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ON THE EFFECTS OF MONITORING SYSTEM ON REDUNDANT HYDRAULIC FLIGHT CONTROLS IN FAILURE CONDITIONS

Matteo D. L. Dalla Vedova

Department of Mechanical and Aerospace Engineering, Politecnico di Torino

ABSTRACT

An active monitoring system is generally required to detect failures of the on-board hydraulic system and overcome its incorrect behaviour. Indeed, several kinds of failures can affect the flight control systems, and sometimes they can be safety-critical. This work is focused on electrohydraulic actuators equipped with proper equalization devices acting when failure modes are present. It studies the most commonly used architecture of redundancy based on the torque-sum arrangement (hydraulic motors generate torques that are summed within the gear reducer connecting the power drive unit with the motion transmission). To simulate and evaluate different behaviours of the system affected by various failure mode conditions, the author developed a dedicated computer program based on a physical-mathematical model of the whole actuation system. It is equipped with a hydraulic motor, electro-hydraulic servovalve, position feedback, and equalization control law. The author pointed out the limits of each kind of failure linked to the related equalization device and concluded that, in case of failure, the disengagement of the equalization device allows a correct actuation operation, thus preventing all critical situations. Equalization devices represent an obstacle because of the reduced contrasting action performed by the operational valve against the failed one.

Keywords: Dynamic simulation, Equalization devices, Failure, Flap controls, Redundancy

1 INTRODUCTION AND AIMS OF THE WORK

Several types of failures, according to their specific layout, may affect the flight control systems and be critical for the safety. In order to avoid critical operation conditions and to let the system operate with one or more failure (with partial or none loss of performance), the flight controls must be conceived with a redundant design, even if, in certain cases, it's not sufficient to face problems created by a failure. Failures can be passive or active. Active failures don't affect the operability of the remaining portion of the servo-mechanism so they must be detected by an appropriate monitoring device able to shutoff the failed hydraulic unit; on the contrary, passive failures affect the operability of the remaining portion of the servo-mechanism and do not need specific corrective actions [1].

To correct improper behaviors caused by the failed part of the hydraulic system, it's necessary an active monitoring system: in fact, through proper devices, it can perform corrective actions to disengage the failed section of the system (without or with minor degradation in the whole system behavior) after its recognition in a sufficiently brief time [2]. It should be noted that the redundancy can provoke some improper behaviors in case of manufacturing defects without any failures. This is due to an unsuitable system design of the redundancy architecture, usually split into two categories: torque-summed and speed summed systems [3]. The present work takes into account a typical electrohydraulic torque summed redundant actuation system represented by a servomechanism in which the servovalve is generally considered the most critical element, because of its complex architecture [4]. A proper equalization device can generally overcome the incorrect behavior, usually a force fighting (speed fighting for speed-summed systems), caused by different offsets or asymmetries between the two sections of the actuation system [5]. In fact, manufacturing imperfections bring consequences considered as "null position servovalve errors" and are generally modelled as an offset current affecting the input of the servovalve coils.

This work takes into account the answer of a servo-mechanism equipped with proper equalization devices acting when failure modes are present and studies the most commonly used architecture of redundancy based on the torque-sum arrangement (hydraulic motors generate torques that are summed within the gear reducer connecting the power drive unit with the motion transmission).
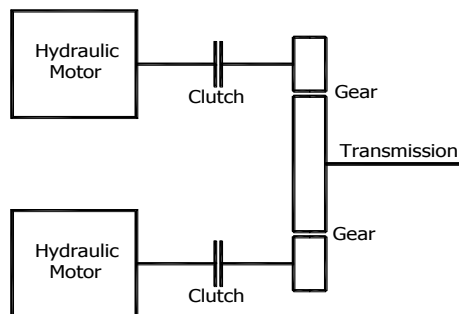

Figure 1 Schematic of the torque-sum arrangement.

Failure modes can generally be divided in two categories:
- null position failure (inability to perform and maintain any spool displacement from the centred position);
- hard-over failure (in which the spool reaches one of its ends of travel as a consequence of a control lack).

