MODEL-BASED FAILURE DETECTION OF A TRIMMABLE HORIZONTAL STABILIZER ACTUATOR WITH TWO PRIMARY LOAD PATHS

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ABSTRACT

A novel drive concept is being designed and analyzed for a trimmable horizontal stabilizer within the framework of the third national research program (LUFO III). The stabilizer actuator is based on two parallel operating ballscrews. The positions of the ballscrew nuts are mechanically synchronized to avoid a force-fight between the drive sections. This mechanical synchronization is realized by a specific configuration of two differential gearboxes.

The purpose of this project is to improve maintainability and incorporate an automatic testing capability for the mechanical integrity of the load carrying devices. Additionally, the mechanical wear of the heavily loaded components should be minimized and non-testable components should be eliminated. An approach for a model-based failure detection is presented in this article. A straightforward concept uses the sensor signals, which are needed for controlling the horizontal stabilizer to minimize the implementation effort. The detection of several mechanical failures is demonstrated by simulation.

KEYWORDS

Trimmable Horizontal Stabilizer Actuator (THSA); Active-parallel; Variable Displacement Hydraulic Motor (VDHM); Load Sharing; Failure Cases; Model-Based Failure Detection, Failure Isolation

1 INTRODUCTION

The pitch trim of transport aircraft is usually controlled by an adjustable horizontal tailplane. In accordance with the airworthiness requirements, all safety critical load bearing components of a Trimmable Horizontal Stabilizer Actuator (THSA) are designed to have two independent mechanical load paths between the horizontal tailplane and the aircraft fuselage structure. Conventionally, the secondary load path is only used in case of a mechanical disconnection of the primary load path, thus the second load path is normally unloaded. Generally, the trimmable horizontal stabilizer is moved by a ballscrew. The secondary load path is realized as a tie rod located inside the hollow ballscrew. The angle of attack of the trimmable horizontal stabilizer is varied by the screw nut, which moves the horizontal tailplane about a pivot hinge. In order to avoid a blow-back by air loads a friction no-back is installed.

Compared to a conventional THSA architecture, the concept presented here uses active-parallel operating load paths to move the stabilizer (cp. Figure 1). Two single load path ballscrew actuators (BS), each connected to the planet carrier shaft of a differential gearbox (DG), displace the trimmable horizontal stabilizer (THS). Each input shaft of the two differential gearboxes is mounted to a variable displacement hydraulic motor (VDHM). The remaining two inputs of the differential gearboxes are connected by a so called cross shaft (CS). This arrangement enables an auto-synchronization of the two ballscrew positions precluding force-fight.

A slight difference between the two ballscrew positions leads to different torques affecting the planet carrier shafts. Owing to the stationary gear ratio of $i_{12} = -1$ and the sum of moments at the shafts of the differential gearboxes, the torque difference will be compensated by the cross shaft, which rotates until the screw nut positions are equal. In order to avoid an uncontrolled movement of the control surface in case of failure, three hydraulic power off brakes (POB) lock the trimmable tailplane in position. Additionally, the mechanical integrity of the drive chains can be checked by activating one motor brake, while driving the system with the second hydraulic motor. In case of a mechanical damage, the motor will start then to rotate.

The advantages of this concept, when compared to conventional systems, are the following: the mechanical integrity of the two load paths can be tested automatically and in situ without the need for specific test equipment. This significantly reduces maintenance effort. Moreover, the mechanical wear on the ballscrews is reduced because of the load sharing between the two load paths. Owing to the concept of the two primary load paths, complex components like fail-safe designed gearboxes or a no-back brake are no longer required.

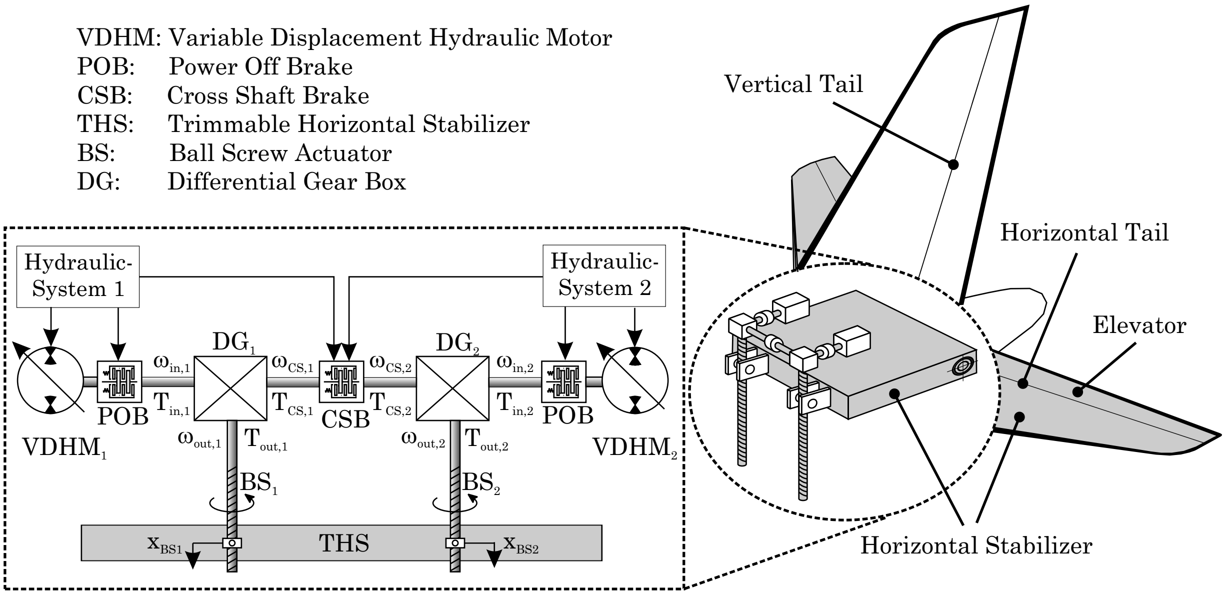

FIG 1: THSA system with two primary load path

In this report, a model-based monitoring concept and a test routine for the structural integrity of the THSA are disclosed. To minimize the amount of additional sensors for failure monitoring, the existing control sensor signals should be used to detect and to isolate mechanical failure cases. The required failure detection (monitoring) is based on either single sensor signals, which indicate a malfunction of the associated component or by a comparison of at least two signals. These may be either two sensor signals or one sensor and one model-based signal (estimated signal called 'reference signal'). The required reference signals are derived from a linear system model of the hydraulic motor. A short overview of the control strategy for the THSA is given, including the self-synchronization under varying external loads and motor speeds. The monitoring concept is validated by simulation, showing that the disturbance input, given by external air loads, does not affect the failure detection and that an isolation of critical failure cases is possible.

