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Active fault tolerant control of an electro-hydraulic servo axis with a duplex-valve-system ⋆

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Abstract: In this paper fault detection, fault diagnosis and active fault tolerant control of an electro-hydraulic servo-axis with a duplex-valve-system are described. The fault detection is based on parity equations. The semi-physical models allow the detection of even small faults in the hydraulic system. The fault diagnosis used on the testbed is based on *fuzzy-logic*. In order to tolerate a failed hydraulic proportional valve, a duplex-valve-system built up with standard direct-driven proportional valves is applied. The fault management module allows the supervision of the hydraulic servo axis and decides on reconfiguration and fault accomodation of the control-loop. The Internal Model Control (IMC)-tracking control structure used for reconfiguration allows bumpless transfer between controllers. Experimental results show the industrial applicability of the approach.

Keywords: Fault Detection, Fault Diagnosis, Fault Tolerance, Fault Accomodation, Electro-hydraulic Servo Axis, Duplex-valve-system, Active Fault Tolerant Control, Reconfiguration, Bumpless-Transfer

1. INTRODUCTION

The reliability and safety of electro-hydraulic servo axis are very important not only in safety-related systems like aircrafts but also in common industrial hydraulic systems (consider Münchhof et al. [2009]). The costs caused by system-downtime, repair time or liability for damages often exceed the costs of a supervision module and redundant components. If redundant components exist in a system different types of redundancy concepts can be applied. Theoretical investigations and measurements on a testbed have shown that the best operation mode of the duplex-valve-system is the operation as a dynamic redundancy system with hot standby (consider Beck [2010]). In this case each valve must be supervised by a fault detection and isolating (FDI) module (see Blanke et al. [2006]). In faultfree operation both valves should be active. The advantages of this operation mode are the avoidance of plugging of the valves and the possibility of permanent supervision of both valves. If a severe fault occurs in either of the valves and is detected by the FDI-modules the spool of the faulty valve must be moved to the neutral position. To achieve the neutral position a passive and an active way exist. The passive way is to simply shutdown the power electronics of the electromagnets, such that the centering springs push the valve spool in the neutral position. The active way is to set the reference value of the faulty valve to zero. The two electromagnets push the valve spool in the neutral position. The active way is only possible, if the spool position control loop is still working. Otherwise only the passive way is applicable. If the valve spool is moved to the neutral position, the cascaded position control loop of the valve (for detailed information see Beck et al. [2009]) should be shut down.

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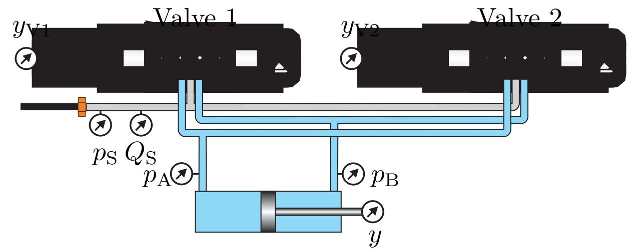

Fig. 1. Schematic assembly of the testbed with a duplex-valve-system

The superposed piston position control loop should be designed fault tolerant to allow for any valve faults. Basically, two different approaches exist to achieve fault tolerance (see Patton [1997]), namely passive and active fault tolerant control (FTC). In the first case the controller is designed to deal with faults without adaption of the controller parameters. Often, methods of robust control theory are used.

The second approach to achieve a fault tolerant control loop is active fault tolerant control. After detection and isolation of a fault the actuating and measured variables are modified (fault accommodation) or the control loop is reconfigured (control reconfiguration). In this paper the focus is on control reconfiguration of an electro-hydraulic servo axis with a redundant duplex-valve-system and in particular on the bumpless switchover between controllers.

The paper is organized as follows. In Section 2 the process model based fault detection with parity equations for the early detection of small faults in the electro-hydraulic servo axis is presented. In Section 2.2 the observed symptoms are evaluated by a fuzzy-logic diagnosis system. The fault accommodation and reconfiguration of the hydraulic system with a duplex-valve-system are explained in Section 3. In this section, the focus is on the IMC-tracking control structure which allows bumpless transfer between controllers. In Section 4 this approach is applied to an electro-hydraulic servo axis with a duplex-valve-system (see Fig. 1). The verification on the testbed points out the industrial applicability of the approach.

2. FAULT DETECTION AND DIAGNOSIS

Fault detection and diagnosis of the electro-hydraulic system are very important if the duplex-valve-system operates as a dynamic redundancy system. Since the spare valve is activated/deactivated dependent on the system state (e.g. faultfree operation or degraded operation), the electro-hydraulic system must be supervised by a Fault Detection and Isolation (FDI) module (see Fig. 2). Theoretical researches and measurements on an electro-hydraulic servo axis (see Münchhof [2006], Beck [2010]) have shown that parity equations (based on semi-physical models) and estimation of physical parameters (see Isermann [2006]) allow an early and reliable detection of small faults. The diagnosis system (see Fig. 3) is based on the inference method. A static fuzzy-logic diagnosis system allows the online diagnosis of detected faults and runs on a rapid control prototyping system (dSPACE-system) in realtime.

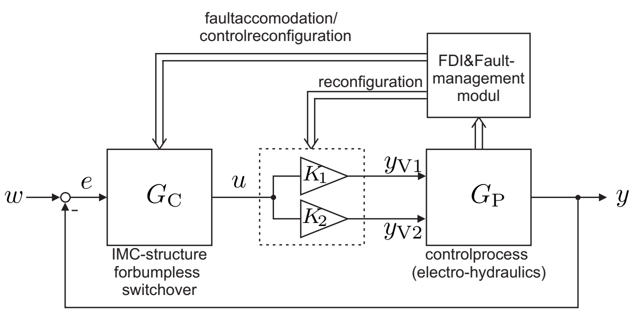

Fig. 2. Fault tolerant control loop (see Beck [2010])

2.1 PROCESS MODEL BASED FAULT DETECTION

In this subsection the process model based fault detection of the direct-driven 4 ports 3 way proportional valves (Manufacturer: Rexroth, Typ: 4WRE Valve) and the hydraulic cylinder is presented. Figure 3 shows the three fault detection modules and the fault diagnosis module that allow the supervision of the electro-hydraulic system. Theoretical investigations and measurements on a testbed (see Beck [2010]) have shown that the process model based fault detection is particularly suitable for the detection of small faults in the proportional valves. Although model-based fault detection presented in Beck et al. [2009] has been developed for direct driven proportional valves the methods can be applied to many other electromagnetic actuators as well.