As reported in [5], the main servovalve (SV) failure modes considered in this work depend fundamentally on the valve arrangement i.e. flapper-nozzle (FN) or jet-pipe (JP):
1. torque-motor coil interruption (FN and JP);
2. first stage filter contamination (FN and JP);
3. jet pipe contamination (JP);
4. spool seizure in centered position (FN and JP);
5. electrical connections disengagement (FN and JP);
6. fixed orifices or nozzles contamination (FN);
7. spool seizure in maximum displacement (FN and JP);
8. electrical amplifier polarization (FN and JP);

The failure modes above listed as numbers 1 to 5 give rise to null position failures because their consequence is the inability to perform and maintain any spool displacement from the centered position; on the contrary, the failure modes above listed as numbers 6 to 8 give rise to hard-over failure (i.e. the spool reaches one of its ends of travel as a consequence of a control lack). In case of failure transient, the equalization device and the redundancy architecture greatly affect the system behaviours. In this work two kinds of equalization devices are considered: one aims to reduce the difference between the spool displacements of the servovalves itself, the other aims to reduce the difference between the differential pressure produced by each servovalve on its coupled hydraulic motor. At the aim to simulate and evaluate different behaviours of the system affected by various failure mode conditions, the authors developed a computer program based on a physical-mathematical model of the servomechanism, equipped with hydraulic motor, electro-hydraulic servovalve, position feedback and equalization control law.

2 PHYSICAL-MATHEMATICAL MODELS OF THE CONSIDERED ACTUATION SYSTEM

In order to evaluate the effects of the failure modes, a physical-mathematical model of the servo-mechanism has been developed, equipped with electrohydraulic servovalve, hydraulic motor, position feedback and equalization control law; from this a computer program has been obtained, by means of which several simulations of the system behavior under different failure mode conditions have been performed. Figures 1 and 2 represent the block diagrams employed for the servo-mechanism representation in the abovementioned configurations:
- torque sum, differential pressure equalization (Fig. 2);
- torque sum, spool displacement equalization (Fig. 3).

As figures show, in every system layout the "error" (Err) arising from the comparison between the commanded (Com) and actual (ThS) positions is computed by means of a proportional position control law (GC) equipped with angular velocity feedback loop (GS) in order to obtain the drive current (Cor - saturated CorM) in input to each of the two servovalves; all its manufacturing defects are contained in the "Offset" quantity summed to the drive current and to the equalization current (DeCor) arising from the equalization device.

From the resulting current, by means of a model characterized by an instantaneous dynamics (GV), the positions of the valves spools (XS – end of travel XSM) are computed [6]. From these, by means of the valve pressure gain (defined GP and properly corrected to take into account the effects of the saturation on the differential pressure GPS), the differential pressures (DePM) effectively acting on the motors are obtained (taking into account the pressure losses caused, within the valve passageways, by the flows through the hydraulic motors QM and related to the valve flow gain GQ) [7].

The assumption of the constant dynamics represented by GV is sufficiently realistic because the incorrect behaviours caused by the manufacturing defects are characterized by slow dynamics [8]. Therefore the use of a more detailed model of the servovalve (characterized by high dynamics) produces no further significant results. These results have been obtained by comparison with a high fidelity (HF) digital twin of the SV fluid dynamic behaviours, accounting for the actual pressure drops caused by the restricted passages of the valve. This HF model was implemented in Matlab-Simulink and validated in turn by comparison with certified numerical codes and experimental data [9-17].

In the considered torque sum case, the differential pressures, by means of the motor displacements (Cil) and the total moment of inertia of the surface-motors assembly (2 JM), taking into account the total load (TR), the viscous (coefficient CM) and dry friction torques [18], give the assembly acceleration (D2ThM); its integration gives the speed (DThM), affecting the viscous and dry frictions and the motor working flows. The last, summed to the leakage ones, give the above mentioned pressure losses through the valve passageways.
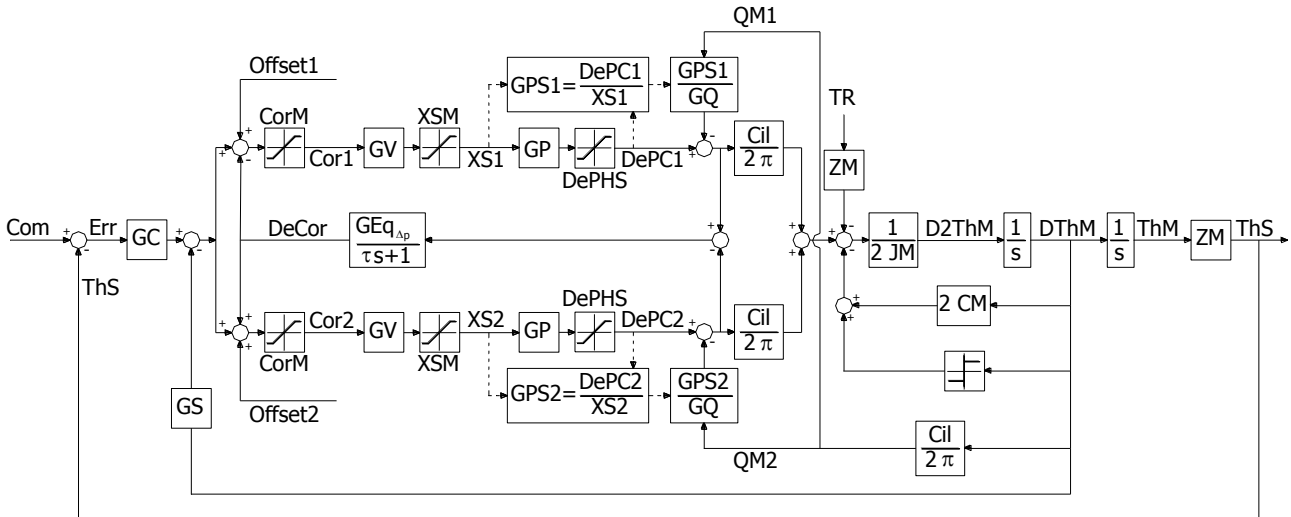

