THE STUDY OF AN ELECTROHYDRAULIC SERVOVALVE FOR FAULT DETECTION AND ISOLATION PURPOSES

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Abstract. *This paper presents the study of an electro-hydraulic servovalve used in aircraft's flight control systems in order to build the knowledge base necessary for the synthesis of model-based fault detection and isolation monitors. This study comprises the operation analysis and understanding of the electro-hydraulic servovalve under normal operation condition in open and closed loop through mathematical modeling and simulation. It comprises also the study of important faults which the component is subject and the analysis of how those faults affects the servovalve's dynamic response. Finally, the faults studied are modeled in a linear fashion allowing for the use of linear system theory in the synthesis of fault detection and isolation systems.*

Keywords: *Modeling and Simulation, Electrohydraulic Servovalve, Fault Analysis*

1. NOMENCLATURE

Δi	Torque motor current	e_g	Torque motor input voltage
R_c	Torque motor internal resistance	N_c	Winding turns
ϕ_a	Total magnetic flux in motor armature	g	Magnetic gap when in neutral position
x	Armature tip position in terms of neutral position	a	Armature arm length
A_g	Magnetic gap area	μ_0	Vacuum magnetic permeability
ϕ_1	Magnetic flux in gaps 1 and 3 of torque motor	ϕ_2	Magnetic flux in gaps 2 and 4 of torque motor
ϕ_c	Magnetic flux due to armature control current	ϕ_g	Magnetic flux due to permanent magnets when in neutral position
M_0	Total magnetomotive force	$\Re_g$	Neutral position magnetic gap reluctance
T_d	Motor total torque due to armature current	J_a	Armature's inertia
B_a	Torque motor viscous friction coefficient	K_a	Torque motor centering spring constant
C_{d_0}	Upstream orifice discharge coefficient	C_{d_f}	Nozzle orifice discharge coefficient
A_0	Upstream orifice area	D_N	Nozzle orifice area
ρ	Hydraulic fluid density	β	Bulk modulus
P_S	Supply pressure	P_1, P_2	Spool chamber pressures
x_{f_0}	Clearance between flapper and nozzle at neutral position	x_f	Flapper position from null point
Q_1, Q_2	Flow in and out the spool tip chambers	V_{v_0}	Volume of each spool chamber at null point
A_v	Spool tip cross section area	M_v	Spool mass
B_v	Spool viscous friction coefficient	K_f	Servovalve feedback spring constant
x_m	Feedback spring tip position without deflexion	Q_A, Q_B	Flow in an out the actuator chambers
V_{p_0}	Volume of each actuator chamber at null point	C_{i_p}	Actuator internal leakage coefficient
C_{e_p}	Actuator external leakage coefficient	x_p	Ram position in terms of null point
M_p	Actuator piston and ram mass	B_p	Piston viscous friction coefficient
e_p	Ram position feedback voltage signal	K_{LVDT}	Position sensor gain
τ_{LVDT}	Position sensor time constant		

2. INTRODUCTION

Fault detection and isolation for industrial processes and critical systems is a widely discussed matter nowadays. The search for equipment robustness and the possibility of condition based maintenance has a financial appeal. For critical systems, smaller redundancy numbers and increased reliability is what attracts specialists to the subject.

The advance in digital computing and control theory brought with it the automation of most of man made routine tasks related to industrial process operation. Still, one task that is in most cases human dependent is the process of Abnormal Event Management (AEM). The activities related to AEM comprise fault detection, identification and diagnosis of root causes, and decision upon correction and reconfiguration measures. The constant increase in systems complexity and the number of variables to be monitored brings need for the automation of this task also. The automatic fault detection and isolation science is the first step in this direction.

In critical systems, the main driver for the study of fault detection and isolation techniques is the ability for using analytical redundancy schemes. For this kind of systems, the high integrity requirement of functions is, by first option, attained with the use of physical redundancy. This kind of solution translates directly to an increase in costs, weight, volume and maintenance needs. In some of these cases, and in others where physical redundancy is impractical, is possible to use an analytical emulation of the target functionality for monitoring purposes by means of output comparison. The analytical redundancy, as the emulation is called, is based on a first principles model of the physical component that runs in parallel to it in a computing platform.

As other model-based fault detection and isolation techniques, analytical redundancy requires profound understanding of the monitored process operation. It depends not only on that but also, for proper fault isolation, also on the influence of the faults to the system. In this aspect, the monitored system knowledge directly affects a monitor performance since discrepancies between expected and observed behavior could lead to spurious detections. To minimize unwanted effects monitor requirements include robustness to operational condition variations and modeling uncertainties.