The direct-driven proportional valve is a typical example of a mechatronic system (see Isermann [2005]). The valve spool of the direct-driven proportional valve is moved by two direct-current electromagnets. Such electromagnetic actuators contain different energy conversion mechanisms. First, the electric energy supplied to the actuator is converted into magnetic energy, which is subsequently converted into mechanical energy. The position of the valve spool measured with a linear variable differential transformer and controlled by a cascaded position controller affects the volume flow rate over the four control edges of the proportional valve. Faults can affect the electromagnetic conversion as well as the magneto-mechanic conversion. In order to detect faults in the closed valve spool control loop, model based fault detection in closed loop described in Isermann [2006] is implemented. Therefore, in Beck et al. [2009] models are presented that cover both the electro-magnetic as well as the magneto-mechanic energy conversion. Two isolating parity equations for the electromagnetic part of direct-current electromagnets and one parity equation for the magneto-mechanic part of the valve allow the detection of small faults in closed loop operation. For detailed informations about the parity equations consider Beck et al. [2009].

The overall duplex-valve-system used for building up a fault tolerant valve-system consists of two standard hydraulic proportional valves that are mounted in parallel (see Fig. 1). Thus, the valves of the duplex-valve-system can be controlled separately. The hydraulic valves can be supervised using two of the fault detection modules presented in Beck et al. [2009].

Not only the valves, but also the hydraulic cylinder should be supervised by a FDI module (see Münchhof et al. [2008]). For this reason model based fault detection for a hydraulic cylinder is presented in this paper. The dif-

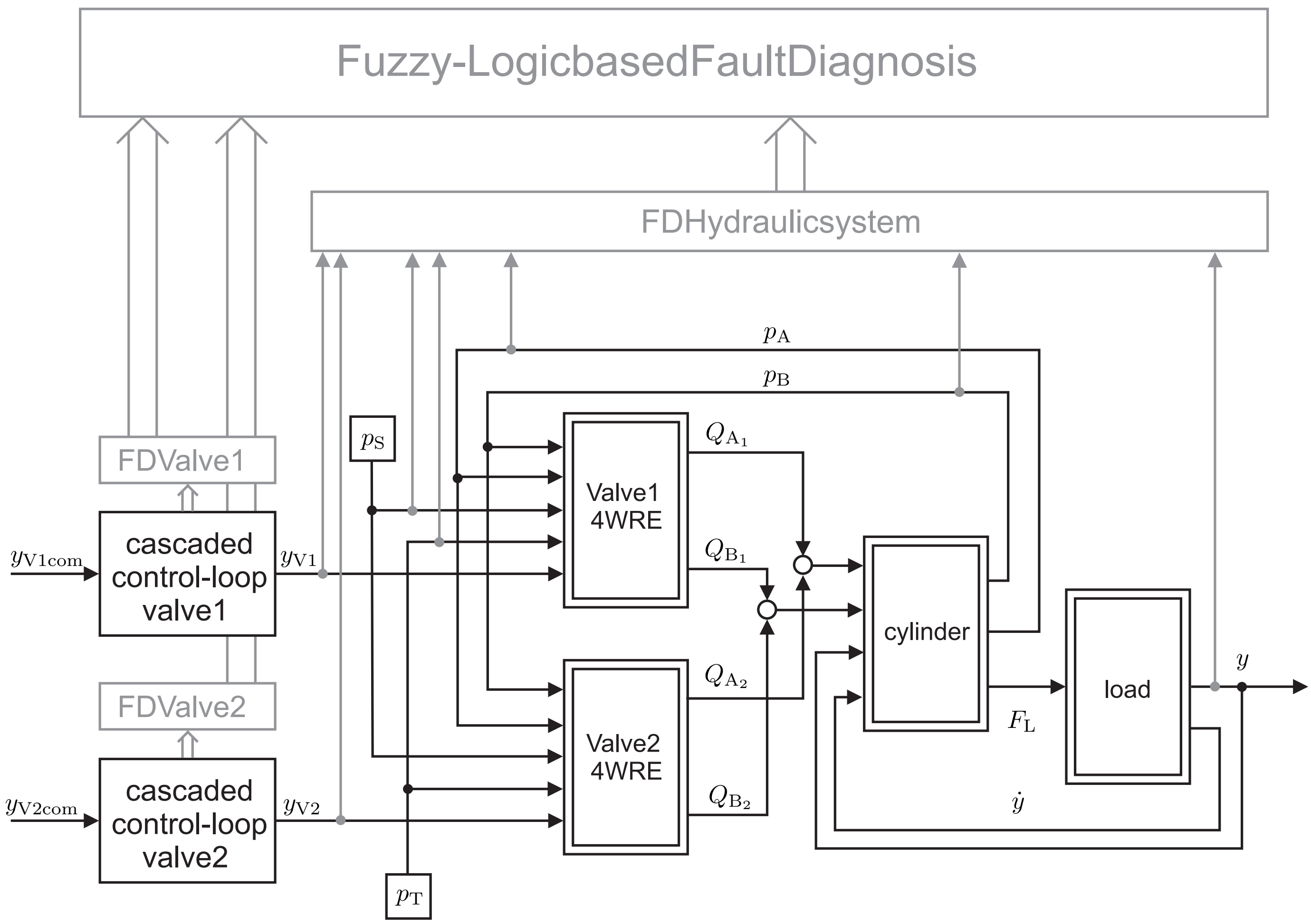

Fig. 3. Fault detection and diagnosis modules of the control process

ferential cylinder (schematically shown in Fig. 1) used on the testbed is a double acting differential cylinder with a one-sided piston rod. The pressure built-up in chamber A and chamber B can be described by (see Münchhof [2006])

