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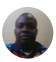
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A distributive approach for position control of clamps in a reconfigurable assembly fixture

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Abstract: Hydraulic actuator is a type of clamp used in a reconfigurable assembly fixture for exact positioning and effective immobilisation of workpiece during the assembly process. However, due to their nonlinearity, there is a need to design a control system for their effective performance. This study presents a distributive approach to mathematical modelling and position control of multiple hydraulic actuators used as a clamping system in a reconfigurable assembly fixture. The electrohydraulic system is verified experimentally in order to observe the synchronisation of the hydraulic actuators. The mathematical model of the system is developed in the Simulink environment. A Simulink model of the system is developed from the mathematical model and simulated with a fuzzy-PID controller in order to obtain the response of all the actuators and other operating characteristics of the system. Simulation results are shown graphically in order to verify the theoretical development.

Keywords: electrohydraulic system; EHS; position control; fuzzy-PID controller; hydraulic actuator; reconfigurable assembly fixture; RAF; distributive modelling.

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1 Introduction

Industrial machinery such as reconfigurable assembly fixture (RAF) achieve mobility of its parts with multiple hydraulic actuators (Olabanji and Mpofu, 2014a, 2014b; Nee et al., 2004). Position control of hydraulic actuator is necessary in order to obtain satisfactory performance during operation. This is usually realised by electrohydraulic systems (EHSs) (Olabanji et al., 2015). As a result of intrinsic nonlinear features of hydraulic systems, it is important to design an effective control system whenever they are used as precision equipment or any engineering application. EHSs are controlled by pump action and valve sequencing. The valve controlled systems are suitable for controlling the direction of flow, and actuator positions (Merritt, 1967; Andrew, 2011). Several mathematical models to control the position of a hydraulic actuator have been developed using a proportional control valve for each actuator in the system (Sun and Chiu, 2001). This article presents a distributive mathematical model to control the positions of multiple actuators when using one directional control valve for all the actuators.

In order to obtain an effective position control of hydraulic actuators in an EHS, feedback controllers are used to ensure that the actual displacement is equal to the desired displacement (Földi et al., 2011; Has et al., 2014; Shailaja et al., 2013). The PID is perhaps the most basic form of feedback control, which is usually used for primary control action (Åström and Hägglund, 2001; Liu and Daley, 2000; Matousek et al., 2011). It is less expensive, popular and highly used in process control industries. PID controllers are usually used due to its straightforwardness, and ease of operation (Madady, 2013; Farahani and Ganjefar, 2012; Çetin and Akkaya, 2010; Alagoz et al., 2013; Fadaei and Salahshoor, 2011; Zhao et al., 2013). However, considering the operation of the RAF, the conventional PID cannot provide an effective set point tracking and stability. The operation of the RAF requires that the travel length of the actuators changes due to the varying dimensions of the workpiece. In view of this, a fuzzy-PID controller is used to provide some level of artificial intelligence. The self-tuning ability of fuzzy-PID controller is employed to provide an adaptable position control to dynamic inputs to the system.

In essence, this article presents a position control system for multiple hydraulic actuators in an EHS using a distributive approach modelling and a fuzzy-PID controller. The rest of the paper is structured as follows. In the next section, a brief description of the system is presented. In Section 3, mathematical modelling of the system is developed and the simulation of the system is presented in Section 4 alongside the results obtained from the simulation. Finally, conclusions are made based on the results obtained from the simulation.

2 System description and experimental verification of the EHS

A description of the RAF and its components is presented in Figure 1. The EHS is made up of two subsystems. Subsystem one comprises of four hydraulic actuators (called the finger cylinders – FC), eight three-way flow control valves, eight pressure reducing valves, and a directional control valve. Subsystem two has two hydraulic actuators (called the movable frame cylinders – MFC), four three-way flow control valves, four pressure reducing valves, and a directional control valve. It is desired to control the positions of the actuators in each subsystems such that their displacement is synchronised for effective positioning of the fingers in the RAF (Olabanji, 2015). The fingers are responsible for gripping and immobilising the frame of press brakes during assembly process. The EHS was experimented using FESTO didactic hydraulic test bench in order to ascertain if the extensions and retractions of the actuators can be controlled (Figure 2). The observation from the experiment indicated that the actuators are synchronised and as such their positions can be controlled using a controller. The control architecture of the EHS used for the RAF is presented in Figure 3. This is necessary in order to create a line of thinking for the system model. The input to the EHS is the travel length of the actuators, which is obtained from a reconfiguration model that transforms the dimensions of the press brake frame to actuator travel length. The controller is expected to control the system through the feedback from the linear variable displacement transducers.