A fly-by-wire flight control system is an example of an aircraft critical system that takes advantage of fault detection and isolations schemes in order to improve dispatchability and safety, and reducing operational costs. The fly-by-wire technology is responsible for the substitution of mechanical links between pilot controls and aerodynamic surfaces to electrical connections. The use of electrical signals for commanding the aircraft attitude enables signal processing techniques to be applied such as advanced control laws, flight envelope protection and fault monitoring algorithms.

In this scenario, electrohydraulic actuators are widely used as a bridge between electrical command signal and power hydraulic surface actuation. One of the key components of this type of actuator is the electrohydraulic servovalve. As a complex component responsible for a safety critical function, the servovalve is often the center of attention in fault monitoring studies. In order to facilitate future works related to fault monitoring of the electrohydraulic servovalve it is proposed a study of this component and the faults related to it, building the profound knowledge necessary in those tasks.

3. SYSTEM MODELING

The electrohydraulic servovalve under study is shown in Fig. 1 together with a hydraulic actuator and a feedback controller. It is a four-way flow control servovalve with two stages, the first one being flapper-nozzle and the second a spool valve with four critical center lands. A feedback spring connects the spool to the flapper providing the latter with force feedback.

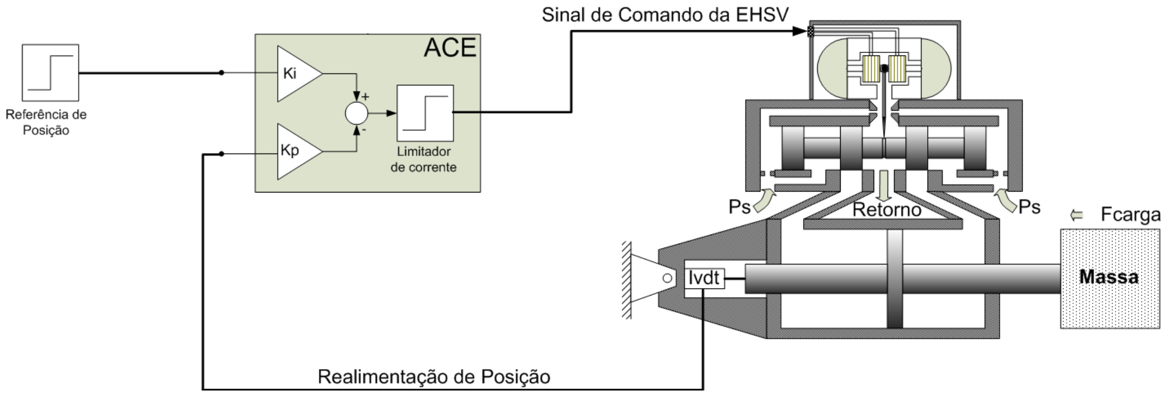

Figure 1. Diagram of the electrohydraulic servovalve under study together with a hydraulic actuator and a feedback controller.

The modeling of the servovalve is done analytically as described in (Merrit, 1967) for the servovalve components of torque motor, first and second stages. The model is made so that the physical characteristics of the component, as orifice diameters or chamber pressures, are treated uniquely and not as variation terms. This allows for individually changing parameters of the servovalve for behavioral investigations. The equations that describe the servovalve behavior are sumarized below.

$$\dot{\Delta}_i = \frac{1}{L_c}\left(\frac{-\Delta_i R_c}{2} - \omega K_b + e_g\right) \tag{1}$$

$$\dot{\theta} = \omega \tag{2}$$

$$\dot{\omega} = \frac{1}{J_a}\left(T_d - T_l - \omega B_a\right) \tag{3}$$

$$\dot{x}_v = v_v \tag{4}$$

$$\dot{v}_v = \frac{1}{M_v}\left((P_1 - P_2)A_v - B_v v_v - K_f(x_m + x_v)\right) \tag{5}$$

$$\dot{P}_1 = \left(\frac{\beta}{V_{v_0} + x_v A_v}\right)(Q_1 - A_v v_v) \tag{6}$$

$$\dot{P}_2 = \left(\frac{\beta}{V_{v_0} - x_v A_v}\right)(A_v v_v - Q_2) \tag{7}$$

For the above equations, the parameters are given by:

$x = a\tan(\theta)$	$x_m = (r+b)\tan(\theta)$
$\phi_c = N_c \frac{\Delta_i}{2R_g}$	$Q_1 = C_{d_0} A_0 \frac{P_S - P_1}{\lvert P_S - P_1\rvert}\sqrt{\frac{2}{\rho}\lvert P_S - P_1\rvert} - C_{d_f}\pi D_N(x_{f_0} - x_f)\frac{P_1}{\lvert P_1\rvert}\sqrt{\frac{2}{\rho}\lvert P_1\rvert}$
$\phi_1 = \frac{\phi_g + \phi_c}{1 - x/g}$	$Q_2 = C_{d_f}\pi D_N(x_{f_0} + x_f)\frac{P_2}{\lvert P_2\rvert}\sqrt{\frac{2}{\rho}\lvert P_2\rvert} - C_{d_0} A_0 \frac{P_S - P_2}{\lvert P_S - P_2\rvert}\sqrt{\frac{2}{\rho}\lvert P_S - P_2\rvert}$
$\phi_2 = \frac{\phi_g - \phi_c}{1 + x/g}$	$T_f = r\left(A_N(P_1 - P_2) + 4\pi C_{d_f}^2\left((x_{f_0} - x_f)^2 P_1 - (x_{f_0} + x_f)^2 P_2\right)\right)$
$T_d = a\left(\frac{\phi_1^2 - \phi_2^2}{\mu_0 A_g}\right)$	$T_l = T_f + K_a\theta + (r+b)K_f(x_m + x_v)$
$x_f = r\tan(\theta)$	

Models of a linear hydraulic actuator and feedback controller are needed for servovalve closed loop analyses. The actuator considered is of piston-cylinder type. The feedback controller excites the servovalve with voltage signals that are proportional to the difference between the actuator ram position and a reference input command. With that, the resultant actuation system model is composed of proportional controller, electrohydraulic servovalve and linear hydraulic actuator.

The model for the actuator is summarized by the following equations.

$$\dot{x}_p = v_p \tag{8}$$

$$\dot{v}_p = \frac{1}{M_p}\left(A_p(P_A - P_B) - B_p v_p - F_L\right) \tag{9}$$

$$\dot{P}_A = \left(\frac{\beta}{V_{p_0} + x_p A_p}\right)\left(Q_A - A_p v_p - C_{i_p}(P_A - P_B) - C_{e_p} P_A\right) \tag{10}$$

$$\dot{P}_B = \left(\frac{\beta}{V_{p_0} - x_p A_p}\right)\left(A_p v_p - Q_B + C_{i_p}(P_A - P_B) - C_{e_p} P_B\right) \tag{11}$$

$$\dot{e}_p = \frac{1}{\tau_{LVDT}}\left(K_{LVDT} x_p - e_p\right) \tag{12}$$

The fluid flow to the actuator controlled by the servovalve is given by:

$$\begin{aligned}
&\text{If} \quad x_v > 0 \begin{cases} Q_A = \frac{P_S - P_A}{\lvert P_S - P_A\rvert} C_d x_v w\sqrt{\frac{2}{\rho}\lvert P_S - P_A\rvert} \\ Q_B = \frac{P_B}{\lvert P_B\rvert} C_d x_v w\sqrt{\frac{2}{\rho}\lvert P_B\rvert} \end{cases} \\
&\text{If} \quad x_v < 0 \begin{cases} Q_A = \frac{P_A}{\lvert P_A\rvert} C_d x_v w\sqrt{\frac{2}{\rho}\lvert P_A\rvert} \\ Q_B = \frac{P_S - P_B}{\lvert P_S - P_B\rvert} C_d x_v w\sqrt{\frac{2}{\rho}\lvert P_S - P_B\rvert} \end{cases} \\
&\text{If} \quad x_v = 0 \begin{cases} Q_A = 0 \\ Q_B = 0 \end{cases}
\end{aligned} \tag{13}$$

The equation for the proportional controller is given by:

$$e_g = K_i V_{ref} - K_p e_p \tag{14}$$

where V_{ref} is a voltage position reference signal in the controller input.

4. SYSTEM SIMULATION

For system simulation, the analytical equations are implemented in a MATLAB script. The servovalve parameters are obtained from different sources ((Arafa and Rizk, 1987), (Thayer, 1965), (Jelali and Kroll, 2003)) forming a fictitious component, nevertheless the fictitious servovalve natural frequency is representative to the ones of MOOG series 31 (Thayer, 1965). Figure 2 shows the servovalve open loop response to a voltage step signal with amplitude corresponding to a torque motor steady state current of 5 mA.

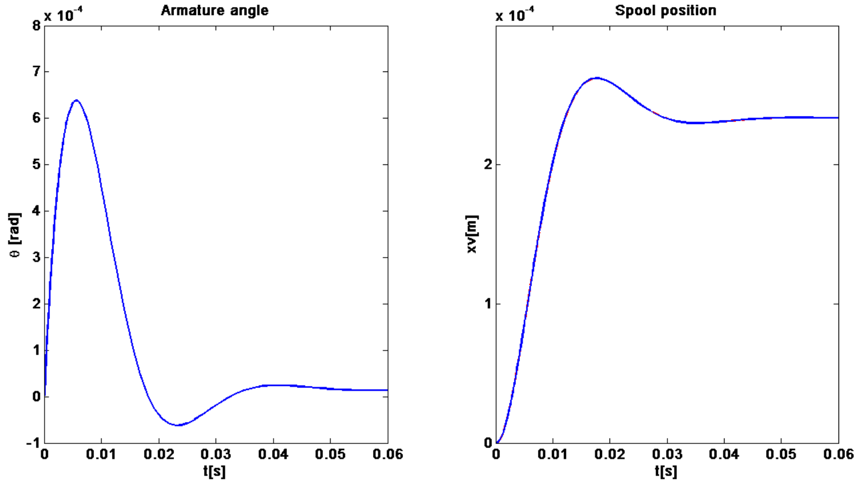

Figure 2. Open loop step response of the servovalve.