$$\dot{p}_{\mathrm{A}}(t) = \frac{E\left(Q_{\mathrm{A}} - G_{\mathrm{AB}}\left(p_{\mathrm{A}}(t) - p_{\mathrm{B}}(t)\right) - A_{\mathrm{A}}\dot{y}(t)\right)}{V_{0\mathrm{A}} + A_{\mathrm{A}}y(t)}, \quad (1)$$

$$\dot{p}_{\mathrm{B}}(t) = \frac{E\left(Q_{\mathrm{B}} + G_{\mathrm{AB}}\left(p_{\mathrm{A}}(t) - p_{\mathrm{B}}(t)\right) + A_{\mathrm{B}}\dot{y}(t)\right)}{V_{0\mathrm{B}} - A_{\mathrm{B}}y(t)}, \quad (2)$$

with

E:	bulk modulus,
A_{A}:	active piston area chamber A,
A_{B}:	active piston area chamber B,
$V_{0\mathrm{A}}$:	dead volume chamber A,
$V_{0\mathrm{B}}$:	dead volume chamber B,
Q_{A}:	volume flow (chamber A),
Q_{B}:	volume flow (chamber B),
G_{AB}	laminar leakage coefficient,
p_{A}	pressure in chamber A,
p_{B}	pressure in chamber B,
y	piston position.

In order to detect faults in the hydraulic system four residuals based on parity equations are formulated. With negligence of internal leakage ($G_{\mathrm{AB}} = 0$) the pressure built-up can be modelled by

$$\dot{\hat{p}}_{\mathrm{A}}(t) = \frac{E\left(\hat{Q}_{\mathrm{PA}} - \hat{Q}_{\mathrm{AT}} - A_{\mathrm{A}}\dot{y}(t)\right)}{V_{0\mathrm{A}} + A_{\mathrm{A}}y(t)}, \quad (3)$$

$$\dot{\hat{p}}_{\mathrm{B}}(t) = \frac{E\left(\hat{Q}_{\mathrm{PB}} - \hat{Q}_{\mathrm{BT}} + A_{\mathrm{B}}\dot{y}(t)\right)}{V_{0\mathrm{B}} - A_{\mathrm{B}}y(t)}, \quad (4)$$

with

$$\hat{Q}_{\mathrm{A}} = \hat{Q}_{\mathrm{PA}} - \hat{Q}_{\mathrm{AT}}, \quad \hat{Q}_{\mathrm{B}} = \hat{Q}_{\mathrm{PB}} - \hat{Q}_{\mathrm{BT}}, \quad (5)$$

and

$\hat{Q}_{\mathrm{PA}}$:	modelled volume flow over the control edge PA,
$\hat{Q}_{\mathrm{AT}}$:	modelled volume flow over the control edge AT,
$\hat{Q}_{\mathrm{PB}}$:	modelled volume flow over the control edge PB,
$\hat{Q}_{\mathrm{BT}}$:	modelled volume flow over the control edge BT.

Due to the negligence of G_{AB} in the above parity equations, an increasing leakage becomes visible in the residuals and can be detected. The modeling of the four way three ports proportional valve and in particular the modeling of the volume flow rates over the control edges is described in Beck [2010], where a semi-physical model of the proportional valve is considered. The parameters of

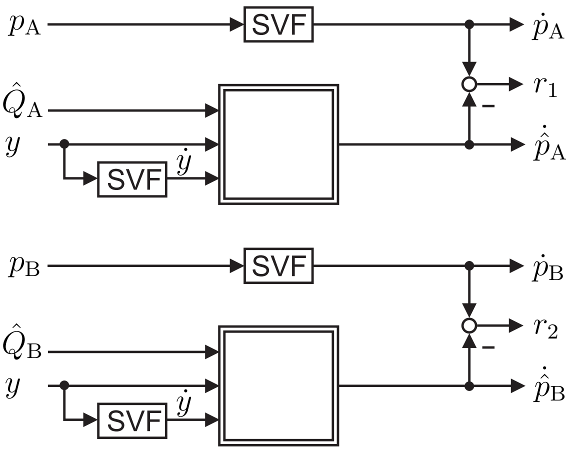

Fig. 4. Structure of residual r_1 and r_2

the model are obtained by measurements on the testbed and identification methods (see Isermann [1992b]). With Equation (3) residual r_1 can be formulated as

$$r_1 = \dot{p}_A - \hat{\dot{p}}_A. \tag{6}$$

Figure 4 shows the schematic structure of residual r_1. The derivation with respect to time of the piston position $\dot{y}$ and the pressures, $\dot{p}_A$, $\dot{p}_B$ are each obtained by a state variable filter (see Fig. 4: SVF). The design of SVF is described in Isermann [1992a].The pressure p_A in chamber A is measured with a common pressure transmitter. The position signal is obtained from the superimposed control loop.

Analog to residual r_1 residual r_2 can be formulated with Equation (4) as

$$r_2 = \dot{p}_B - \hat{\dot{p}}_B. \tag{7}$$

The residuals r_3 and r_4 are also based on parity equations. The velocity of the hydraulic piston, $\dot{y}$, is modelled. With rearrangement of Equation (3) the residual r_3 follows to

$$r_3 = \dot{y} - \hat{\dot{y}}. \tag{8}$$

Figure 5 shows the schematic structure of residual r_3. The derivation of the pressure p_A with respect to time $\dot{p}_A$ is obtained by a SVF (see Fig. 5).