Figure 2 Block diagram of the torque summed system with differential pressure equalization.
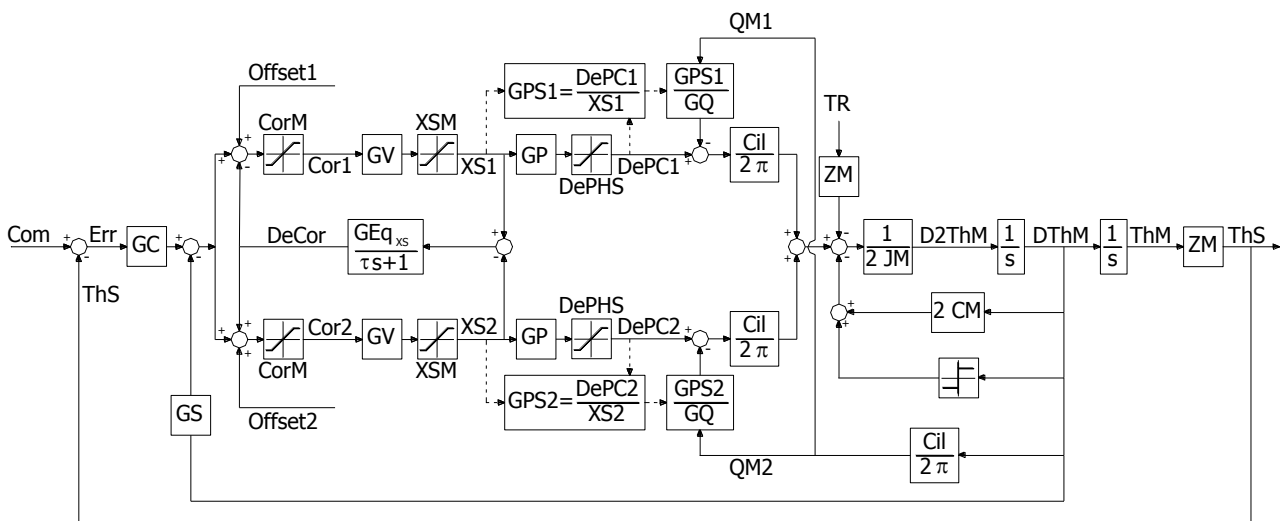

Figure 3 Block diagram of the torque summed system with spool displacement equalization.

The speed integration gives the actual motor position (ThM) and, by the gear ratio (ZM), the position of the moving surface (ThS - controlled element) which is reported in feedback on the command comparison element.

The present work is not focused on the optimization of the design parameters; component dataset is derived from the parameters set by important specialized firms. Moreover, the authors used parametrical analysis to set equalization laws and values of the gains used in the control. Parametrical analyses are more appropriate when the simulation model involves specific mechanical non-linearities as dry friction, end of travel, etc. For each servovalve, the monitoring system uses a device that matches values of input current and of spool displacement. When the two values are not correctly related each other for more than a due lapse of time, a failure declaration is performed and then the failed subsystem is disconnected.