2 SYSTEM DESCRIPTION

The variable displacement hydraulic motor operates in a cascade control loop with a swashplate position control loop and a motor speed control loop (Biedermann 2005). The motor speed is controlled by the swashplate, which is actuated by a linear hydraulic cylinder. Air loads and inertia loads induce a load torque $T_{L,HM}$ on the hydraulic motor. The main components of this novel THSA are the two differential gearboxes coupled by the cross shaft. Each differential gearbox consists of two input shafts and one output shaft connected to the planet carrier. One input shaft is connected to the hydraulic motor (VDHM). The two remaining inputs of the differential gearboxes are coupled by a cross shaft. The output shafts of the two differential gearboxes are connected to the ballscrews, transforming the rotational motion to a linear displacement. (cf. figure 1). The differential gear consists of two large gear rings; each wheel is joined to one input shaft. The two gear rings are coupled by three pairs of planetary wheels (cf. figure 2) In faultless mode, each gearbox transfers the mechanical power to the attached ballscrew.

2.1 Mechanical Power Transfer

To understand the kinematic behavior of the two coupled differential gearboxes, it is necessary to look at the equation of motion of a differential gearbox. First, the main rotational speed equation of the three shafts is given by (Müller 1998):

$$(1)\qquad \omega_1 - i_{12}\cdot\omega_2 - (1-i_{12})\cdot\omega_{PC} = 0,$$

with the rotational speed of the two input shafts ω_1, ω_2 and the planet carrier shaft ω_{PC}.
The stationary gear ratio i_{12} describes the ratio between the two gear rings when the planet carrier is fixed. With the stationary gear ratio of $i_{12} = -1$ and an additional transfer gear (i_{SG}) located at the input shafts, the main speed equation of the used differential gearbox is derived (cf. figure 1):

$$(2)\qquad \frac{1}{i_{SG}}\cdot\omega_{in,1,2} + \frac{1}{i_{SG}}\cdot\omega_{CS,1,2} - 2\cdot\omega_{out,1,2} = 0.$$

Taking into account the mechanical couplings through the cross shaft between the differential gearboxes and through the structure of the horizontal tailplane between the ballscrew nuts:

$$(3)\qquad \omega_{CS,1} = -\omega_{CS,2}, \qquad \omega_{out,1} = \omega_{out,2},$$

the superposition of the main speed equation (2) of both differential gearboxes results in:

$$(4)\qquad \begin{aligned} &\tfrac{1}{i_{SG}}\cdot\omega_{in,1} + \tfrac{1}{i_{SG}}\cdot\omega_{CS,1} - 2\cdot\omega_{out,1} = \dots \\ &\dots\tfrac{1}{i_{SG}}\cdot\omega_{in,2} - \tfrac{1}{i_{SG}}\cdot\omega_{CS,1} - 2\cdot\omega_{out,1} \end{aligned}$$

(5) $\Rightarrow 2 \cdot \omega_{CS,1} = \omega_{in,2} - \omega_{in,1}$.

In normal mode, when both hydraulic motors operate at the same rotational speed $\omega_{in} = \omega_{in,1} = \omega_{in,2}$, the cross shaft, as calculated in equation (5), does not move ($\omega_{CS} = 0$). The rotational speed of the planet carrier shafts $\omega_{out} = \omega_{out,1} = \omega_{out,2}$ in normal mode can be written as (eqn. (2)):

$$(6) \quad \omega_{out} = \frac{1}{2i_{SG}} \cdot \omega_{in}.$$

In case of fixed motor shafts, either by no movement commanded or activated power off brakes $\omega_{in} = \omega_{in,1} = \omega_{in,2} = 0$, the cross shaft speed is given by equation (5):

$$(7) \quad \omega_{CS,1} = \omega_{CS,2} = 0.$$

Thus, the mechanical coupling through the cross shaft ensures a fixation of the horizontal stabilizer without using the cross shaft brake.
If only one motor operates the actuation system and the second motor is deactivated with its pressure off brake applied $\omega_{in,1} = 0$, $\omega_{in,2} \neq 0$, for example in failure mode, the cross shaft speed results from equation (4):

$$(8) \quad \omega_{CS,1} = \frac{1}{2} \cdot \omega_{in,2},$$

and from equation 2 the planet carrier shaft speed is derived:

$$(9) \quad \omega_{out} = \frac{1}{4i_{SG}} \cdot \omega_{in,2}.$$

A summary of the presented operating conditions is given in the following table (table 1).

TAB 1: Rotational speed ratio in normal mode, stopped mode, 1-motor mode

ω_{in}	ω_{CS}	ω_{out}
$\omega_{in,1} = \omega_{in,2}$	$\omega_{CS} = 0$	$\omega_{out,1} = \omega_{out,2} = \frac{1}{2i_{SG}} \cdot \omega_{in}$
$\omega_{in,1} = \omega_{in,2} = 0$	$\omega_{CS} = 0$	$\omega_{out} = 0$
$\omega_{in,1} = 0$, $\omega_{in,2} \neq 0$	$\omega_{CS} = \frac{1}{2} \cdot \omega_{in,2}$	$\omega_{out} = \frac{1}{4i_{SG}} \cdot \omega_{in,2}$
$\omega_{in,2} = 0$, $\omega_{in,1} \neq 0$	$\omega_{CS} = \frac{1}{2} \cdot \omega_{in,1}$	$\omega_{out} = \frac{1}{4i_{SG}} \cdot \omega_{in,1}$