Analog to residual r_3 residual r_4 is obtained by the rearrangement of Equation (4). The residual r_4 follows to

$$r_4 = \dot{y} - \hat{\dot{y}}. \tag{9}$$

The schematic structure is shown in Figure 5. The derivation of the pressure p_B with respect to time $\dot{p}_B$ is also obtained by a SVF (see Fig. 5).

Additionally, we monitor the operating range of the actuating signal y_v. If the actuating signal is below or above of two given tresholds the symptom is set to one, i.e., we obtain

$$r_5 = \begin{cases} 0, & \text{if } y_{v,\min} < y_v < y_{v,\max} \\ 1, & \text{else} \end{cases} \tag{10}$$

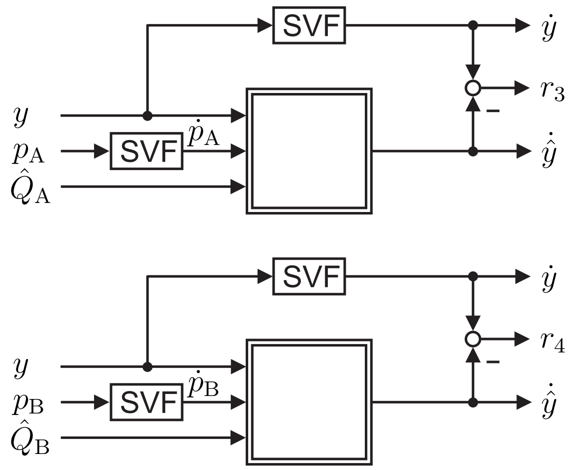

Fig. 5. Structure of residual r_3 and r_4.

Symptom r_5 is especially suited for detecting stuck faults of the piston.

In faultfree operation of the electro-hydraulic servo axis the residuals r_1, r_2, r_3 and r_4 show only small deflections caused by model uncertainty and measurement noise.

2.2 FAULT DIAGNOSIS

Table 1. Fault-Symptom table of the proportional valves (Beck et al. [2009]). The residuals are defined as $r_{x1} = y_x - \hat{y}_x$, $r_{x2} = I_{A_x} - \hat{I}_{A_x}$ and $r_{x3} = I_{B_x} - \hat{I}_{B_x}$, where y_x is the spool position of the xth valve and I_{A_x}, I_{B_x} are the coil currents.

Fault	r_{x1}	r_{x2}	r_{x3}
Partial winding short (coil A)	+	+	o
Partial winding short (coil B)	-	o	+
Current offset ΔI_A	+/-	+/-	o
Current offset ΔI_B	+/-	o	+/-
Overheated coil A	o	-	o
Overheated coil B	o	o	-
Blocked piston valve	+/-	o	o
Offset position sensor	+/-	o	o

$x \in 1, 2$	Valve 1,2
o	no deflection
+	positive deflection
-	negative deflection

Both fault symptom tables 1 and 2 (Beck et al. [2009]) show a causality between the induced faults and the reaction of the residuals. The goal of fault diagnosis is to derive the existence of faults from the observed symptoms. In general, there exist two different approaches for fault diagnosis. One is based on classification theory and the other on inference. The fault detection and diagnosis system described in this paper is based on a fuzzy-logic based diagnosis approach which allows to abandon the crisp separation of different fault states and uses a soft transition from one state to the other. The states are described by linguistic terms such as “reduced” or “increased”. The

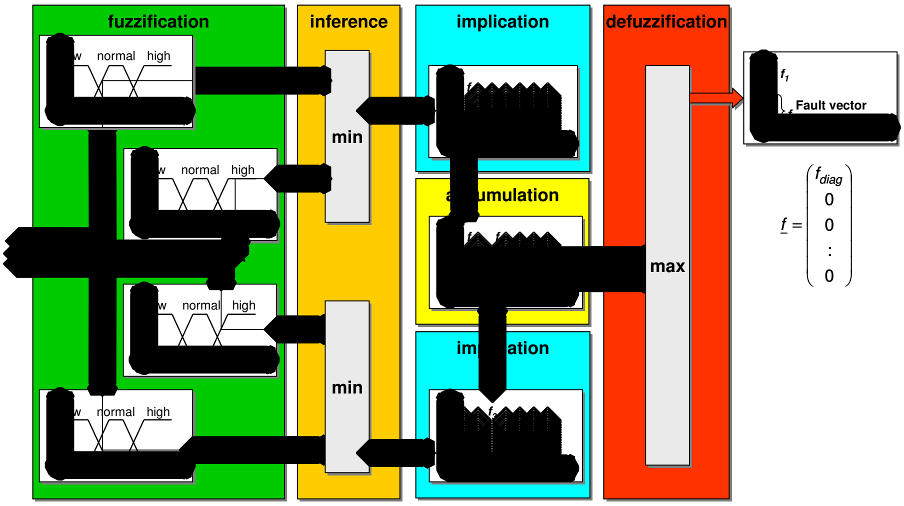

Fig. 6. Fuzzy-logic based system for fault diagnosis.

Table 2. Fault-Symptom table of the hydraulic system.

Fault	r_1	r_2	r_3	r_4	r_5
faultfree	o	o	o	o	o
pressure offset $\Delta p_A = 5\%$ (+/-)	+/-	o	+/-	o	o
pressure offset $\Delta p_B = 5\%$ (+/-)	o	+/-	o	-/+	o
internal leakage G_{AB}	+	-	+	+	o
piston blocked $y = cst.$	o	o	o	o	+
position sensor fault $y = cst.$	-	+	-	-	+

o no deflection
\+ positive deflection
\- negative deflection

heuristic knowledge about the causality between symptoms and faults is implemented to the diagnostic system by means of IF-THEN rules. The first step of the fuzzy-based diagnosis is the *fuzzification* (see Fig. 6) which comprises the process of transforming crisp values into grades of membership for linguistic terms. The membership function is used to associate a value to each linguistic term. The *inference* of the diagnosis system is implemented using the *Minimum*-operator while the assignments of the "'IF"' parts to the "'THEN"' parts of the "'IF-THEN"'-rules are implemented in the *implication* (see Fig. 6). The forth step of a fuzzy-based diagnosis is the *accumulation* where the results of the rules are combined. The last step of the fuzzy-diagnosis is the *defuzzification* which is implemented using the *Maximum*-operator.