Figure 3 shows the closed loop step response of the servovalve combined to the hydraulic actuator and proportional controller.

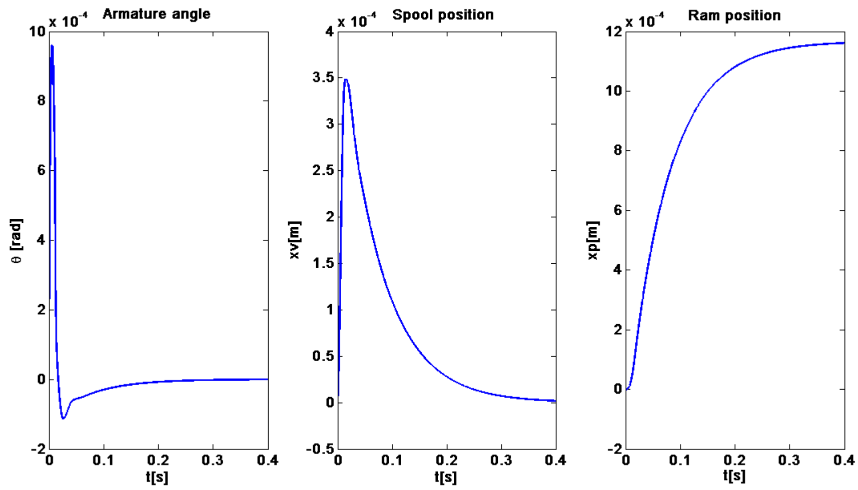

Figure 3. Closed loop step response of the servovalve and hydraulic actuator.

For a survey of the servovalve normal operation envelope in terms of temperature and pressure it is considered the changes of fluid physical characteristics in respect to those values. The fluid physical characteristics are parameters used in the servovalve non-linear model so they can be easily altered for simulation purposes. The characteristics of the fluids used, MIL-H-5606, are briefly described in (Merrit, 1967) and (Blackburn, Reethof, and Shearer, 1960).

By constructing a grid of test-cases with temperature ranging from 0 to 100 Celsius degrees and pressure from 500 to 3000 psi it is possible to simulate the servovalve response in all conditions inside that grid. With the frequency response of the system in all test-cases the maximum and minimum values of phase and magnitude for each frequency value can be plotted, what results in the envelope shown in Fig. 4.

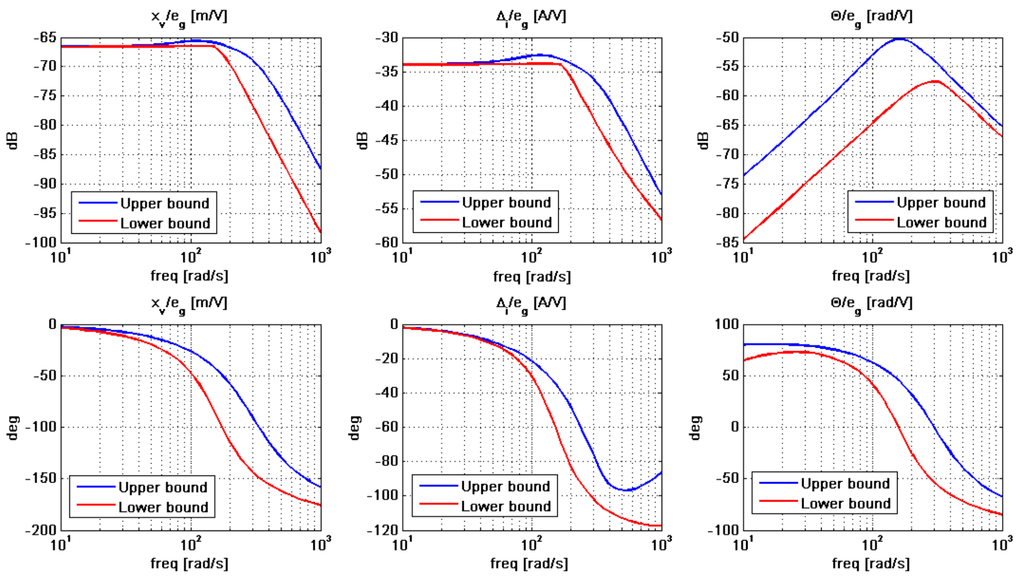

Figure 4. Frequency response operation envelope for the servovalve.

