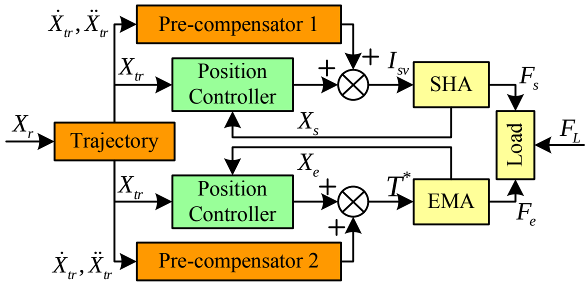

Fig.4. Schematic of Pre-compensator Strategy

4.2.1 Trajectory Generator

The trajectory generator can be considered as the reference model and is designed on basis requirements over position pursuit dynamics. Because the SHA traditional model is third-order, the reference model is also designed as a third-order system to keep consistency.

$$G_{tr}=\frac{1}{(t_i s+1)\left(\frac{1}{\omega_i^2}s^2+\frac{2\xi_i}{\omega_i}s+1\right)} \tag{8}$$

The trajectory is composed by one first-order filter and one second-order filter, with the selected parameters as ω_i=60rad/s, ξ_i=0.707 and t_i=0.017s, which are consistent with the required position dynamics.

4.2.2 Pre-compensators

In order to accurately pursuit the reference model, not only the actuators position, but also the velocity and acceleration should be precisely controlled. With the reference velocity $\dot{X}_{tr}$ and acceleration $\ddot{X}_{tr}$ generated by the trajectory as inputs, two pre-compensators are introduced to compensate the position controllers which are designed for single actuators.

The two pre-compensators are designed on basis physics principle. The necessary control inputs for driving actuators are calculated.

For the SHA, the compensation is added onto the servovalve current I_{sv}. The related linearized valve flow equation is:

$$Q_{sv}=K_{sg}I_{sv}-K_{sc}P_f=A_t\dot{X}_s+\frac{V_t}{4E_y}\dot{P}_f+K_{ac}P_f \tag{9}$$

And the motion equation of load which is solely driven by the SHA is:

$$M_t\ddot{X}_s=A_tP_f-F_{sf}-F_L \tag{10}$$

where:
Q_{sv}: *servovalve output flow*
K_{sg}: *servovalve flow-current gain*
K_{sc}: *servovalve flow-pressure gain*
A_t: *jack rod area*
V_t: *SHA effective volume*
E_y: *hydraulic oil bulk modulus*
P_f: *jack load pressure*
K_{ac}: *jack leakage coefficient*
F_{sf}: *jack friction*

Because the compensation is for position pursuit dynamics, the influence from external force F_L is removed. Moreover the jack friction F_{sf} is so small compared with the rated actuator output force that its influence is also ignored. Finally the compensation can be calculated by combining equations (9) and (10):

$$I_{sv}=\frac{1}{K_{sg}}\left(\dot{X}_{tr}A_t+\ddot{X}_{tr}\frac{M_t}{A_t}\left(\frac{V_t}{4E_y}s+K_{sc}+K_{ac}\right)\right) \qquad (11)$$

The SHA pre-compensator is designed on basis equation (11):

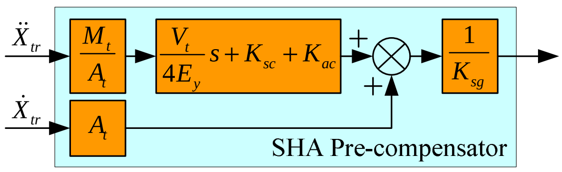

Fig.5. Schematic of SHA Pre-compensators

The servovalve flow-current gain is set as a linear part. The nonlinear effect caused by load pressure is unconsidered. This helps simplifying the calculation and benefitting the system stability. The simulations in virtual test bench prove that the accuracy requirement is met.