For a detailed description of fuzzy-logic systems for fault diagnosis consider Isermann [2006]. Figure 6 shows the overall setup of the fuzzy-logic based diagnostic system.

3. FAULT ACCOMMODATION AND RECONFIGURATION

This section presents the fault accommodation and reconfiguration of the control loop of an electro-hydraulic servo-axis with a redundant duplex-valve-system. The fault accommodation for the electro-hydraulic servo axis is described in Beck [2010]. In the research project "'Fehlertoleranzstrategien fuer mechatronische Systeme"' (Beck [2010]) dynamic redundancy concepts with cold standby and hot standby and static redundancy concepts for the duplex-valve-system are investigated. Since a redundant valve is used, nearly all valve faults can be tolerated. If the redundant duplex-valve-system operates as a "'dynamic redundancy system"' the controller of the positioning control loop should be reconfigured subject to the operation of the duplex-valve-system and the detected and isolated faults. Without reconfiguration of the control loop in case of a valve failure the controller must be very conservative to achieve robustness. Hence, we need a control reconfiguration in order to achieve a good control performance even in case of valve faults. Since controller 1 is designed to operate with both valves being active, while controller 2 is designed to operate with only one valve being active, a switching between the controllers is required. The switching should provide a smooth transition of the actuating variables. A good overview of switching techniques for controllers can be found in Schwung [2007] and Zaccarian and Teel [2002]. Generally, hard switching and soft switching techniques are distinguished. In this paper a hard switching approach with bumpless transfer of the actuating variables is considered, which is based on tracking control (see Zaccarian and Teel [2002], Graebe and Ahlen [1996]). In order to avoid bumps to the process when switching between the outputs of an active controller (u_{active}) and inactive controller (u_{inactive}), the outputs of the controllers should be nearly the same. In industrial applications the position of the hydraulic piston is often controlled by a standard proportional plus integral plus derivative (PID) controller. Since the PID Controller is unstable according to the bibo- (bounded input bounded output) criterion, the inactive controller must be operated in closed loop in order to avoid drifting. The tracking control loops has two functions. The first function is to stabilize the inactive controller and the second function is to track the output of the inactive controller to the output of the active controller. In order to solve this

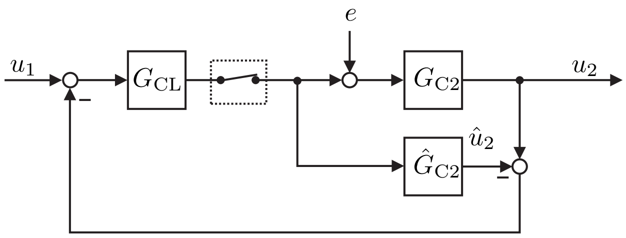

Fig. 7. IMC-structure used for the tracking control loop.

bumpless transfer problem the tracking control structure proposed by Graebe and Ahlen [1996] and the tracking control structure based on internal model control (IMC) were investigated in Beck [2010]. In direct comparison the IMC based tracking control shows better performance and easier design. For detailed information about internal model control consider Garcia and Morari [1982].

In case of the tracking control, the inactive controller G_{C2} represents the controlled system. If the controllers are implemented in digital hardware (e.g. Rapid-Control-Prototyping System) the parameters of the controllers are exactly known. Thus, the internal model $\hat{G}_{\mathrm{C2}}$ (see Fig. 7) accord with the controlled system and no model uncertainty must be considered during the design of the IMC controllers. In this paper only the switchover between two controllers is shown for clarity. However, the presented approach based on IMC-structures is also valid for stabilizing and tracking of any number of inactive controllers. Fig. 7 shows the IMC structure used for stabilization and tracking of the inactive controller G_{C2}. The command variable of this control loop is the output signal u_1 of the active controller (G_{C1}). The control deviation of the tracking loop is the difference between u_1 and u_2 and therefore the difference between the controller output signals. Since model and controlled system match exactly, the controller G_{CL} only acts if there exist disturbances (see Fig. 7). In case of the tracking control the main disturbance is the control derivation e of the superordinate position control loop.
The transfer function that describes the input-output behavior of the tracking loop is

$$G_{\mathrm{IMC2}} = \frac{u_2}{u_1} = \frac{G_{\mathrm{CL}} G_{\mathrm{C2}}}{1 - G_{\mathrm{CL}} \hat{G}_{\mathrm{C2}} + G_{\mathrm{CL}} G_{\mathrm{C2}}}. \tag{11}$$

In case of exact known controller parameters ($G_{\mathrm{C2}} = \hat{G}_{\mathrm{C2}}$) Equation (11) can be simplified to

$$G_{\mathrm{IMC2}} = G_{\mathrm{CL}} G_{\mathrm{C2}}. \tag{12}$$

Equation (12) points out, that the transfer function G_{IMC2} is equal to a pilot-control. The disturbance transfer function G_{NL} of the tracking control loop is

$$G_{\mathrm{NL}} = \frac{u_2}{e} = \frac{\left(1 - G_{\mathrm{CL}} \hat{G}_{\mathrm{C2}}\right) G_{\mathrm{C2}}}{1 - G_{\mathrm{CL}} \hat{G}_{\mathrm{C2}} + G_{\mathrm{CL}} G_{\mathrm{C2}}}. \tag{13}$$

Analog to the simplification of Equation (11) Equation (13) can be simplified to

$$G_{\mathrm{NL}} = \frac{u_2}{e} = \left(1 - G_{\mathrm{CL}} G_{\mathrm{C2}}\right) G_{\mathrm{C2}}. \tag{14}$$