5. FAULT STUDY

For the selection of the faults to be analyzed and monitored, some characteristics have to be taken into account, such as:

- The observability of the phenomena, that is, if it is possible to infer the parameters that have changed due to a fault from available measurements and how subtle is the impact on these measurements.
- The probability of occurrence in a normal operation environment.
- The impact of the fault in the equipment and in its ability to perform its required function.
- The impact of loosing the equipment functionality in terms of failure propagation.
- Dispatchability matters of the aircraft that the component is part of.

In order to conclude on a fault probability and its safety impacts methods of Failure Mode and Effect Analysis (FMEA) and Failure Mode and Criticality Analysis (FMECA) are commonly used. Such methods aim for the analysis of a target system in respect to its associated functions, faults, fault effects, recommended actions and other relevant issues. Those types of information are of great importance and are required for the certification of critical systems.

With an analysis of this kind for the servovalve it is possible to identify the most relevant faults to be studied. Table 1 shows this result omitting failure modes related to sensors that, although very common, would require a dedicated study.

5.1 Fault Model Dynamic Response

The faults that will be analyzed are:

1. Nozzle orifice clogging.
2. Braking of the feedback spring.

These faults are mechanical related and are classified, regarding their degradation characteristics, as gradual and abrupt respectively. They are inserted in the model as a variation in the parameters that relate to them.

The simulation of the system is made for each fault condition in two steps. The first is the analysis of open loop response without considering external influences such as fluid temperature and pressure variations. For this simulation the servovalve is excited with a step voltage signal in amplitude needed for a steady state current of 5 mA. The system response is given in terms of spool position in meters and flapper deflection in radians.

The second analysis is the response of the system when in closed loop, in set with an hydraulic actuator, feedback sensors and loop closure electrical device. In this simulation, the input of the set is excited with a step voltage signal of 0.5 mV. No external forces are applied to the actuator and the condition of the hydraulic fluid is kept constant.

Table 1. Electrohydraulic servovalve faults and failure modes considered relevant for the study.

EFFECTS	CRITICALITY	FAILURE RATE	FAULT	DISPATCHABILITY IMPACT
Reduced servovalve flow rate or gain change	Minor	From 1E-6 to 1E-7 per flight hour	First and second stage failures	Yes
			Torque motor armature blocked	
			Feedback spring deformed	
			Upstream orifice clogging	
			Slide binding	
			Slide/sleeve erosion	
Null shift	Minor	From 1E-7 to 1E-8 per flight hour	Upstream orifice clogging	Yes
			Nozzle orifice clogging	
			Feedback spring deformed	
Servovalve uncontrollable (potencial oscillation)	Minor	From 1E-7 to 1E-8 per flight hour	Armature spring broken	Yes
			Feedback spring broken	
			Flapper broken	

5.1.1 Effect of nozzle orifice clogging

A nozzle orifice clogging causes a change in the pressure drop across the upstream orifice due to altered fluid flow. It also alters the influence of the nozzle fluid stream over the flapper and vice versa. These influences are simulated by the reduction of one nozzle orifice diameter D_N in steps of 10% its nominal value.

Figure 5 shows the servovalve open loop response in terms of spool position and flapper angle. As it is shown, the spool reaches its hard stop, with the considered input signal, for clogging starting from 40% of nominal diameter reduction. A nozzle orifice clogging has no or little influence to the system's dynamic characteristics, however it alters significantly the servovalve point of null.

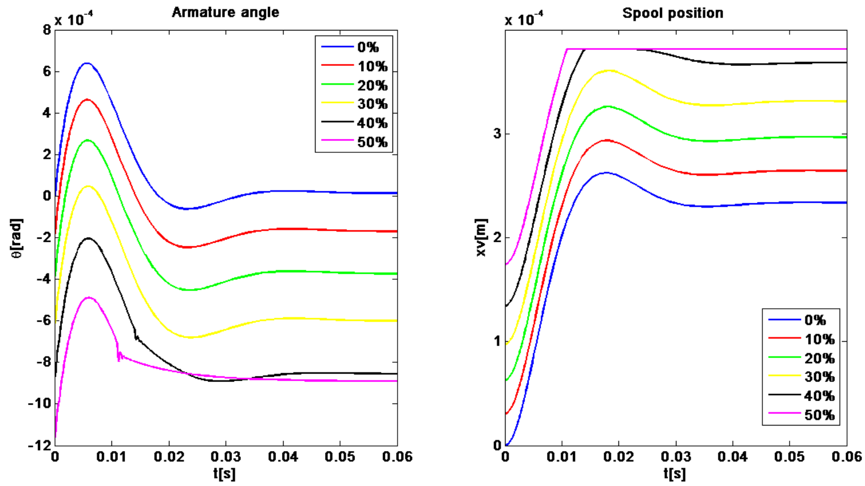

Figure 5. Open loop servovalve response with nozzle orifice clogging. The nozzle orifice diameter is reduced in steps of 10% its nominal value.