The sum of moments, affecting the input and output shafts of the differential gearboxes $T_{M,1}$, $T_{M,2}$, $T_{BS,1}$, $T_{BS,2}$, has to result in zero. The given torques are classified as $T_{M,1} = T_{in,1}$, $T_{M,2} = T_{in,2}$, $T_{BS,1} = T_{out,1}$, $T_{BS,2} = T_{out,2}$, such that the sum of moments of the THS actuation system can be calculated:

$$(10) \quad T_{in,1} + T_{in,2} + T_{out,1} + T_{out,2} = 0.$$

The base equation of differential gearboxes describes the relation of the two input torques T_1, T_2 considering the power flow (Müller 1998):

$$(11) \quad \frac{T_2}{T_1} = -i_{12} \cdot \eta_{DG}^{w1}, \quad \text{with}$$

$$(12) \quad \eta_{DG}^{w1} = \begin{cases} \eta_{12} & w1 = +1 \\ \frac{1}{\eta_{21}} & w1 = -1 \end{cases} \qquad w1 = \frac{T_1\,(\omega_1 - \omega_{PC})}{|T_1\,(\omega_1 - \omega_{PC})|}.$$

Taking into account the sum of moments of a common differential gear $T_1 + T_2 + T_{PC} = 0$ and equation (11) the torque at the planet carrier shaft T_{PC} is computed:

$$(13) \quad \frac{T_{PC}}{T_1} = i_{12} \cdot \eta_{DG}^{w1} - 1.$$

As it can be seen in equations (11) and (13), the torque ratio of a differential gearbox depends only on the stationary gear ratio and not on the rotational speed of the shafts. Neglecting the efficiency factor η_{DG}^{w1}, the torque ratio is independent of the operating status. With a coefficient $\eta_{DG}^{w1} = 1$, a stationary gear ratio $i_{12} = -1$ and the gear ratio of the spur gear i_{SG}, the torque relations of the THSA are written as (cf. figure 1):

$$(14) \quad \frac{T_{CS}}{T_{in,1}} = 1,$$

$$(15) \quad \frac{T_{out,1}}{T_{in,1}} = -2 \cdot i_{SG},$$

$$(16) \quad \frac{T_{out,1}}{T_{CS}} = -2 \cdot i_{SG}.$$

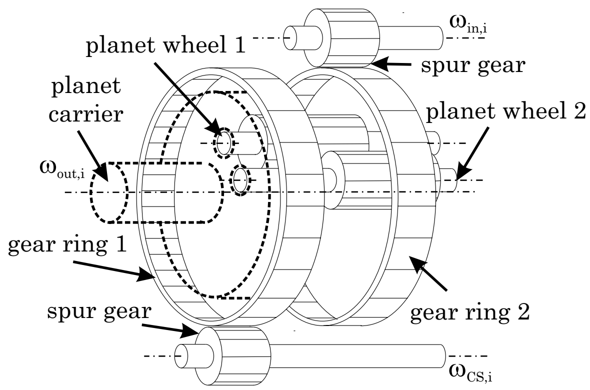

FIG 2: Gear ring, planet carrier and planetary wheels of the differential gear

2.2 Control Concept and Operating Conditions

In this control system, the plant consists of two drive chains each closed by proportional controller. The motor speed $\omega_{HM,c}$ represents the control variable of the position control loop and the angle position φ_{PC} of the planet carrier shaft constitutes the controlled variable (cf. figure 3). Assuming that both drive chains and control loops are dynamically identical, the design of the proportional controller will be conducted for a single drive chain.

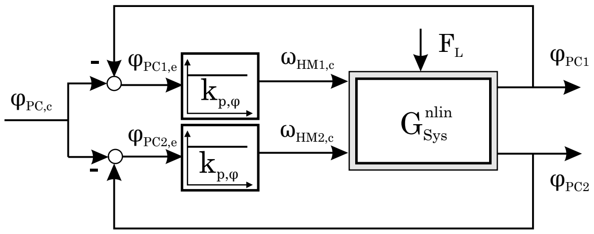

FIG 3: Decentralized position control loop

The dynamic requirements of the THSA control loop are listed below:

- maximum actuation speed ($\varphi_{THS} = \pm 1,2\,°/\mathrm{s} \Longrightarrow \omega_{PC} = \pm 40\,\mathrm{rad/s}$);
- amplitude margin of the open-loop system ($A_M > 10\,\mathrm{dB}$);
- phase margin of the open-loop system ($\Phi_M > 50°$).

The control gain is calculated with the help of the loop-shaping method (Lunze 1996). For this approach, the controller gain is determinated by means of the given amplitude margin. The amplitude and phase margins of the open loop are composed of the amplitude $|G_P(s)|_{\mathrm{dB}}$ and phase $\arg G_P(s)$ margins of the plant and the controller $|K(s)|_{\mathrm{dB}}$, $\arg K(s)$:

$$\begin{aligned} |G_0(s)|_{\mathrm{dB}} &= |G_P(s)|_{\mathrm{dB}} + |K(s)|_{\mathrm{dB}}\,, \\ \arg G_0(s) &= \arg G_P(s) + \arg K(s)\,. \end{aligned} \tag{17}$$

Because of the mechanical couplings inherent in the system, a transfer of power through the cross shaft only takes place if the motor speeds differ. The torque, affecting the cross shaft, has always the same value as the input shaft torques, owing to the stationary gear ratio $i_{12} = -1$ of the differential gearboxes (Müller 1998). Figure 4 highlights two fundamental features of the controlled THS drive system, which are explained in the following. In the case of different motor speeds, the cross shaft compensates the unequal power inputs by transferring energy from the drive chain with power excess to the other one (n_{CS}). In case of failure, the mechanical self-synchronization assures a force fight free moving of both ballscrew actuators. In figure 4, the rotational speeds of the hydraulic motors n_{HM1} and n_{HM2} and the cross shaft n_{CS}, as well as the linear displacement of the ballscrew nuts x_{BS} are displayed. The simulation starts with a normal movement of the actuation system by commanding an angle position φ_{PC}, which results in a linear displacement of the screw nut of 1 dm. Both hydraulic motors are activated at a time of 1 second to move the ballscrews. The required motor speed can be calculated from the maximal THS angular speed (cp. requirements and equation (6)):

$$\begin{aligned} &\omega_{HM,max} = \omega_{PC,max} \cdot i_{SG} \cdot 2 = 330.91\,\mathrm{rad/s}, \\ &\text{with} \quad i_{SG} = 4.136, \quad \omega_{PC,max} = 40\,\mathrm{rad/s}, \end{aligned} \tag{18}$$

which corresponds to a rotational speed of $n_{HM} = 3160\,\mathrm{rpm}$. For comparability, a motor speed of $3000\,\mathrm{rpm}$ is chosen.