The influence of the disturbance e (see Fig. 7) decays in time, if the static gain of the IMC-Controller G_{CL} is the inverse of the static gain of the controlled system (G_{C2}). Optimal tracking performance and best disturbance rejection is achieved if the IMC-Controller G_{CL} is chosen to be the inverse transfer function of the controlled system (G_{C2})

$$G_{\mathrm{CL}} = G_{\mathrm{C2}}^{-1}. \tag{15}$$

However, the inverse of the controlled system G_{C2}^{-1} is often not realizable. If the controller G_{C2} is minimum-phase, but has no direct feedthrough, the inverse does not exist. By expansion with a low pass G_{LP} filter of order n the IMC-Controller G_{CL} can be made realizable. Thus, the IMC-Controller can be described by

$$G_{\mathrm{CL}} = G_{\mathrm{C2}}^{-1} G_{\mathrm{LP}} = G_{\mathrm{C2}}^{-1} \frac{K_{\mathrm{LP}}}{(1 + T_1 s)^n}. \tag{16}$$

The low pass filter G_{LP} should be designed in dependance of the bandwidth of the input signals. If the bandwidth of the low pass filter is high enough and if the design rules for the IMC-Controller are considered, only small bumps occur after switching and the actuating variable shows a smooth characteristic.
If the IMC-Controller is designed according to Equation (16), the transfer function G_{IMC2} is

$$G_{\mathrm{IMC2}} = G_{\mathrm{C2}}^{-1} G_{\mathrm{LP}} G_{\mathrm{C2}} = G_{\mathrm{LP}}. \tag{17}$$

The disturbance transfer function G_{NL} is due to Equation (16)

$$G_{\mathrm{NL}} = \frac{u_2}{e} = \left(1 - G_{\mathrm{C2}}^{-1} G_{\mathrm{LP}} G_{\mathrm{C2}}\right) G_{\mathrm{C2}} = \left(1 - G_{\mathrm{LP}}\right) G_{\mathrm{C2}}. \tag{18}$$

A bidirectional switching between the controllers is possible, if a tracking control loop for the active controller is also considered. Analog to the design of the latent control loop with the controller G_{CL} an IMC-Controller G_{CA} for the active controller G_{C1} can be designed. After the switching to the formerly inactive controller this IMC-Controller stabilizes the formerly active controller. Figures 8 and 9 show the tracking control loops that are used for the bidirectional switching between the two controllers G_{C1} and G_{C2}. The bold lines point out the information flow from the input to the output of the control module. In Figure 8 the controller G_{C1} is the active controller while the controller G_{C2} is stabilized by the IMC-controller G_{CL}. Thus, the switch S_1 is open, the switch S_2 is closed and the toggle switch S_3 is in the upper position. In Figure 9 the controller G_{C2} represents the active controller while the controller G_{C1} is stabilized by the IMC-controller G_{CA}. Thus, the switch S_2 is open, the switch S_1 is closed and the toggle switch S_3 is in the lower position.

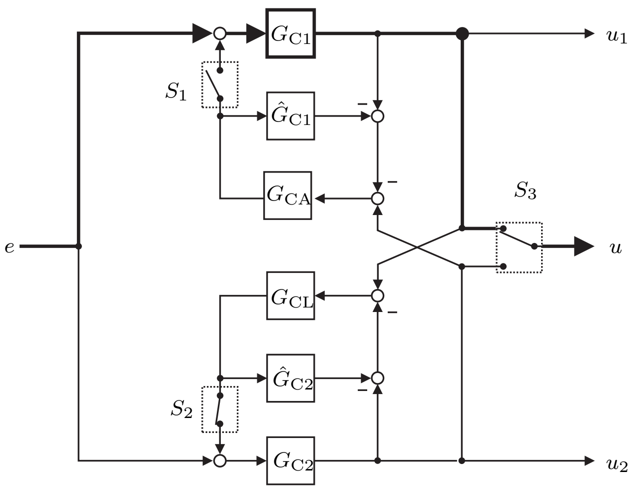

Fig. 8. Bumpless transfer among controllers using IMC-Structure (controller G_{C1} active, controller G_{C2} inactive).

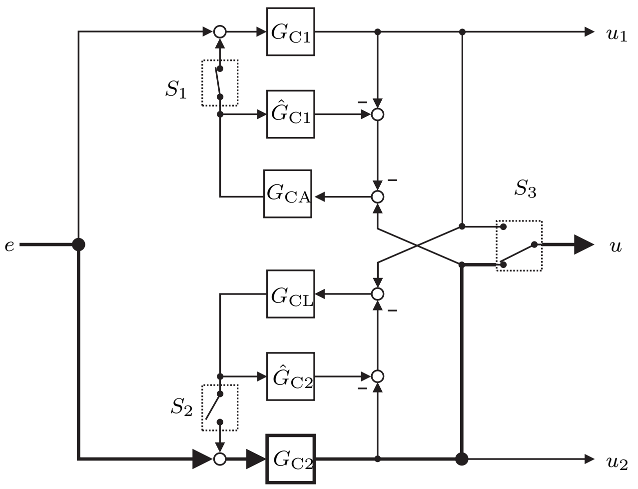

Fig. 9. Bumpless transfer among controllers using IMC-Structure (controller G_{C2} active, controller G_{C1} inactive).