Figure 1 Schematic diagram of the RAF showing all hydraulic subsystems (see online version for colours)

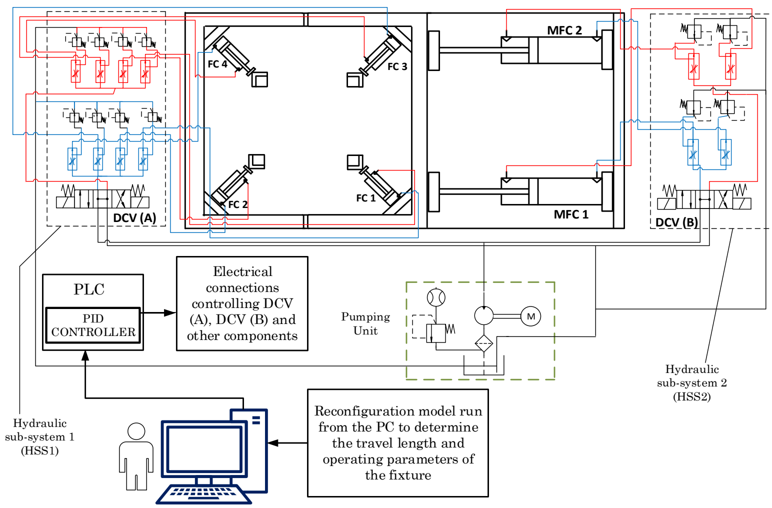


Figure 2 Experimental investigation of the EHS (see online version for colours)

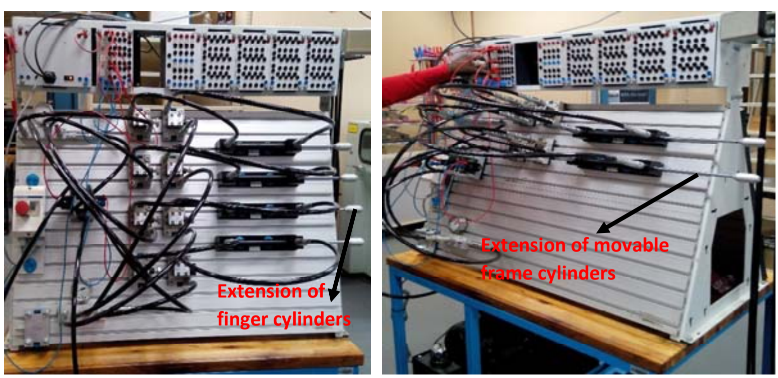
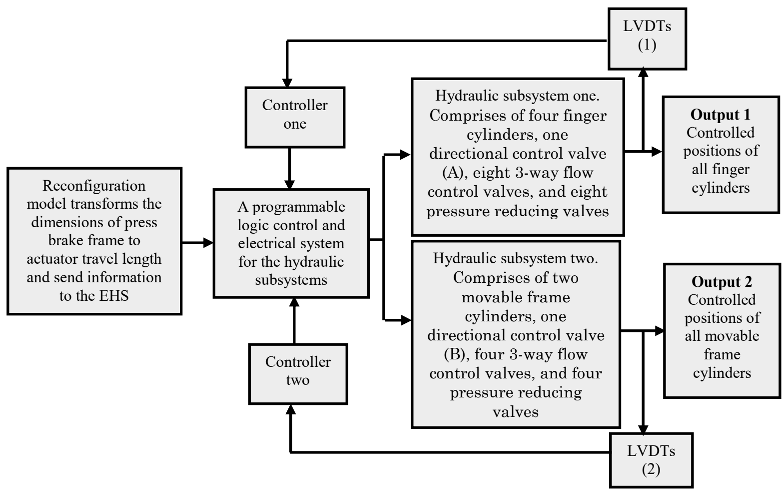


Figure 3 Control architecture of the RAF



3 Mathematical modelling of the EHS

The distributive approach is based on the assumption that each identical actuator in a subsystem has different control ports as shown in Figure 4. The control ports are expected to have uniform flow conditions for all identical actuator. In essence, the pressure, flow, and velocity of the fluid in all the identical actuators are unvarying. The objective of the control is to regulate the positions of all identical actuators in the same subsystem to the desired position (Kalyoncu and Haydim, 2009; Poley, 2005; Šitum, 2011).