For the EMA, the compensation is added onto the motor torque demand T^*. The motion equation of single EMA can be expressed as:

$$T^*=T_m=\ddot{X}_t\left(\frac{2\pi}{l}J_m+\frac{l}{2\pi}M_t\right)+T_{ef}(\dot{X}_t)+\frac{l}{2\pi}F_L \qquad (12)$$

where:
T_m: *motor output torque*
J_m: *EMA equivalent rotation inertia*
l: *roller-screw lead*
T_{ef}: *EMA roller-screw friction torque*

In this actuator, the dynamics of the motor driver is so high (about 600Hz) compared with the EMA working domain (about 10Hz) that the motor output torque can be considered as equal to the demanded value. In addition the effect of the load mass M_t being so small in comparison with the one of rotor inertia J_m is ignored in designing the pre-compensator. The influence of external force F_L is removed for the same reason as in designing the pre-compensator for SHA. The schematic is shown in Fig.6.

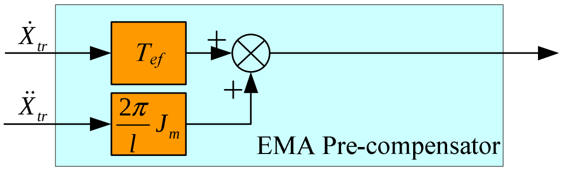

Fig.6. Schematic of EMA Pre-compensators

4.2.3 Simplifying

In order to simplify the structure and improve robustness, the effects of each part in pre-compensators have been studied in detail. By simulations and comparisons, it has been observed that the main contributors of two pre-compensators are the functional flow demand (hydrostatic area times velocity) of SHA and the parasitic inertial torque (rotor inertia times rotor angular acceleration) of EMA.

In addition the key parameters of these two main contributors are the jack area A_t and the EMA rotor inertia J_m which are constant in the whole service life. Oppositely the effects of SHA oil compression and EMA friction are very difficult to anticipate as their models involve unknown or rapidly changing parameters.

So removing the two uncertain parts and only keeping the two main contributors can benefit the performance robustness.

In this strategy, the compensations are only added for balancing the actuator position pursuit dynamics, which are the $1/G_1(s)$ and $1/H_1(s)$ in equations (1) and (2). This means only the force fighting caused by the position demand X_r is considered; the influence of external force F_L is not concerned. Therefore the actuators position controller that can influence the $G_2(s)$ and $H_2(s)$ should be carefully parameterized to balance the system performance on these two factors.

4.3 Force Fighting Feedback Strategy

In upper strategy, the objective is to avoid force fighting. In this strategy, the force fighting is to be decreased after it happens. The force fighting is evaluated and introduced in the actuators position control loops.

This strategy has been widely used in the hydraulic redundant actuation systems working in a parallel force summing configuration. In these systems, the load pressure is a good indicator of the actuators output force because the jack friction and the inertial force are negligible in comparison with the actuators output force. So the pressure difference between actuators is used to estimate the force fighting.

Oppositely the EMA motor current cannot accurately image the EMA output force due to the huge rotor inertial torque and the roller-screw friction. So a force sensor is used on the EMA to measure its output force, meanwhile another one on the SHA to keep consistency.

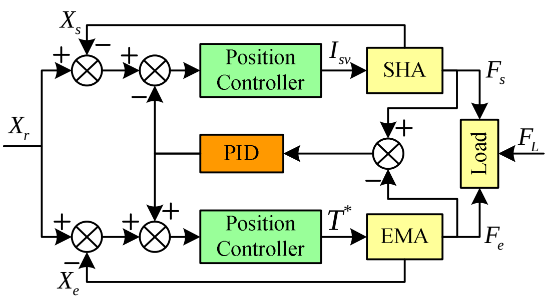

Fig.7. Schematic of Force Fighting Feedback Strategy

In this strategy, the force fighting signal is introduced to compensate the actuators position demand signals after being filtered by one PID controller. Each part of the PID controller has important effects on the force equalization. In simulations, the static force fighting is mainly removed by the integral part; the dynamic one is mainly balanced by the derivative part; and both are influenced by the proportional part.