When in closed loop, given the characteristic of the controller used for loop closure, the steady state error of the actuator position is not compensated, which leads to a variation in the point of null of the system due to the fault. As shown in Fig. 6, the changes in the system dynamic are hardly visible if not considered the cases when the spool position is hard limited.

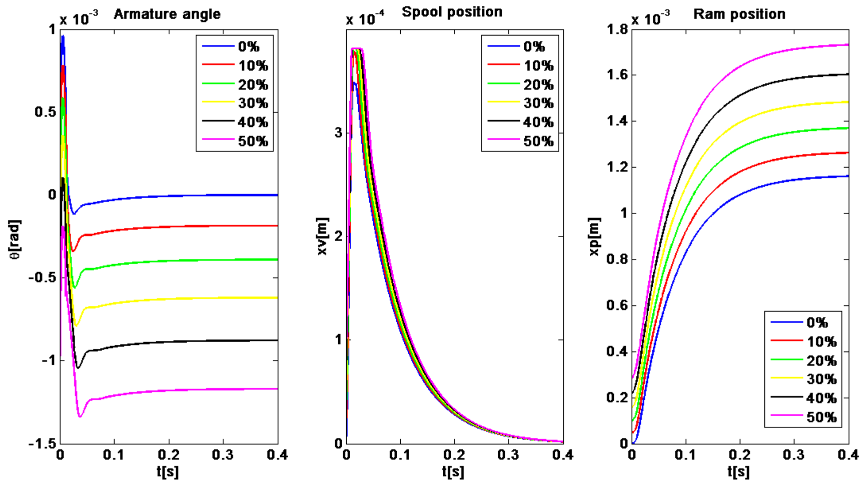

Figure 6. Closed loop response of the servovalve with nozzle orifice clogging. The nozzle orifice diameter is reduced in steps of 10% its nominal value.

5.1.2 Effect of braking the feedback spring

The rupture of the feedback spring is an abrupt fault that might lead to a system instability or a degraded performance operation depending on the control system used for loop closure. This fault is one of the causes of servovalve hard-overs, where the spool goes directly to one of its hard-stops with no chance of position control. This kind of problem is one that a critical system must be able to detect and to manage either by a control law reconfiguration or by a control authority disengagement.

The open loop response of the system without the feedback spring is shown in Fig. 7. In this scenario the position feedback from the spool to the flapper is inexistent and the only coupling between this two elements is through the spool speed and the associated fluid flow. As the spool hits its hard-stop there is an abrupt variation in fluid flow to the spool chambers which affects the flow through the nozzles.

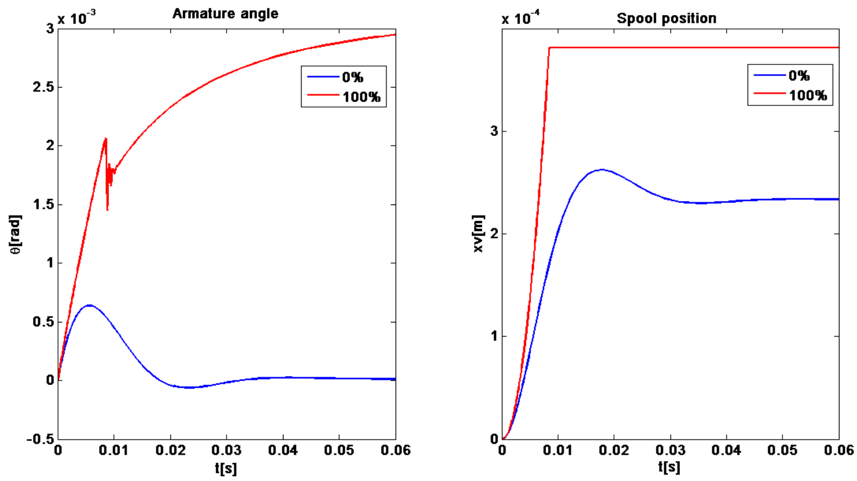

Figure 7. Open loop response of the servovalve with total loss of feedback spring.

In case of closed loop operation, the flapper and the spool valve oscillates between their respective hard-stops, resulting in an oscillation of the actuator nearby its commanded position (Fig. 8).

The maximum amplitude allowed for an oscillatory response of a flight control actuator is specified according to the dynamic pressure to which an aerodynamic surface is subject. Oscillation amplitudes beyond specified limits could lead to structural damages due to cyclic loads or flutter phenomenon.