In order to simplify the analysis, the following assumptions are necessary in the modelling:

1 At relaxed state of the system, the spool displacements of the directional control valves, the piston displacements of the actuators and the induced pressure in the control ports of all identical actuator are equal to zero,

$$X_{FC_i}\big|_{i=1-4} = X_{MFC_m}\big|_{m=1-2} = U_s\big|_{s=1,2} = P_{IP_s}\big|_{s=1,2} = 0 \tag{1}$$

where U and P_{IP} are the spool displacement of the directional control valves, and the induced pressure in the control ports of the actuators respectively, while X_{FC} and X_{MFC} are the displacement of the FC and MFC.

2 The difference between the summation of the leakage pressure entering and leaving all identical actuators in the system is zero at equilibrium state.

$$\Delta P_{le_s}\Big|_{s=1,2} = \sum_{i=1}^{i=n}\left\{\left(P_{su} - P_{A_i}\right) - \left(P_{A_i} - P_{re}\right)\right\} - \left\{\left(P_{su} - P_{B_i}\right) - \left(P_{B_i} - P_{re}\right)\right\}_s\Big|_{s=1,2} = 0 \quad (2)$$

where P_A and P_B are the pressures in the piston and annulus side respectively, while P_{su} and P_{re} are the supply pressure and return pressure respectively and 'n' is the number of actuators in each subsystem 's'.

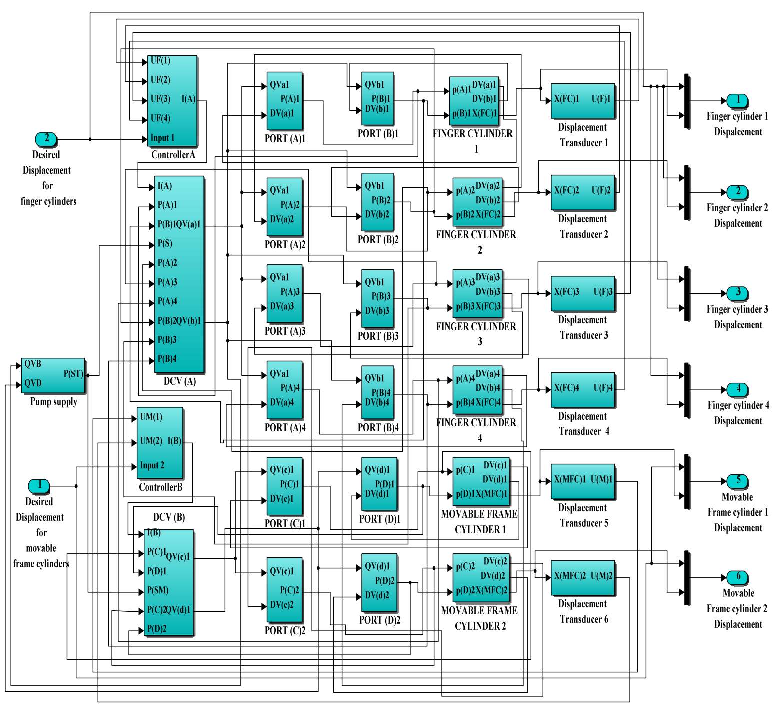
3 The dimensions (such as area, volume, and travel length) of all identical actuators is uniform, and the total is a summation of individual dimensions.

4 For any small perturbation around the relaxed state, the volume of hydraulic fluid as a result of the piston displacement is far lesser than the initial volume of hydraulic fluid in the chambers of the actuator.

$$A_{FC_i}.X_{FC_i} \ll V_{FC_i}\Big|_{i=1-4};\ A_{MFC_m}.X_{MFC_m} \ll V_{MFC_m}\Big|_{m=1-2} \quad (3)$$

where A_{FC} and A_{MFC} are the areas of the FC and MFC respectively, while V_{FC} and V_{MFC} are the initial volume of hydraulic fluid in the FC and MFC respectively.

Figure 4 Mathematical model of the EHS in Simulink (see online version for colours)



Considering Figure 4, as the fluid leaves the directional control valves it is equally distributed into the ports of all identical actuators in the same subsystem. The equal distribution is achieved by the three-way flow control valves. The pressure in the ports are kept constant by the pressure regulating valves allocated to each port. This situation is the same for the annulus side of the actuators. It is desired to determine the mathematical model for each stage in the system in order to obtain the mathematical model of the whole system.