4.4 SHA Force/EMA Position Strategy

It is known that a linear multi input multi output (MIMO) system is decoupled so that one output is only fixed by one input and the property of each channel can be set separately. On basis this principle, it is proposed to operate one actuator in position control to be in charge of position dynamics while the other one is force controlled to be responsible for force equalization. There are two candidate configurations: either the EMA or the SHA is force controlled.

As the EMA suffers from huge rotor inertia, its force control performance is significantly altered. Oppositely the SHA performance when operated in force control is much better as it is mainly limited by the servovalve dynamics that is quite high (about 80Hz).

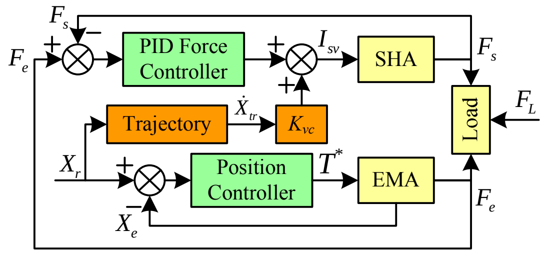

Fig.8. Schematic of SHA force/EMA Position Strategy

In this strategy, except for the redesigned SHA force controller, the velocity compensation is also introduced to help improving SHA force dynamics. The reference velocity is got by the trajectory generator which simulates the system position dynamics. The velocity compensation gain K_{vc} is directly defined from the jack area. Because this velocity compensation is forward, it does not change the system static stiffness. Should this velocity signal be got from the EMA, the closed loop static stiffness would be decreased as it introduces a velocity feedback.

4.5 Simulations over Force Equalization

The pursuit, rejection and force equalization performances of the proposed control strategies are assessed using the virtual test bench.

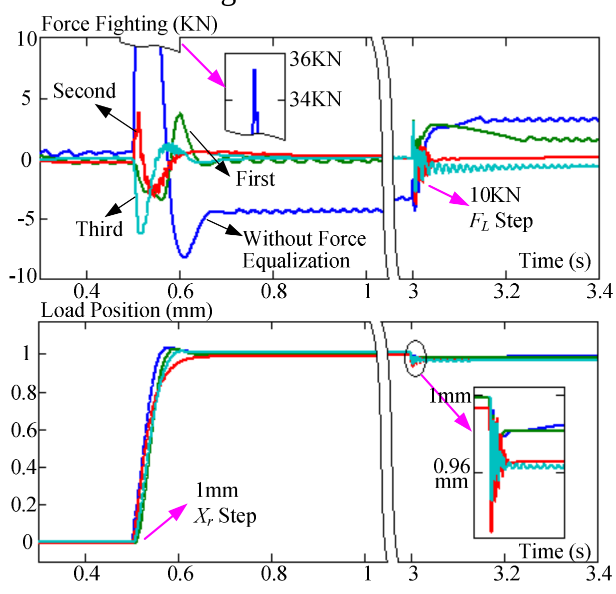

Fig.9. Simulation Results of Control Strategies

Firstly a 1mm position step demand is sent at time 0.5s without external force. As displayed in Fig.9, the dynamic force fighting is greatly reduced while the position pursuit dynamics is acceptable after employing the three proposed force equalization strategies. The peak value of dynamic force fighting is decreased from 35KN to about 6KN (5 times at least). The static force fighting is also well cancelled to almost zero. In these three strategies, the second one has the best performance because of its smallest force fighting and fastest dynamics; the third one is a little worse in reducing the force fighting peak value; the first one is a little worse in dynamics.

Then a 10KN external force step demand is sent at time 3s without any change in position demand. There is almost no reduction in the dynamic force fighting because its frequency range (about 90Hz) is well over the bandwidth of actuators control loop (about 10Hz). Not like the force fighting caused by the position input, its frequency is equal to the position loop's one. This high frequency comes from the actuators equivalent stiffness (about 2×10^8N/m) and load mass (600Kg). With the integral actions in force fighting compensator of the second strategy and in the force controller of third strategy, the static force fighting is almost removed. Under the first strategy, there is no integral effect over the force fighting, so the static force fighting is decreased but not totally removed. In addition the load position sensitivity to external air load is not significantly altered; the position static errors are all smaller than 0.04mm under 10KN external force (make the actuation system static stiffness greater than 2.5×10^8N/m).