Due to identical rotational speeds of the two motors, the cross shaft (n_{CS}) does not rotate. After two seconds motor 2 is shut down and the associated brake is activated. A motor deactivation can be caused by e.g. a pressure loss or a motor damage. Figure 4 shows that the cross shaft starts to turn when motor 2 is deactivated. Consequently, the ballscrew moves after deactivation of motor 2 with half speed (cp. equation (6) and (9)). The mechanical power supplied by the intact hydraulic motor is divided by the associated differential gearbox; one half is transmitted directly to the attached ballscrew and through the cross shaft and the second gearbox to the second ballscrew. The symmetrical distribution of the power flow is emphasized by the fact that the cross shaft rotates with half the speed compared to the input shaft (n_{HM1}), keeping in mind that the torque ratio is not affected by the rotational speeds.

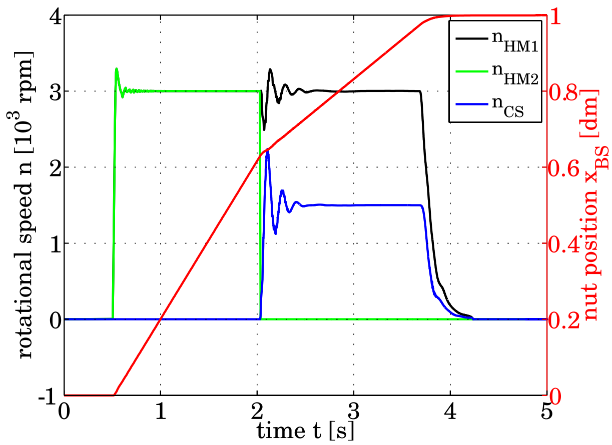

FIG 4: Rotational speeds and ballscrew position

The friction torque, affecting the elements of the differential gears, causes a bending stress in the coupling between the two ballscrew nuts, if the drive chains do not operate at exactly the same speed. In faultless operation a controller-based synchronization of the ballscrew nut position is possible. To avoid a permanent bending stress, an additional feedback of the planet carrier shaft speed and position differences is applied to the commanded motor speed signal as a so called correction speed.

In case of failure the mechanical self-synchronization assures a safety back up for moving both ballscrew actuators.

3 Failure Classes and Sensor Signals

The potential failure cases of the THSA with two primary load paths are classified into three groups:

- mechanical failure;
- hydraulic failure;
- electrical failure.

These failure classes include several different errors, whose effects and origins are explained in the next section.

3.1 Failure Cases

The mechanical failure cases include a disconnection and a jamming of a mechanical device (e.g. ballscrew jam or transmission shaft rupture). The hydraulic failure cases include a pressure loss and malfunctions of hydraulic control equipment (e.g. servo valve). The electrical failure cases comprise wiring error (sensor disconnection) and perturbances in the actuator control electronics.
Almost all failure cases, if not detected and counteracted, lead to an uncontrolled movement of the horizontal stabilizer, which is classified as a catastrophic event. The uncontrolled movement of the horizontal tailplane can be either a blow-back of the trimmable stabilizer by air loads or a powered runaway of a hydraulic motor. A mechanical disconnection of one shaft element, a gear wheel breakage or ballscrew disruption causes a blow-back.
A powered runaway is caused by a failure in the control functions, an electrical disconnection of a sensor signal or by a jamming of a servo valve. Jamming of the trimmable stabilizer is classified as an immobilization. In conclusion the critical failure cases are listed below:

- motor swashplate jam;
- pressure loss;
- disruption of one motor shaft or cross shaft;
- disruption of one ballscrew;
- jamming of one ballscrew.

3.2 Sensor Signals and Location

The sensors and signals used to detect the critical failure cases are as follows:

- hydraulic motor shaft position and speed sensor;
- hydraulic motor pressure sensor;
- hydraulic motor swashplate angle sensor;
- planet carrier position and speed sensor.

The location of the sensors and the potentially occurring failure cases at the trimmable horizontal stabilizer actuator are illustrated in figure 5. The position sensor, the pressure sensors and the swashplate angle sensors are integrated in the motor housing and the valve manifold. The position sensor of the planet carrier shaft of the differential gearbox is used to close the position control loop.
Two different failure scenarios may occur, when a single failure interferes the system. If one power drive unit, i.e. motor is affected by the defect (e.g. pressure loss of one hydraulic system) the motor has to be deactivated by closing the enable valve and applying the brake. The horizontal stabilizer is then moved by the second motor. A pressure loss of one hydraulic system must therefore not lead to an application of the cross shaft brake. The cross shaft brake is thus operated by two hydraulic supplies. In case of a mechanical disruption (e.g. ballscrew, transmission shaft) the whole system has to be shut down.
A comparison of the available sensor signals with suitable reference signals should lead to reliable fault detection. In order to be able to distinguish between the failure cases, the generated detection signals have to be combined logically. Thus, it is possible to identify the failure and to decide which action has to be taken. Moreover, maintenance information is made available.

4 MONITORDESIGN

In this chapter, the sensor signals for residual generation are chosen and a combination of the residuals is presented to isolate different failure cases. Four residuals are needed to detect and isolate the following three failure cases.

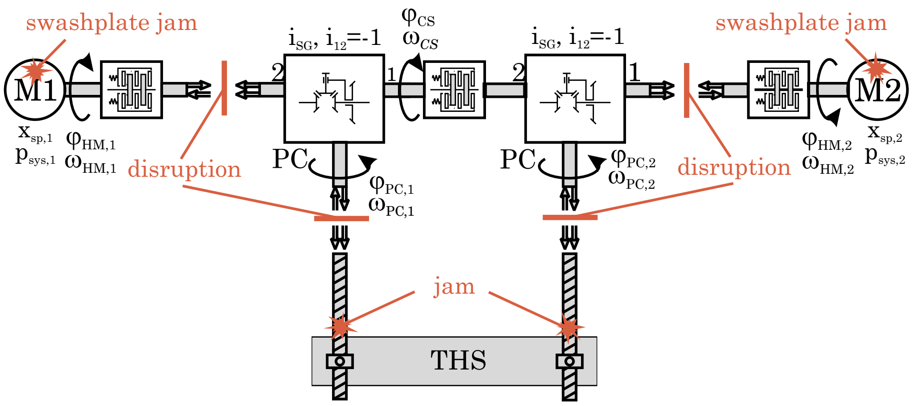

FIG 5: Failure location and sensor signals

These cases are chosen, because they lead directly to an uncontrolled movement of the horizontal stabilizer.