3.1 Directional control valves

The inductance, current and resistance is a function of the voltage across the terminals of the solenoid. Since the flux linkage is also a function of the current and plunger spool displacement, which depends on the electromagnetic force created by the solenoid (Zięba, 2003), then the dynamic equation of the plunger spool assembly can be represented by its acceleration (Hosseini et al., 2013; Shailaja et al., 2013; Kalyoncu and Haydim, 2009). If the two subsystems are represented by '*s*', then the acceleration of the spool can be represented by (4).

$$\left.\frac{\partial^2 U_s}{\partial t^2}\right|_{s=1,2} = \frac{1}{m_s}\left.\left\{\frac{K_{bs} I_s}{2} - d_{cts}\dot{U}_s - U_s\left(F_{rs} + K_{cs}\right)\right\}\right|_{s=1,2} \tag{4}$$

where m_s, K_b, K_c, d_{ct}, F_r are the mass of the spool, back e.m.f of the solenoid, spring constant, damping coefficient, and force spring rate of the spool respectively.

In order to maintain equilibrium at the relaxed state, the leakage flow and it is accompany pressures must be kept constant (Assumption 2: $\Delta P_{le} = 0$). If the number of actuators in each subsystem is represented by n (where $n = 4$ in subsystem one and $n = 2$ in subsystem two), then the leakage pressure of an actuator in subsystem one must be equal to a quarter of the total leakage pressure and half the total leakage pressure for subsystem two. This condition is important in order to obtain an effective position control of identical actuators in the RAF as depicted in Figure 1. A slight displacement of the spool causes an infinitesimal change from the relaxed state, hence $\Delta P_{le} > 0$. The differential leakage pressure attains an induced pressure (P_{IP}) as a result of the differential pressures in the pistons (ϑP_A) and annulus (ϑP_B) ports. This can be represented by (5).

$$\left.\Delta P_{le_s}\right|_{s=1,2} = \left.\sum_{i=1}^{i=n}\left(\partial P_{A_i} - \partial P_{B_i}\right)_s\right|_{s=1,2} = \left.P_{IP_s}\right|_{s=1,2} \tag{5}$$

The flow at the ports of the directional control valves can be related to an orifice flow [with a coefficient of discharge (C_d)] in which the gradual opening of the ports (w) is a function of the spool displacement and net pressure across the ports of the valve. The net pressure in this case is a function of the supplied pressure (P_s), return pressure (P_R) and leakage pressure. This can be represented by the nonlinear equation of flow considering the differential pressures at the inlet of the piston and annulus sides of the actuators as presented in equation (6) (Kalyoncu and Haydim, 2009; Shailaja et al., 2013; Li and Thurner, 2013).

$$Q_s\big|_{s=1,2} = C_d w_s U_s \left[\frac{P_{S_s} - P_R - \left(\sum_{i=1}^{i=n} \partial P_{A_i} - \partial P_{B_i} \right)}{\rho} \right]^{0.5} \Bigg|_{s=1,2} \tag{6}$$

In essence for any small perturbation around the relaxed state, the linearised equation of flow for the directional control valves in both subsystems can be represented by (7);

$$Q_s\big|_{s=1,2} = \frac{\partial f}{\partial U_s}\bigg|_{U_s=0} U_s - \frac{\partial f}{\partial P_{IP_s}}\bigg|_{P_{IP_s}=0} \sum_{i=1}^{i=n} \left(\partial P_{A_i} - \partial P_{B_i} \right)_s \Big|_{s=1,2} \tag{7}$$

In essence, the linearised equation of flow at relaxed state $(U_s = P_{IP_s} = 0)$ from the directional control valves to the ports of the actuators can be represented by (8) (Li and Thurner, 2013; Merritt, 1967; Alleyne and Liu, 2000; Liu and Daley, 2000).

$$Q_s\big|_{s=1,2} = K_{a_s} U_s - K_{b_s} P_{IP_s}\big|_{s=1,2} \tag{8}$$

where K_a and K_b are the flow gain and flow pressure coefficients for the directional control valves.