3 SYSTEM BEHAVIOUR ANALYSIS

In order to evaluate system behaviours in different configurations (differential pressure equalization and spool displacement equalization) and operating conditions, the authors elaborated two dynamic simulation programs on the basis of the above mentioned mathematical models. Equalization control laws develop a supplementary current proportional to the difference between the two differential pressures (differential pressure equalization) or the two spool displacements (spool displacement equalization), as already shown in figures 2 and 3 respectively.

Figures 4-12 are referred to a system having an input step command Com raising from 0 to 0.1 rad. In all considered cases, the failure event occurs at time t = 0.02 s and regards only the subsystem 1, retaining the full operability of the subsystem 2. The effects of different failure conditions and their interaction with the equalization modes are evaluated
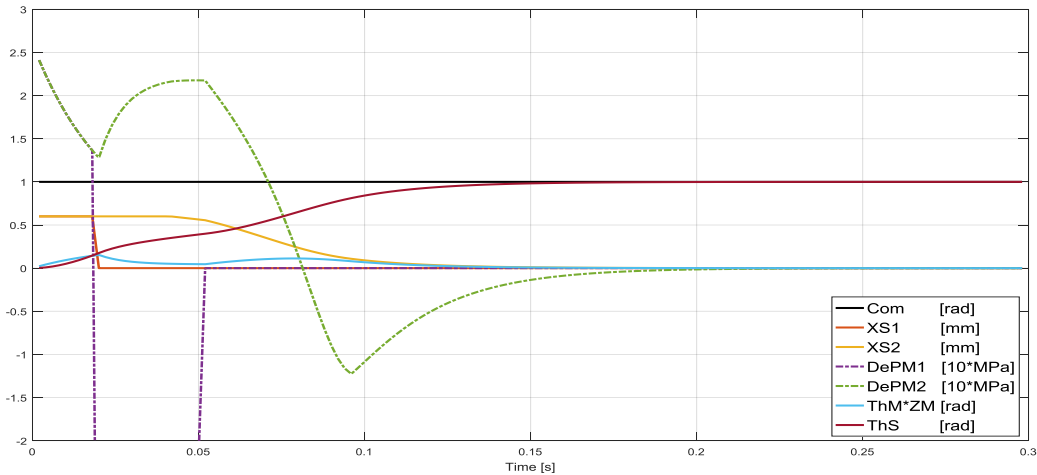

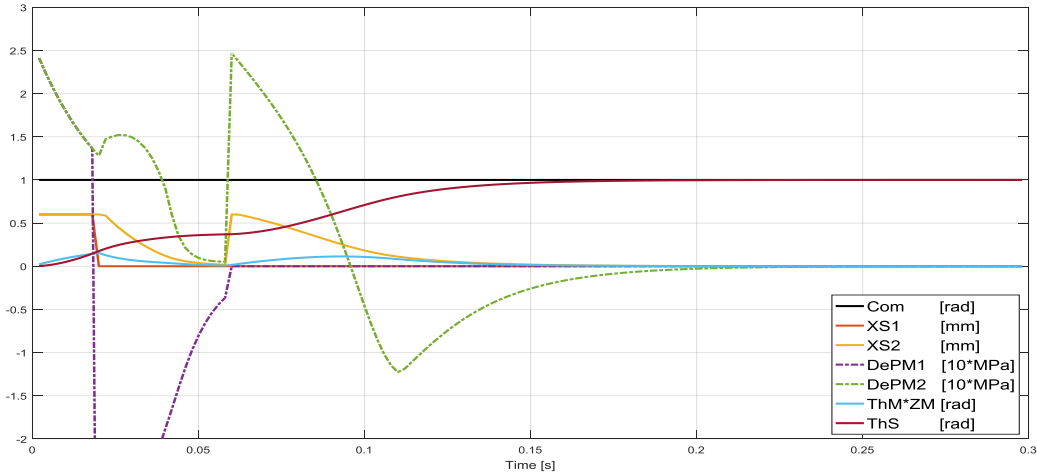Figure 4 System affected by a null position failure without any type of equalization device.