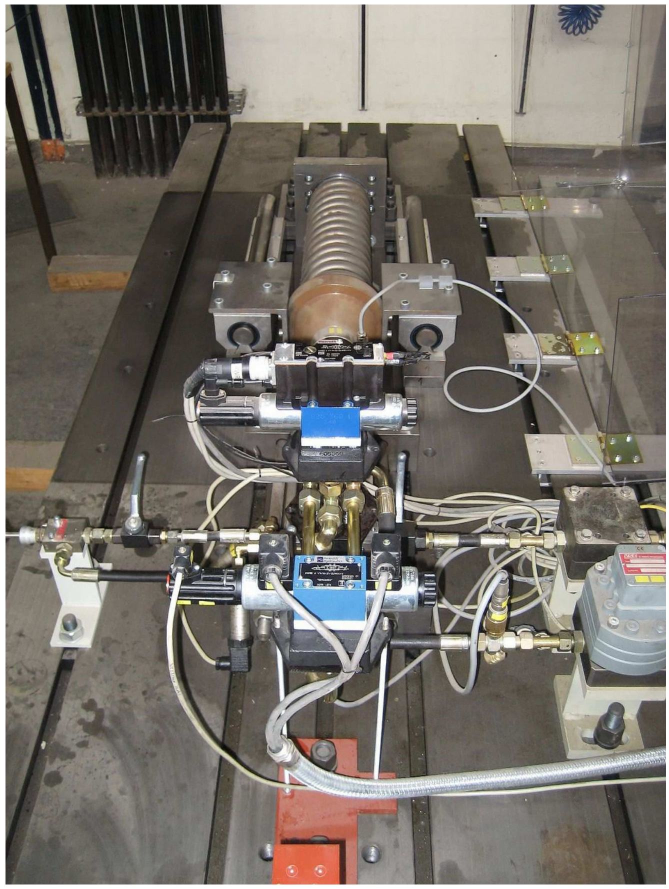

Fig. 10. Testbed with duplex-valve-system.

4. EXPERIMENTAL RESULTS

In this section the tracking performance of the proposed IMC structure and bumpless transfer between two controllers are presented. The two controllers are standard proportional plus integral plus derivative controllers with fixed parameters. The parameters of the controllers are obtained using common PID controller tuning methods (see Lunze [2008] and Lunze [2006]). The section closes with a comparison of passive and active fault tolerant control in the worst case of a failed valve. The experimental results have been obtained on a testbed depicted in Figure 10.

4.1 Verification of the tracking performance

In this subsection the tracking performance of the IMC structure is verified. The electro-hydraulic servo axis is operated in closed loop position control. The tracking performance should be verified using a highly dynamical signal with a large frequency bandwidth. The excitation must be high enough to create a considerably control deviation e in the superimposed control loop. The control deviation e of the position control loop represents a disturbance for the IMC tracking control loop (see Fig. 7: variable e). Thus, an amplitude-modulated pseudo random binary signal is used as command variable (see Fig. 2: variable w) for the piston position control loop. In faultfree operation both valves of the duplex-valve-system are active and identically actuated. Thus, the gains K_1 and K_2 (see Fig. 2) are the same. In faultfree operation controller G_{C1} is active and controller G_{C2} is inactive (see Fig. 8). Figure 11 shows the time plots of the command and controlled variable as well as the plots of the actuating variables in the faultfree case. In this case the output of the inactive controller u_2 is tracked to the output of the active controller u_1. Figure 12 shows the time plots of the signal u_1 and u_2 as well as the difference $u_1 - u_2$. The plots in Figure 12 and in particular the course of the difference illustrate the tracking performance of the IMC structure.

4.2 Comparison of passive and active FTC

In this subsection passive fault tolerant (no reconfiguration) and active fault tolerant control (reconfiguration of the control loop) are compared. Passive fault tolerance is achieved if the position controller is designed to operate with both valves being active as well as only one valve being active. In case of a valve failure the controller is capable to keep the system controllable and stable, but the controller performance is rather low. Figure 13 shows the time plots of the command and controlled variable (upper

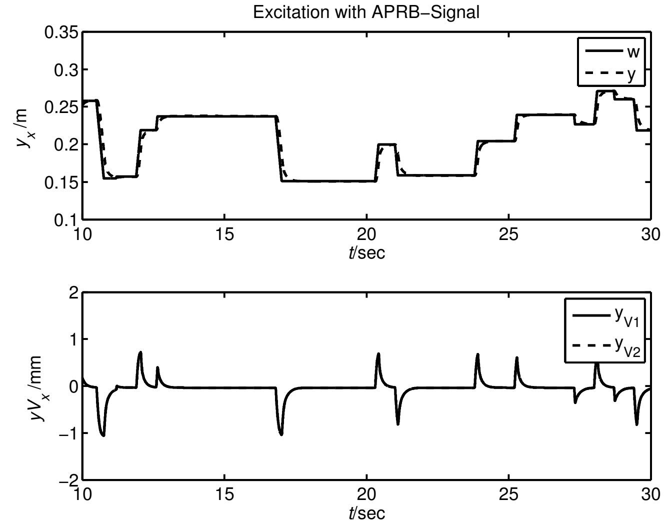

Fig. 11. Time plots of the command and controlled variable (upper figure) and of the actuating variables (lower figure) (faultfree case).

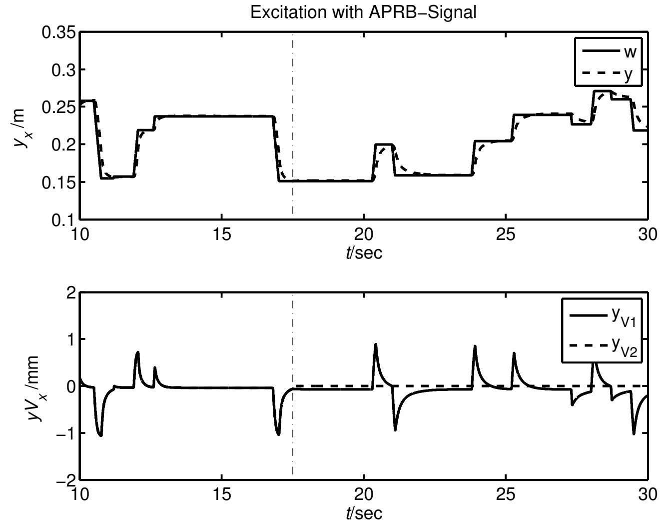

Fig. 13. Time plots of the command and controlled variable (upper figure) and of the actuating variables (lower figure) (valve 2 failed, no reconfiguration).