- swashplate jam;
- disruption of one motor shaft;
- disruption of one ballscrew.

To identify failures of the hydraulic motors, reference signals have to be created for comparison with the measured motor shaft position. For this a linear system model of the hydraulic motor is used to generate this calculated reference signal.

4.1 Residual Generation

The approach used to detect failures is based on calculating the difference of a measured signal and the corresponding reference signal. A faultless range of the difference signal $r_{DS,i}$ is defined by evaluating thresholds $\varphi_{TH,i}$, which denote the error margin.
The thresholds are defined under consideration of the disturbance value to the system. The disturbance variable of the THSA are the air loads, acting on the stabilizer; their maximum design value is known. If the difference signal $r_{DS,i}$ exceeds the error margin $[-\varphi_{TH,i}, \varphi_{TH,i}]$, a confirmation cycle is initialized. If this cycle is not interrupted, the associated residual r_i is set to $[-1, 1]$, depending on whether the positive or negative threshold is crossed.
In consequence, the monitor concept consists of four residual values, which are calculated from the measured angle positions of the hydraulic motors φ_{HM}, the cross shaft position φ_{CS} and the planet carrier shaft position φ_{PC}. Additionally, reference signals are needed to compare them with the measured signals $\varphi_{i,R}$.

$$
\begin{aligned}
r_{DS,1} &= \varphi_{HM1,R} - \varphi_{HM1} \\
r_{DS,2} &= \varphi_{HM2,R} - \varphi_{HM2} \\
r_{DS,3} &= \varphi_{CS,R} - \varphi_{CS} \\
r_{DS,4} &= \varphi_{PC1} - \varphi_{PC2}.
\end{aligned} \tag{19}
$$

The residuals attain the following values according to the threshold values $\varphi_{TH,i}$:

$$
r_i = \begin{cases} 1, & r_{DS,i} > \varphi_{TH,i} \\ 0, & -\varphi_{TH,i} \leq r_{DS,i} \leq \varphi_{TH,i} \\ -1, & -\varphi_{TH,i} > r_{DS,i} \end{cases} \tag{20}
$$

For $r_{DS,4}$, the measured signals of both planet carrier shafts are used instead of an additionally calculated model-based reference signal. The signal $r_{DS,4}$ provides information on a ballscrew rupture. An uncontrolled movement of the stabilizer will be detected because in this case the measured position signals are uncorrelated to the reference signals. Because of the mechanical couplings and the speed- and torque relations of the inputs and outputs of the differential gearboxes in the THSA system, the analyzed failure scenarios cause different behavior of the residuals.
In order to consider the VDHM dynamic behavior, using the commanded motor speed $\omega_{HMi,c}$ as the comparison signal, the VDHM dynamic is described by a 4th order linear model. To generate the reference signals $\varphi_{i,R}$, the commanded speed signal $\omega_{HMi,c}$ is filtered with the 4th order VDHM model.

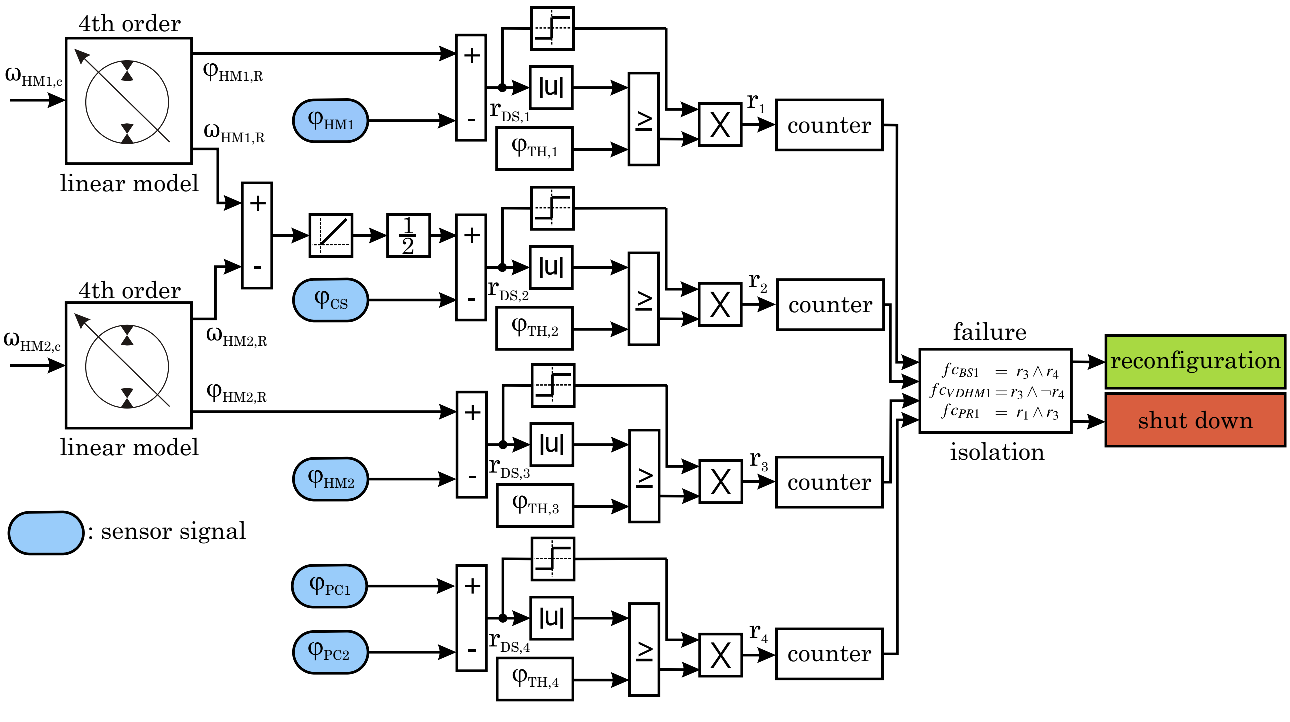

FIG 6: Residual generation

The motor model provides directly the reference signals $\varphi_{HM1,R}$, $\varphi_{HM2,R}$. The value $\varphi_{CS,R}$ has to be calculated from the kinematic equations of the differential gearbox (cf. eqn. (5)) and the speed signals:

$$\varphi_{CS,R} = \int \frac{\omega_{HM1,R} - \omega_{HM2,R}}{2}. \tag{21}$$

The monitor structure is shown in figure 6. The linear model of the VDHM includes the swashplate actuation system. The swashplate is displaced by a linear hydraulic cylinder, which is controlled by a servo valve. The linear swashplate actuator model consists of the electromechanical conversion of the solenoid valve current to the valve spool position, as well as the pressure equation of the linear cylinder and the equation of motion of the actuator piston (Backé 1992).