4.6 General Comparison

The proposed control strategies are compared here with respect to complexity, segregation and robustness as it is mandatory for safety critical embedded applications.

The complexity criterion is used to assess the architecture of strategies, including design procedures and number of parameters. From this point of view, the second strategy is the simplest one, except for the two position controllers, only one PID force fighting compensator is added. The third strategy follows. It needs redesigning one PID force controller and adding a velocity compensator for SHA. The first strategy is the most complex one as it needs redesigning two compensators and one trajectory generator.

Robustness is one item used to describe how the system performance is sensitive to the uncertainties (parameters, disturbance, etc). The second strategy still offers the best robustness. However under the first and the third strategies, it has been found that the force equalization performance is not robust against the variation of the servovalve flow-current gain. In the first strategy, the gain is used in the pre-compensator of the SHA to calculate the servovalve current. Incorrect parameters will cause over or under compensation and enlarge the actuators force fighting. In the third strategy, the proportional and derivative parts of SHA force controller are found very sensitive to the channel total forward gain in which the flow-current gain plays an important role. This is different from the position controllers which are not so sensitive.

Segregation is a property required for the redundant systems regarding safety issues on aircrafts. The less the cross links between channels, the better the segregation and the higher the safety level. Obviously the first strategy has the best segregation. There is no cross signal at all between actuators which only take a same position demand. The third strategy is also good as only one cross signal is used by the SHA from the EMA output force (the EMA is totally segregated from the SHA). The worst one is the second strategy in which both actuators need the output force signal from the other one. In case one actuator is faulty, the other one also cannot work correctly.

The general comparison of these strategies is summarized in table 1.

Table.1. Comparison of Force Equalization Strategies

Items	Original Controller	First Strategy	Second Strategy	Third Strategy
Pursuit	+++++	+++++	++++	+++++
Rejection	++++	++++	++++	++++
Dynamic FE	+	++++	+++++	+++
Static FE	+	+++	+++++	++++
Complexity	+++++	++	++++	+++
Robustness	++++	++++	+++++	++++
Segregation	+++++	+++++	+++	++++
General	++	++++	+++++	++++

Notation: + means worst, and +++++ means best.

With the simulations at part 4.5, all three strategies were proved to meet the requirements mentioned at part 4.1, not only for the position loop but also for the force equalization. Then the advantages and drawbacks of each strategy are listed and compared. On the basis of this, any proposed strategy can be selected according to the current application. In case force sensors are to be avoided or full segregation is strictly demanded, the first strategy is a good choice. Opposite to this, in case force sensors can be employed and without strict requirements over segregation, the second one is the best choice. The third strategy can be selected for some special applications.

5 Conclusion

The research work presented in this paper aimed at providing dynamic force equalization control strategies for an active/active control redundant actuation system involving one SHA and one EMA. For that, a virtual test bench was built, detailed and proved to work well on describing the measured system performances. On basis a simplified system model, the expression of the dynamic force fighting was calculated. One solution which aimed at balancing the actuators position loop dynamics was proposed as the first strategy. It involves a model reference control and forces the two actuators to track the same model outputs by compensating the actuators velocity and acceleration. Following that, two strategies were introduced taking benefits of secondary inputs representing the force fighting: one strategy uses this signal to compensate the original position controllers while the other one directly uses this additional signal to control the SHA. Finally the virtual test bench was used to simulate responses to different types of inputs. All these strategies were confirmed producing efficient dynamic force equalization. At the same time the characteristics of each strategy were studied and compared. The selections to different applications were recommended.

Now both the static and dynamic force equalizations of the hybrid redundant actuation system have been well studied, the future work will be focused on combining research results and maturing them for industrial applications as well as addressing performance robustness more widely.

6 Acknowledgment

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