3.2 Control ports and the actuators

As the directional control valves allow the flow of hydraulic fluid into the subsystems, it is expected that the pressure and volume of hydraulic fluid in the ports of all identical actuators will increase uniformly (Assumption 3). The control volume of all identical actuator ports is a function of the piston effective area, actuator displacement and initial volume of hydraulic fluid in the chambers of the actuator. In essence, considering the effect of leakage in the actuators (K_I and K_E), the flow in the piston and annulus ports of the actuators in each subsystem can be represented by the continuity and state equations (9)–(10) (Kalyoncu and Haydim, 2009; Has et al., 2014, 2015; Guan and Pan, 2008; Zhao et al., 2008).

$$Q_{su_s} - \sum_{i=1}^{i=n} \left(K_{I_i} \left(P_{A_i} - P_{B_i} \right) - K_{E_i} P_{A_i} \right) = \sum_{i=1}^{i=n} \left(\frac{\partial V_{A_i}}{\partial t} + \frac{V_{A_i}}{\beta} \frac{\partial P_{A_i}}{\partial t} \right) \Bigg|_{s=1,2} \tag{9}$$

$$-Q_{re_s} + \sum_{i=1}^{i=n} \left(K_{I_i} \left(P_{A_i} - P_{B_i} \right) - K_{E_i} P_{A_i} \right) = \sum_{i=1}^{i=n} \left(\frac{\partial V_{B_i}}{\partial t} + \frac{V_{B_i}}{\beta} \frac{\partial P_{B_i}}{\partial t} \right) \Bigg|_{s=1,2} \tag{10}$$

As stated earlier, if the volume of fluid in the piston and annulus ports of the cylinders (V_A and V_B respectively) is a function of the piston area (A_{FC} and A_{MFC} for FC and MFC respectively), the initial volume of fluid in the ports and piston displacement (X_{FC} and X_{MFC}), then subtracting the continuity equations of the annulus ports from the piston ports and obtaining the net pressure across the ports from the differential pressures will yield the mean flow rate of hydraulic fluid to each cylinders in the subsystems. Furthermore, it is evident that the displacement of the pistons in all identical actuators is far lesser than the initial volume of hydraulic fluid in the chambers for a small perturbation around the relaxed state (Assumption 4). If the number of cylinders in subsystems one and two are represented by i and m respectively, then this mean flow rate can be represented by equations (11) and (12).

$$Q_s\Big|_{s=1} = \sum_{i=1}^{i=4}\left(A_{FC_i}\frac{\partial X_{FC_i}}{\partial t} + \frac{V_{FC_i}}{2\beta}\frac{\partial P_{IP_s}}{\partial t} + K_{FC_i}P_{IP_s}\right)\Bigg|_{s=1} \tag{11}$$

$$Q_s\Big|_{s=2} = \sum_{m=1}^{m=2}\left(A_{MFC_m}\frac{\partial X_{MFC_m}}{\partial t} + \frac{V_{MFC_m}}{2\beta}\frac{\partial P_{IP_s}}{\partial t} + K_{MFC_m}P_{IP_s}\right)\Bigg|_{s=2} \tag{12}$$

where K_{FC} and K_{MFC} are the leakage flow coefficient in each FC and MFC respectively.

Considering assumption one, it is obvious that the terms $A_{FC_i}\frac{\partial X_{FC_i}}{\partial t}$ and $A_{MFC_m}\frac{\partial X_{MFC_m}}{\partial t}$ becomes zero at the equilibrium state. In essence, the linearised equation for the differential pressure of the hydraulic fluid in the control ports of the actuators in subsystem one and two can be represented by (13) and (14) respectively.

$$\frac{\partial P_{IP_s}}{\partial t}\Bigg|_{s=1} = \left[\frac{2\beta}{V_{FC_i}}\left(K_{a_s}U_s - K_{b_s}P_{IP_s} - K_{FC_i}P_{IP_s}\right)\right]\Bigg|_{i=1-4}\Bigg|_{s=1} \tag{13}$$

$$\frac{\partial P_{IP_s}}{\partial t}\Bigg|_{s=2} = \left[\frac{2\beta}{V_{MFC_m}}\left(K_{a_s}U_s - K_{b_s}P_{IP_s} - K_{MFC_m}P_{IP_s}\right)\right]\Bigg|_{m=1-2}\Bigg|_{s=2} \tag{14}$$

Considering equations (13) and (14), it is evident that the directional control valves will supply the net pressures across the ports of the cylinders in each subsystem in the form of a transfer function as presented in equations (15) and (16).

$$\frac{P_{IP_s}(s)}{U_s(s)} = \frac{2K_{a_s}\beta}{\left\{V_{FC_i}(s) + 2\beta\left(K_{FC_i} + K_{b_s}\right)\right\}\Big|_{i=1-4}}\Bigg|_{s=1} \tag{15}$$

$$\frac{P_{IP_s}(s)}{U_s(s)} = \frac{2K_{a_s}\beta}{\left\{V_{MFC_m}(s) + 2\beta\left(K_{MFC_m} + K_{b_s}\right)\right\}\Big|_{m=1-2}}\Bigg|_{s=2} \tag{16}$$