The integration of the equation of motion for the hydraulic motor into the model leads to an 8th order model with the inputs being commanded speed $\omega_{HM,c}$ and load torque T_L and the outputs swashplate position x_{SP}, spool position y_S, output shaft angular position $\varphi_{HM,R}$ and output speed $\omega_{HM,R}$. This linear model is reduced to 4th order by the balanced truncation method (Werner 2004).

The difference signal generation is dependant on the disturbance input of the THSA nonlinear model. The air loads, affecting the stabilizer, cannot be taken into account in the calculation of the reference signals because the air loads are unknown. Thus, the thresholds have to be adapted to the maximum amplitude of the difference signal in faultless operation.

The difference signals $r_{DS,i}$ for most unfavorable constellation of load and motor speed is shown in figure 7. The signals and thresholds are standardized to one. In this case the THSA is driven with one motor only because the differential signals $r_{DS,3}$ and $r_{DS,4}$ are only effected, if the drive chains operate at different speeds. The hydraulic motor is accelerated at $t = 0.5\,\mathrm{s}$ to maximal rotational speed (3000 rpm). At $t = 2\,\mathrm{s}$ the load of the THSA is set to maximum design loads, which correspond to 1.7-times normal operation loads. The difference values $r_{DS,1}$ and $r_{DS,3}$ are influenced by the increase of the hydraulic motor speed because the linear motor model does not respect the load torque, which is not measured directly. Thus, as motor 2 is stopped, the value of $r_{DS,2}$ is always zero. The difference signal $r_{DS,4}$ varies because of the nonlinear effects like backlash and friction, which are not considered in the linear model. If only one motor actuates the system, the backlash and friction torque affecting the mechanical components (esp. the differential gearbox) lead to a slight position difference of the ballscrew nuts. This position difference is detected by the difference signal $r_{DS,4}$, since $r_{DS,4}$ is calculated from the angle difference of the planet carrier shaft position. The planetary carrier is directly coupled to the ballscrews. As it can be seen in figure 7, the threshold value is not exceeded. Both the step to maximum motor speed and the step to maximum design loads do not cause a false alarm.

There exists a possibility to obtain the load torques of the hydraulic motors by utilizing the swashplate position signal and the motor pressure signal. The required driving torque can be derived from the swashplate position and the pressure difference of the motor and inserted in the linear system model. Since this approach is a more complex one and failures (e.g. sensor failures) in the load calculation cause unclear effects, this strategy will not yet be used. Additionally, jamming of the THSA system can be predicted with the same method described above. For similar reasons the jam detection is a subject of further investigations.

If all input values of the plant are limited and the limit values are known, it is possible to calculate analytically the error margin by the reachability analysis (Geilsdorf, Raksch and Carl 2007). Since the input values of the horizontal stabilizer actuator are restricted, this new method to calculate error margin will be applied to this system.

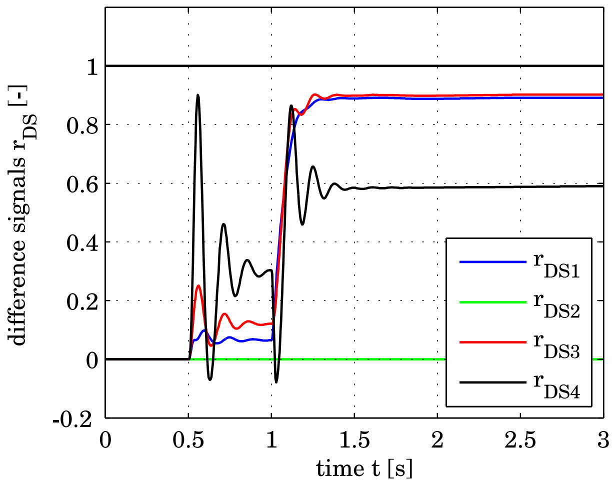

FIG 7: Threshold validation

4.2 Failure Isolation

A logical combination of the four generated residuals leads to an isolation of the different mechanical failure cases. Two different actions can be taken if a failure occurs. In case one motor fails (swashplate jam), the affected motor has to be deactivated and the system is driven by one motor only, transferring mechanical energy via the cross shaft. In case of disruption or jamming of one of the mechanical elements (shafts and ballscrews), the whole system has to be shut down. The behavior of the residuals is summarized in Table 2.

TAB 2: Failure isolation

Failure case	r_1	r_2	r_3	r_4
BS1 disruption	0	0	1	1
BS2 disruption	0	0	-1	-1
VDHM1 disruption	0	0	1	0
VDHM2 disruption	0	0	-1	0
VDHM1 swashplate jam	1	0	1	0
VDHM2 swashplate jam	0	1	1	0

It is possible to distinguish between the listed failure cases by logically combining the four residuals (r_1, r_2, r_3 and r_4) to the failure indicator value fc_i:

$$\begin{aligned} fc_{BS1} &= r_3 \wedge r_4\,, \\ fc_{VDHM1} &= r_3 \wedge \neg r_4\,, \\ fc_{PR1} &= r_1 \wedge r_3\,. \end{aligned} \tag{22}$$

Table 2 points out that all analyzed failure cases can be detected with the residual value r_3. The monitoring of the cross shaft position is sufficient for a basic failure detection. The three additional residuals are used for failure isolation.