Figure 5 System affected by a null position failure equipped with a differential pressure equalization device.
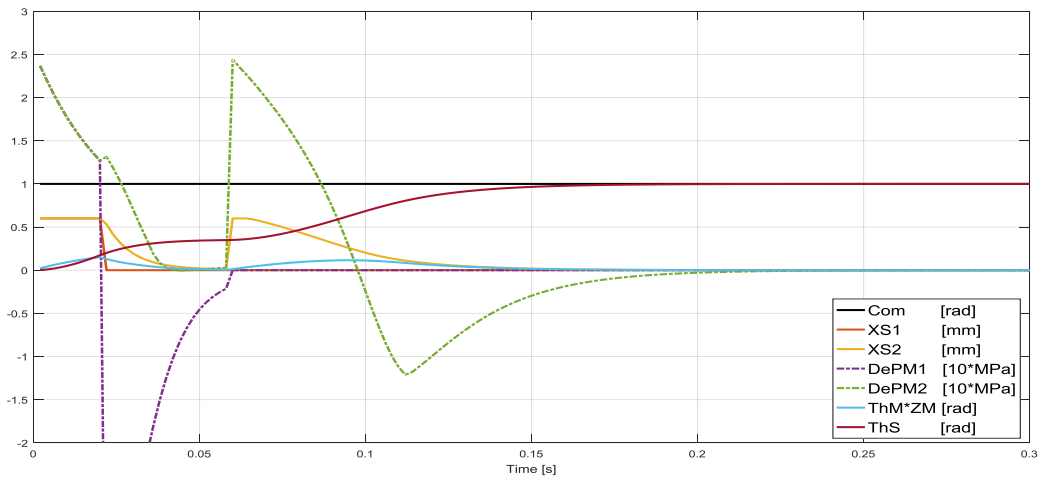
Figure 6 System affected by a null position failure equipped with a spool position equalization device.
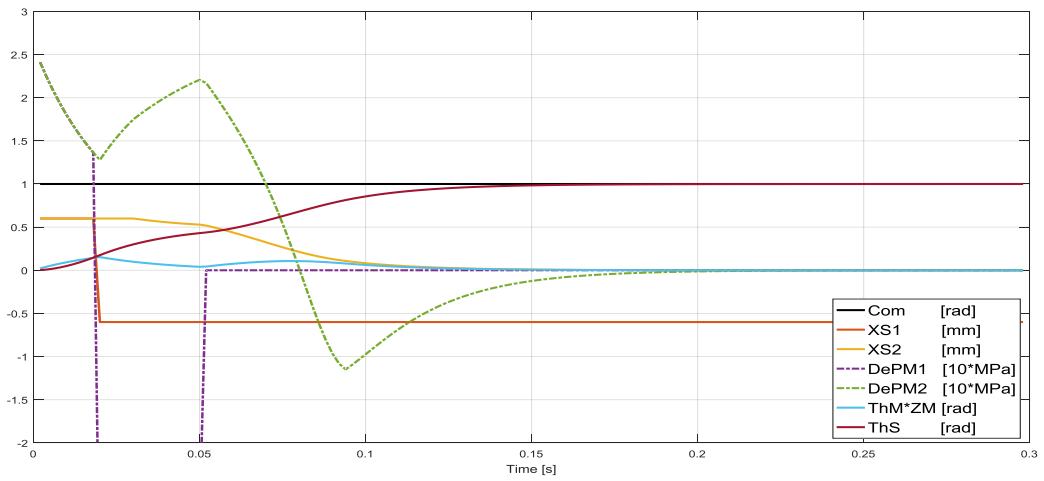

Figure 7 System affected by a negative hard over failure without any type of equalization device.
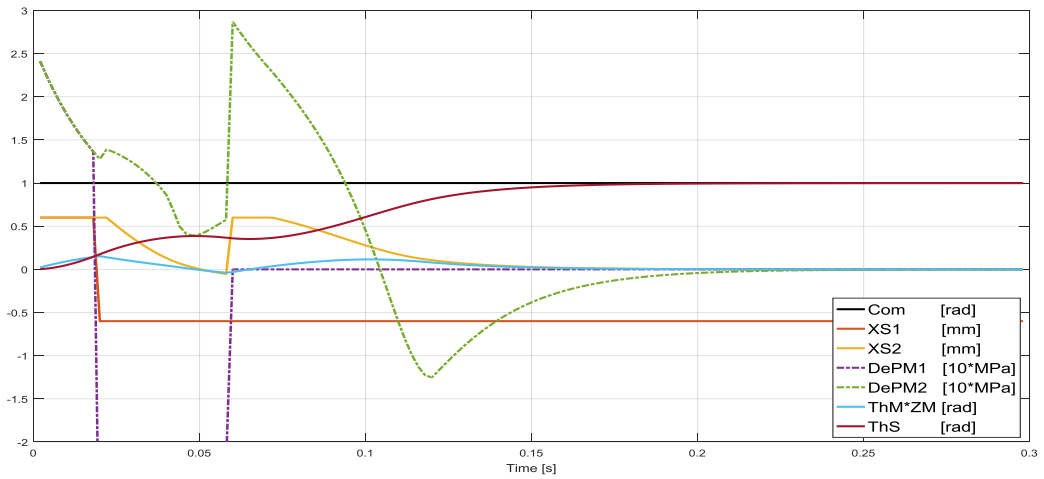