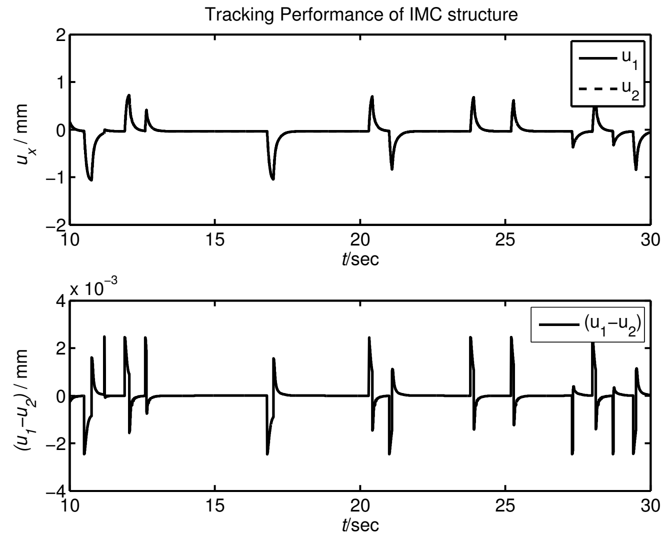

Fig. 12. Tracking Performance of IMC structure (faultfree).

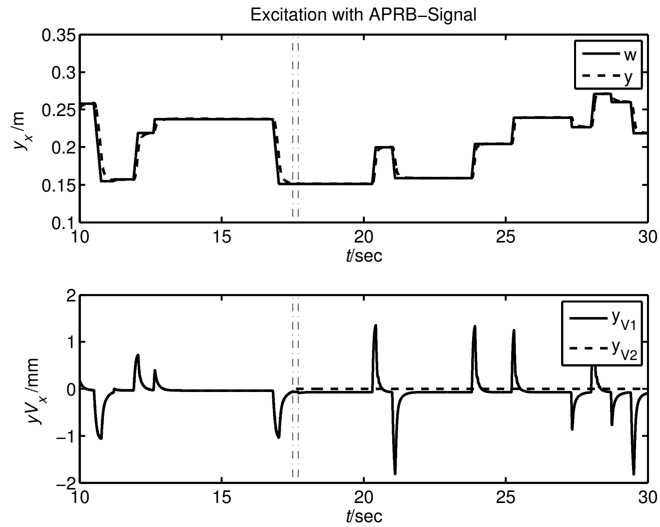

Fig. 14. Time plots of the command and controlled variable (upper figure) and of the actuating variables (lower figure) (valve 2 failed, with reconfiguration).

figure) and of the actuating variables (lower figure) in case of a sudden valve failure at time $t_1 = 17.5$sec (see Fig. 13: dash-dotted line). This fault has been injected by a simple shut down of the valve electronics. The plots of the command and controlled variable in Figure 13 illustrate the low controller performance after a valve failure. The strong centering springs of the valve push and hold the spool in the neutral position (see Fig. 13: $y_{V2} = 0 \; for \; t > t_1$)

The results of active fault tolerant control with bumpless transfer between two controllers is depicted in Figure 14. Figure 14 shows the time plots of the command and controlled variable (upper figure) and of the actuating variables (lower figure) in case of a sudden valve failure at time $t_1 = 17.5$sec (see Fig. 14: first dash-dotted line). The fault detection and isolation as well as the control reconfiguration is completed at time $t_2 = 17.7$sec (see Fig. 14: second dash-dotted line). After switching to the formerly inactive controller (G_{C2}), the formerly active controller (G_{C1}) is stabilized by the second IMC control loop. After switching the output of the inactive controller (u_1) is tracked to the output of the active controller (u_2). Figure 15 illustrates the high performance of the IMC tracking control loop.

In direct comparison, passive fault tolerant control (see plots in Fig. 13) provides a considerably lower controller performance than active fault tolerant control (see plots in Fig. 14). Thus, active fault tolerant control should be used for all practical purposes.

5. CONCLUSION AND OUTLOOK

In this paper process model based fault detection based on parity equations in combination with a fuzzy-logic diagnosis module are considered. Moreover, active fault tolerant control of an electro-hydraulic servo axis with

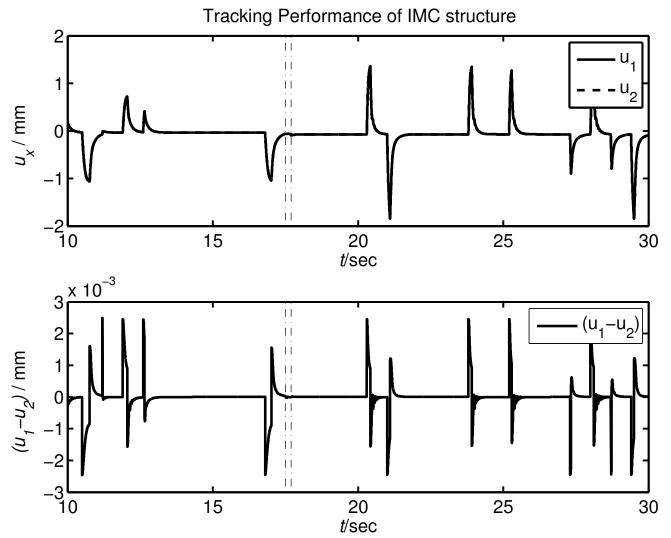

Fig. 15. Tracking performance of IMC structure in case of valve failures.

a redundant duplex-valve-system is presented. The IMC-structures for bumpless transfer between controllers allow a smooth transition and hence, the reconfiguration of the control loop in case of faults. The obtained results confirm the applicability of the proposed approach to a wide range of industrial applications.

The enhancements of active fault tolerant control as well as using adaptive control methods to design the inactive controller of the IMC-structure are topics of further research. Additionally, the approach presented in this paper will be applied to a redundant duplex-electrical-power-steering system.

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