5 SIMULATION

In this chapter, the complex model including all non-linear components and effects is utilized (e.g. backlash, stiction, saturation effects) to show THSA non-linear dynamic behavior. Simulation results for the failure cases swashplate jam and ballscrew disruption are discussed in the following passages. In order to facilitate the comparison of the results a set point profile is defined (cf. Figure 8):

$t \in [0, 0.5]$ s	commanded stop.
$t = 0.5$ s	commanded position step of planet carrier position from $\phi_{PC,c} = 0$ rad to $\varphi_{PC,c} = 91$ rad which corresponds to 100 mm ballscrew stroke.
$t = 1$ s	failure cases (ballscrew disruption or swashplate jam) are initialized
$t \in [0.5, 2.9]$ s	in faultless mode stationary commanded motor speed of 3000 rpm.
$t \in [2.9, 3.1]$ s	deceleration because the final position value is reached.
$t \in [3.1, 5]$ s	commanded stop.

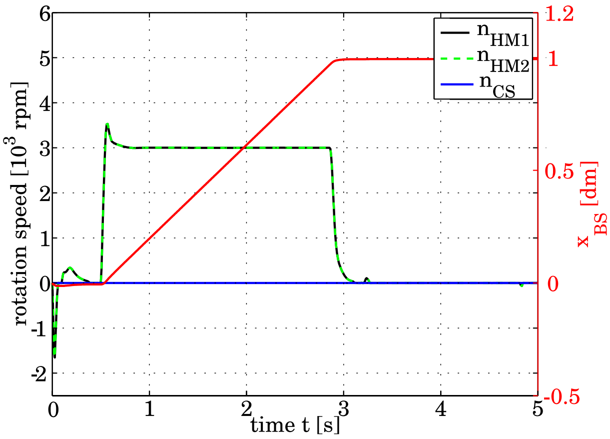

FIG 8: Rotational speeds at normal operation

The model also includes the air loads affecting the THSA. A slight difference at the beginning of the simulation $[0, 0.5]$ s is a result of the simulated stiction. The control error is too small for the actuator to overcome the breakaway torque. Because of the integrator in the speed control loop of the VDHM, the motors will however move after a certain time and come to rest again (limit cycling due to integrator wind-up).

For failure detection one important characteristic concerning the air load direction of action has to be mentioned. Owing to the aerodynamical configuration (wing-tailplane configuration) of transport aircraft, the horizontal tailplane has to generate negative lift to achieve a stable flight attitude. Thus, the sign of the disturbance value of the THSA is independent of the flight condition.

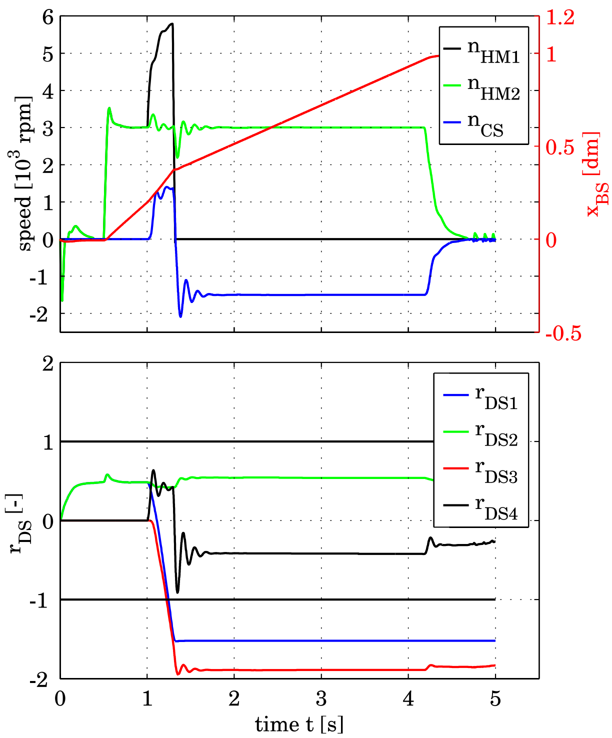

FIG 9: Powered Runaway of motor 1

5.1 Swashplate Jam

The swashplate jam is simulated in the nonlinear actuation system by setting the swashplate displacement to maximum value. The output torque of the hydraulic motor therefore is also raised to the maximum value. The motor starts to accelerate until the output torque is equal to the load torque. This motor behavior is named powered runaway. As initial condition a powered runaway against external loads is chosen, because this failure case is most difficult to detect. The effects of the runaway against loads to the THSA system are not as serious as a runaway in load direction. Hence, only a slight variation of the difference values will be the result.

Figure 9 shows the rotational speed and difference signal behavior in case of a swashplate jam at motor 1. As it can been seen in the upper chart, the swashplate jam causes a speed-up of the affected motor to an amplitude of 5800 rpm. Owing to the speed difference of both hydraulic motors the cross shaft starts to rotate. The difference signals $r_{DS,1}$ and $r_{DS,3}$ are affected by this error case, because these signals are calculated with the help of the shaft angular position of motor 1 and the cross shaft position. After a time of 290 ms the power off brake decelerates the motor to a full stop and the enable valve is closed. Additionally, the commanded motor speed is set to zero. The difference value exceeds the error margin after a time period of 240 ms. The time difference between shut down and failure detection results from the conformation cycle of 50 ms. The motor is stopped after 325 ms. The difference values $r_{DS,2}$ and $r_{DS,4}$ are slightly influenced by the failure, but they do not exceed the error margin. Therefore the residuals are not set.

The second motor drives the ballscrews with half speed to the final position. The cross shaft turns with half motor speed (1500 rpm) after shut down of the defect motor and transfers power to the other drive chain. The mechanical self-synchronization permits, although one motor does not work, a movement of the trimmable horizontal stabilizer.