Figure 8 System affected by a negative hard over failure equipped with a spool position equalization device.
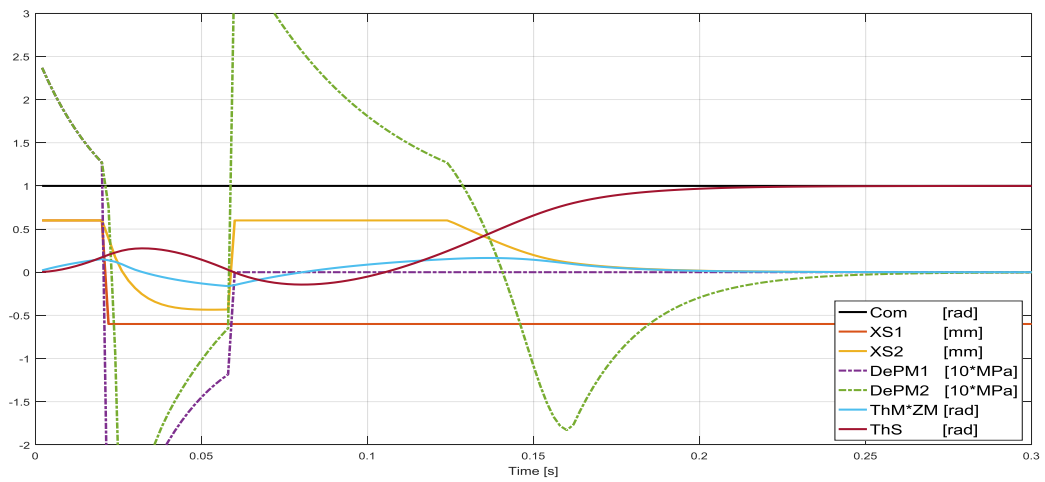
Figure 9 System affected by a negative hard over failure equipped with a differential pressure equalization device.
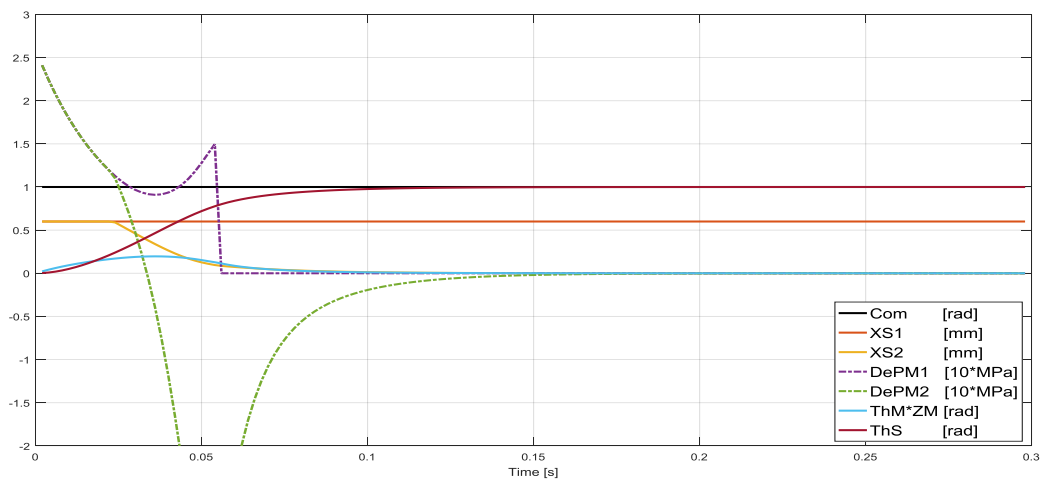