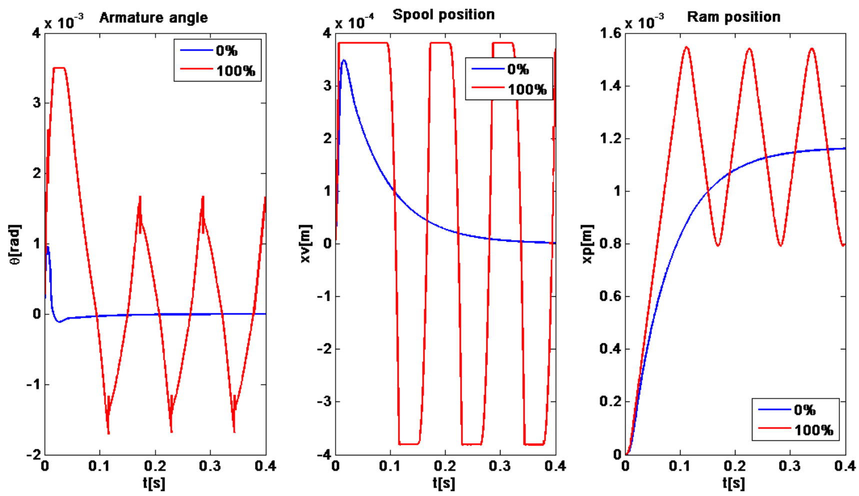

Figure 8. Closed loop response of actuator and servovalve with total loss of feedback spring.

5.2 Linear fault modeling

The use of linear systems theory methods for the design of fault detection and isolation systems depends on a linear fault modeling in an additive or multiplicative fashion: this is, modeled as additional inputs to a system or as changes to system parameters and its dynamic behavior (García, Seliger and Frank, 1998). In general, for fault detection, the dynamic model of the plant to be monitored is enough if not considered the effects of modeling uncertainties and external signal or disturbances. The fault model knowledge is important in the process of fault isolation in order to design filter banks that are sensitive to one fault and immune to others, state observers that generate different direction residual vectors or any other scheme able to distinguish between different kinds of anomalous influences acting over a system.

5.2.1 Multiplicative fault model

It is considered that a non linear system, as the electrohydraulic servovalve, is described in the form of Eq. 15.

$$\begin{aligned} \dot{\mathbf{x}}(t) &= F(\mathbf{x}(t), \mathbf{u}(t), \theta) \\ \mathbf{y}(t) &= \mathbf{C}\mathbf{x}(t) \end{aligned} \tag{15}$$

In this equation $\mathbf{x}$ is the system states vector, $\mathbf{u}$ the system inputs vector and θ the system parameters at a given condition. $\mathbf{C}$ is a matrix that relates the system states to its outputs. By linearizing the non-linear relations of the model it is possible to obtain a linear system in the state space format that approximates the behavior of the Eq. 15 nearby an operating point. If considered variations to the parameters θ in that linear system, one could describe these parameter changes as a change in the state matrix $\mathbf{A}$ as shown below:

$$\begin{aligned} \dot{\mathbf{x}} &= (\mathbf{A} + \mathbf{\Delta A})\,\mathbf{x} + \mathbf{B}\mathbf{u} \\ \mathbf{y} &= \mathbf{C}\mathbf{x} \end{aligned} \tag{16}$$

where,

$$\mathbf{\Delta A} = \left. \frac{\partial^2 F\left(\mathbf{x}, \mathbf{u}, \theta\right)}{\partial \mathbf{x} \partial \theta} \right|_{\theta=\theta_0} \mathbf{\Delta}\theta \tag{17}$$

The parameter vector variation $\mathbf{\Delta}\theta$ is considered as being due to a time dependent faulty behavior of the system (Eq. 18).

$$\mathbf{\Delta}\theta \equiv \mathbf{f}(t) \tag{18}$$

An example of fault with multiplicative behavior is the feedback spring rupture. This conclusion is taken from the non-linear fault simulation where a change in the dynamic characteristics of the servovalve was made apparent.

The linear fault model is obtained by the application of Eq. 18 to the non-linear model of the servovalve. Figure 9 shows the comparison of the linear and non-linear faulty system response. The divergence in the responses is due to the saturation behavior, not present in the linear model. The responses similarities, despite that, are apparent.