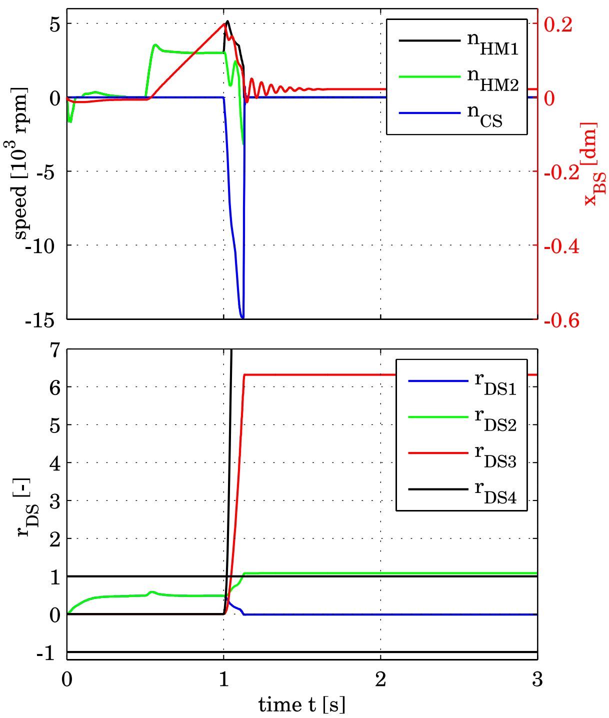

FIG 10: Ballscrew disruption

5.2 Ballscrew Disruption

To simulate a ballscrew disruption, the torque linkage between ballscrew and planet carrier shaft is cut. The horizontal stabilizer cannot be held in position any longer until the power off brakes are activated. Due to the torque relations between the inputs and output of the differential gear (cp. equations (11) and (13)), a free-wheel of the planet carrier shaft results in the differential gear not being able to carry any torque. The consequent taken is to activate all power off brakes, if a ballscrew or a transmission shaft disruption occurs, in order to fix the horizontal stabilizer with the intact ballscrew. The cross shaft brake and the motor brake mounted to the faultless drive chain then hold the applied load torque, which is transferred from the unharmed ballscrew through the differential gearbox to the two brakes. In figure 10 the speed and difference signal behavior in case of a simulated ballscrew disruption (ballscrew 1) is shown. A position command of 100 mm is given to the position control loop after half a second. The breakage takes place after one second. Up to that point, the drive system moves the horizontal stabilizer against the air loads.

The air loads drive the horizontal tailplane back, which leads to a fast rotation of the cross shaft. This high velocity is reached because the disruption takes place at the operating point with maximum air loads. The difference signals $r_{DS,3}$ and $r_{DS,4}$ cross the threshold after a time period of 47 ms. The pressure off brakes are set after 97 ms and the system is stopped after 154 ms. The nuts travel back 17.6 mm until the brakes are activated. The speed of motor 1 increases when the ballscrew disrupts because there is no load affecting the motor subsequently. Because of the remaining mechanical couplings between the differential gearboxes and friction torque influencing the gearboxes the motors are accelerated by the moving ballscrew 2 until the brakes are activated.

5.3 Discussion

Both the swashplate jam and the ballscrew disruption are detected by the presented failure detection approaches. An isolation of the individual failure cases is possible.

In case of a powered runaway, the defect motor is completely shut down after 325 ms. After the failure has been detected, the horizontal stabilizer actuator is driven by the remaining motor. Figure 9 points out that the motor power is divided and the cross shaft transfers power from the drive chain with power excess to the other one. Thus, the torque amplitudes effecting the cross shaft and the motor are the same (cf. equation (11) and $i_{12} = -1$) and the amplitude of the rotational speed of the cross shaft is half the motor speed (cp. figure 9 and equation (4), (8)) the motor power is shared exactly between the two ballscrews.

The whole system has to be shut down after a ballscrew or a transmission shaft disruption, because a safe movement of the stabilizer is no longer possible. Figure 10 shows that the system is fully stopped after a time period of 154 ms. The screw nuts travel 17.5 mm after the failure has occurred, which corresponds to a deflection of the horizontal tailplane of 0.5°.

A detection of a jam case is possible with the help of the swashplate sensor signal, the pressure signal and the motor speed value. To obtain information on the jam location is not possible without additional effort.

Several additional thresholds have to be defined and considered in the failure detection routines. These are simple error margin monitors like an overspeed or an overtravel threshold to protect moving parts from high dynamical loads. For example, a powered runaway in effective direction of the air loads leads to high rotational velocity of the affected motor. The detection of this failure case with an overspeed monitoring will be a few milliseconds faster in some circumstances (high external loads), which is import-

ant in this case.
A pressure loss of one hydraulic system and a resulting reduced power failure at the attached motor can be detected with the same residuals used for the powered runaway detection. The failure detection times are listed in table 3
In conclusion the analyzed failure cases are detected by the presented monitoring strategy. Similarly, an isolation of the failure cases is possible with exception of the jam case. The THSA is transferred to a safe operating condition in case of failure.

TAB 3: Failure detection times

Failure case	Detection time	Fully stopped
BS disruption	47 ms	154 ms
Shaft disruption	98 ms	156 ms
Swashplate jam	240 ms	325 ms
Pressure loss	97 ms	183 ms

6 CONCLUSION

This article presents a model-based monitoring strategy for mechanical failure cases of a trimmable horizontal stabilizer actuator with two ballscrew actuators, working in active-parallel configuration. An approach for failure detection and failure isolation in case of a mechanical disruption is given. The method uses the available control sensor signals of the THSA, especially the motor signals. The measured signals are compared to reference signals which are generated by a 4th order linear system model of the variable displacement hydraulic motor. Four residual values are calculated from the sensor signals and these values are logically combined to isolate the analyzed failure cases: swashplate jam of one hydraulic motor, transmission shaft disconnection, ballscrew disruption and pressure loss of one hydraulic system. A cross shaft rupture shows identical residuals as a motor shaft disconnect.
If only one motor is affected by a failure, the motor is shut down and the actuation system can be operated by the remaining motor. Owing to the mechanical synchronization of the ballscrew nut positions by means of the differential gears, the mechanical power is transferred through the cross shaft to the drive chain with stopped motor. The trimmable horizontal stabilizer can be actuated with half speed then.
A shaft disconnection or a ballscrew disruption lead to a complete shut down of the THSA. An uncontrolled movement is avoided by activating the installed power off brakes. The logical combination of the residuals permits a decision which action has to be taken.
Simulations show that the chosen threshold values allow a fast and safe failure detection and isolation without producing a false alarm. Nevertheless, the jamming case cannot be identified with the presented monitor concept. Further investigations will show if a jam detection and its location is possible with additional, but already existing sensor signals. With the help of a reachability analysis the error margins will be calculated analytically.
In order to validate the presented THSA monitor concept, a test rig is currently being constructed.

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