Figure 10 System affected by a positive hard over failure without any type of equalization device.
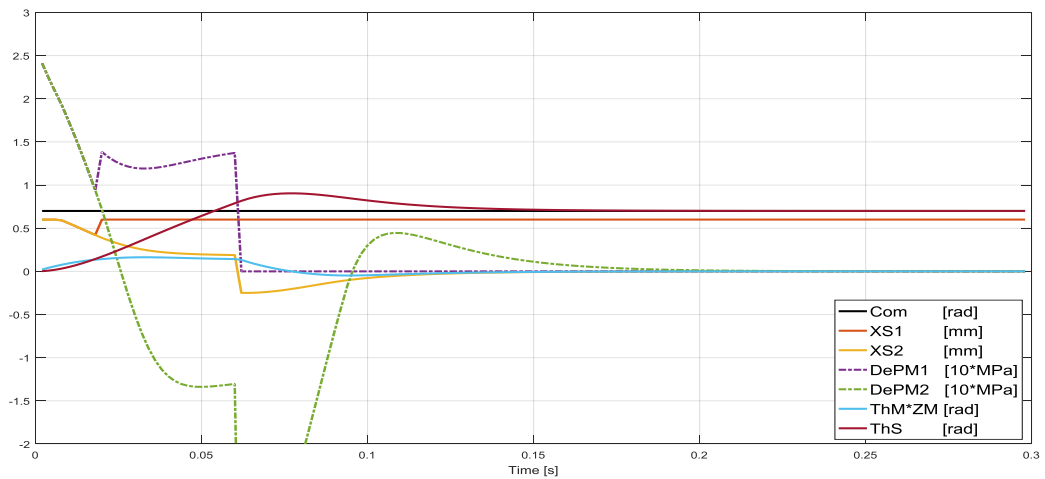

Figure 11 System affected by a positive hard over failure equipped with a spool position equalization device.
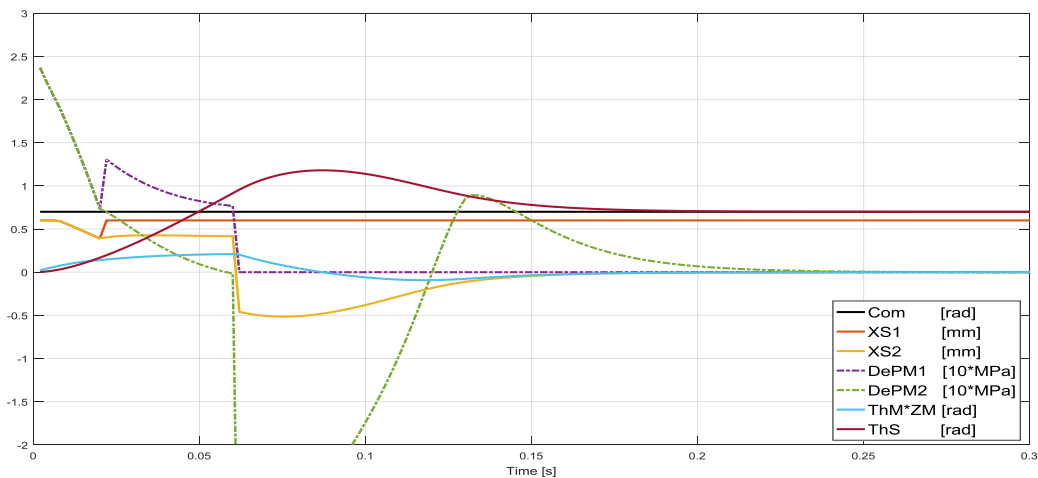
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Figure 12 System affected by a positive hard over failure equipped with a differential pressure equalization device.

Figures 4-5-6 are referred to a system affected by a null position failure without any type of equalization device (Fig. 4), with a differential pressure (Fig. 5) or spool position (Fig. 6) equalization device.

Figure 4 shows a sound damping action of the valve of the subsystem 1 that, in case of failure, suddenly returns to the centered position. On the contrary, in the subsystem 2, the reduced actuation rate causes an increased differential pressure of the valve; the equalization device is not present as proves the spool position of the operational servovalve that is not reduced by the corrective action. Therefore, the monitoring system declares the failure at time t = 0.059 s: as a consequence, the failed subsystem and the equalization device are disconnected and the fluid dynamic action produced by the failed servovalve and associated motor becomes null. After the disconnection, the actuation rate returns nearly to its original value until the reaching of the commanded position. In case of failure, the presence of the equalization system results disadvantageous: in fact its absence diminishes the total actuation time.

Figure 5 shows the effects of the differential pressure equalization device with respect to the case shown in Fig. 4. The equalization device decreases the spool displacement of the operational valve in order to shrink the difference between the differential pressures; as a consequence, at the beginning, its action significantly reduces the actuation rate. The actuation rate can be correctly guaranteed until the commanded position because, as soon as the monitoring system disengages the failed subsystem and the equalization device, the operational subsystem can reach an actuation rate similar to the full operative condition and therefore achieve its almost full performance.