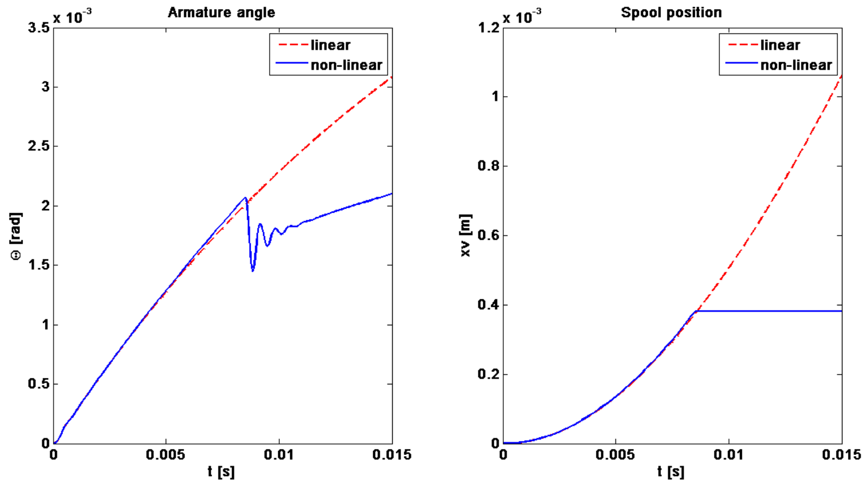

Figure 9. Open loop response of the servovalve with total loss of feedback spring. Comparison of the non-linear model to the linear one.

5.2.2 Additive fault model

In the additive fault model the fault is described as an additional input to the linear system obtained from Eq. 15. In this case, as shown in Eq. 20, the fault signal $f(t)$ influences the system state derivatives by multiplying an input vector E_f.

$$\dot{\mathbf{x}} = \mathbf{A}\mathbf{x} + \mathbf{B}\mathbf{u} + \mathbf{E_f}\mathbf{f} \tag{19}$$

$$\mathbf{y} = \mathbf{C}\mathbf{x} \tag{20}$$

$\mathbf{E_f}$ is what determines the actual fault effects on the system and it is obtained as presented below.

$$\mathbf{E_f} = \left.\frac{\partial F(\mathbf{x}, \mathbf{u}, \theta)}{\partial \theta}\right|_{\theta=\theta_0} \tag{21}$$

where θ_0 is the nominal parameters vector of the system in normal operation.

From the studied faults, the nozzle orifice clogging is a fault that clearly presents additive characteristics: changes in the system null point and no visible effects in the dynamics. Figure 10 compares the responses from the linear and non-linear faulty model of the servovalve. These responses are obtained when one of the nozzle orifice has its diameter reduced to 80% its nominal value.

6. FINAL COMMENTS

It was shown, for an electrohydraulic servovalve, how to obtain through detailed analytical modeling and simulation some of the knowledge needed for the design of model-based fault detection and isolation monitors.

In the analytical modeling, the importance of having all physical terms available in the equations raised the possibility for changing individual parameters. With this capability one could easily insert fault behaviors to the models by altering the related terms.

The frequency response envelope of the component to be monitored was also obtained once the characteristics of the fluid in regard to temperature and pressure was known. This envelope is used as robustness requirement to fault monitoring schemes since a normal system response variation cannot be interpreted as a faulty behavior.

The FMEA and FMECA were mentioned as important tools in the definition of which faults were to be monitored. Once defined, the faults were simulated for characteristics survey and model-based validation of fault impacts on the overall system.

An easy method of linear fault modeling was presented. This kind of modeling is useful for fault isolation purposes where one should know the aspects of each fault and how they differ from others.

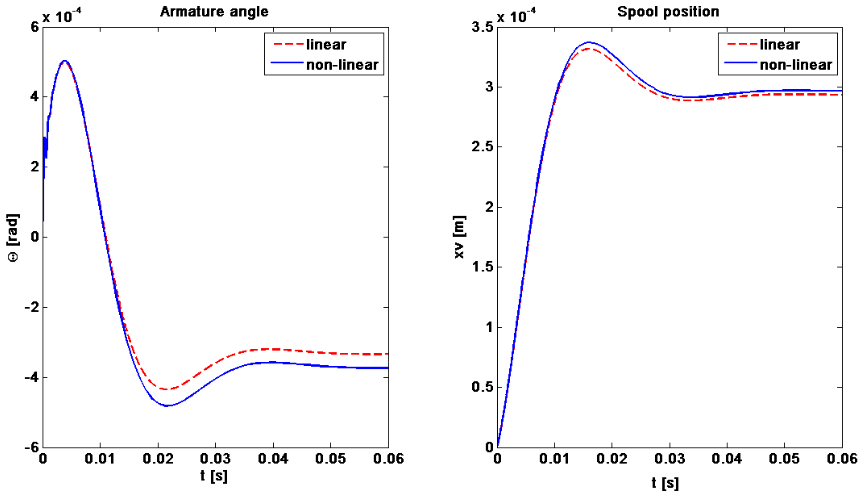

Figure 10. Open loop response of the servovalve with one nozzle orifice diameter reduced of 20% of its nominal value. Comparison of the non-linear model to the linear one.

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