Figure 6 represents the same system characterized by a spool position equalization device. The equalization device performs the same reduced actuation rate as in Fig. 4, but the corrective action, based on the differential spool position, leads to a faster transient following the failure.

Figures 7-8-9 are referred to a system affected by a negative hard over failure without any type of equalization device (Fig. 7), with a differential pressure (Fig. 8) or spool position (Fig. 9) equalization device.

The behaviour of the system affected by negative hard over without equalization (Fig. 7) is similar to the one with null position failure without equalization (Fig. 4); however, in case of failure, it occurs a torque contrasting the required actuation rate produced by the opposite displacement of the failed valve together with a damping action. As already reported for the case of Fig. 4, the full system operability and the correct achievement of the commanded position are guaranteed by the disengagement (at time t= 0.0585 s) of both the failed subsystem and the equalization device by the monitoring device.

The behaviour of the actuation system affected by negative hard over and equipped with the said differential pressure equalization device (Fig. 8) is similar to the dynamic response analysed in the previous case (shown in Fig. 7). The difference relies on the transient between failure event and disengagement because both the negative hard over condition of the failed valve and the incorrect action of the equalization device cause a temporary backward movement of the actuation system.

The system affected by negative hard over with spool displacement equalization device (Fig. 9) shows behaviour similar to the previous case, with a longer backward movement.

Figures 10-11-12 are finally referred to a system affected by a positive hard-over failure without any type of equalization device (Fig. 10), with a differential pressure (Fig. 11) or spool position (Fig. 12) equalization device.

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The system affected by positive hard over without equalization (Fig. 10) acts in a similar way as the one affected by negative hard-over without equalization (Fig. 7), but it occurs a high actuation rate because the failed subsystem performs a torque aiding (not opposing) the commanded movement.

The behaviour of the system affected by positive hard-over with differential pressure equalization device (Fig.11) is similar to the system with negative hard over with differential pressure equalization device (Fig. 8) and the behaviour of the system affected by positive hard over with spool displacement equalization device (Fig. 12) is similar to the system with negative hard over with spool displacement equalization device (Fig. 9).

Anyhow, it occurs a higher actuation rate and a marked overshoot because the failed subsystem performs a torque aiding (not opposing) the commanded movement.

4 CONCLUSIONS

The results put in evidence the criticality of each type of failure related to the corresponding equalization device. Indeed, as shown in the considered cases, in case of failure, the presence of the equalization device worsens the scenario. In fact, in this case, the operational valve (affected by equalization effects) reduces the contrasting action performed against the failed one. The proposed results point out the criticality of each type of failure concerning the corresponding equalization device. In conclusion, the author's opinion is that a proper actuation operation and the prevention of all critical situations could be obtained with a proper quick disengagement of the equalization device following the failure onset.

5 LIST OF SYMBOLS

Cil	hydraulic motor displacement [m^3]
CM	motor viscous damping coefficient [N s/rad]
Com	command [rad]
Cor1,2	valve 1, 2 input electric current [A]
CorM	Cor max absolute value [A]
D2ThM	hydraulic motors acceleration [rad/s^2]
DeCor	equalization differential current [A]
DePC1,2	valve 1, 2 commanded pressure [Pa]
DePHS	supply max differential pressure [Pa]
DThM	hydraulic motors angular rate [rad/s]
Err	position error [rad]
GC	proportional gain of the position loop and servovalve amplifier [A/rad]
Geq$_{\Delta p}$	pressure equalization device gain [A/Pa]
Geq$_{XS}$	valve position equalization device gain [A/m]
GP	valve pressure gain [Pa/m]
GPS1,2	valve 1, 2 secant pressure gain [Pa/m]
GQ	valve flow gain [m^2/s]
GS	speed loop gain [A/rad/s]
GV	global control valve position gain [m/A]
JM	hydraulic motor total equivalent inertia [kg m^2]
Offset1,2	valve 1, 2 offset equivalent current [A]
QM1,2	flow through hydraulic motor 1, 2 [m^3/s]
ThM	hydraulic motors angular position [rad]
ThS	moving surface position [rad]
TR	load acting on the moving surface [N]
XS1,2	valve 1, 2 second stage position [m]
XSM	max XS value [m]
ZM	surface/motor gear ratio
τ	equalization device time constant

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