In order to create an accelerated force on the piston of the actuators, the net differential pressure acting on the piston must overcome the inertia and stiffness of the load and the viscous friction of the pistons in all the cylinders. This net pressure is distributed across the ports of all identical actuators in each subsystem. In view of this, since the net force on the pistons is a function of the acceleration force, stiffness force and the force due to viscous friction, then the acceleration of each actuator in subsystems one and two can be represented by (17) and (18) respectively.

$$\frac{\partial^2 X_{FC_i}}{\partial t^2}\Bigg|_{i=1-4} = \frac{1}{M_{f_i}}\left(A_{FC_i}P_{IP_s}\Big|_{s=1} - b_{v_i}X_{FC_i} - K_{e_i}X_{FC_i}\right)\Bigg|_{i=1-4} \tag{17}$$

$$\left.\frac{\partial^2 X_{MFC_m}}{\partial t^2}\right|_{m=1-2} = \frac{2}{M_{mf_m}}\left(A_{MFC_m} P_{IP_s}\Big|_{s=2} - b_{v_m} X_{MFC_m} - K_{e_m} X_{MFC_m}\right)\Bigg|_{m=1-2} \tag{18}$$

3.3 System model

It is necessary to develop a mathematical representation of the two hydraulic subsystems in transfer functions or state space. The state and output equations of subsystems one and two can be represented by (19) and (20), respectively. Let A_1, B_1, C_1 and D_1 denote the system matrix, control matrix, output matrix and feed forward matrix of subsystem one respectively. Similarly, let A_2, B_2, C_2 and D_2 denote the system matrix, control matrix, output matrix and feed forward matrix of subsystem two respectively. Considering (4), (13)–(14), and (17)–(18), then A_1, B_1, C_1, D_1, A_2, B_2, C_2 and D_2 are presented in (21)–(28). The transfer functions can also be obtained from (4), (13)–(14), and (17)–(18).

$$\begin{aligned} \dot{x}_1 &= A_1 x_1 + B_1 u \\ y_1 &= C_1 x_1 + D_1 u \end{aligned} \tag{19}$$

$$\begin{aligned} \dot{x}_2 &= A_2 x_2 + B_2 u \\ y_2 &= C_2 x_2 + D_2 u \end{aligned} \tag{20}$$

where x and y are the state and output equations respectively. Also, x_1 and x_2 are state variables representing the spool displacement for subsystems one and two respectively.

$$A_1 = \begin{bmatrix} 0 & 1 & 0 & 0 & 0 \\ \frac{-\left(K_{c_1} + F_n\right)}{m_1} & \frac{d_{ct_1}}{m_1} & 0 & 0 & 0 \\ \left.\frac{2\beta K_{a_1}}{V_{FC_i}}\right|_{i=1-4} & 0 & \left.\frac{-2\beta}{V_{FC_i}}\right|_{i=1-4} & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 \\ 0 & 0 & \left.\frac{A_{FC_i}}{M_f}\right|_{i=1-4} & \left.\frac{-K_{e_i}}{M_f}\right|_{i=1-4} & \left.\frac{-b_{v_i}}{M_f}\right|_{i=1-4} \end{bmatrix} \tag{21}$$

$$B_1 = \begin{bmatrix} 0 \\ \frac{K_{b_1} I_1}{2m_1} \\ 0 \\ 0 \\ 0 \end{bmatrix} \tag{22}$$

$$C_1 = [1 \quad 0 \quad 0 \quad 0 \quad 0] \tag{23}$$

$$D_1 = 0 \tag{24}$$

$$A_2 = \begin{bmatrix} 0 & 1 & 0 & 0 & 0 \\ \dfrac{-\left(K_{c2} + F_{r2}\right)}{m_2} & \dfrac{d_{ct2}}{m_2} & 0 & 0 & 0 \\ \left.\dfrac{2\beta K_{a2}}{V_{MFC_m}}\right|_{m=1-2} & 0 & \left.\dfrac{-2\beta}{V_{MFC_m}}\right|_{m=1-2} & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 \\ 0 & 0 & \left.\dfrac{2A_{MFC_m}}{M_{mf}}\right|_{m=1-2} & \left.\dfrac{-2K_{e_m}}{M_{mf}}\right|_{m=1-2} & \left.\dfrac{-2b_{vm}}{M_{mf}}\right|_{m=1-2} \end{bmatrix} \tag{25}$$

$$B_2 = \begin{bmatrix} 0 \\ \dfrac{K_{b_2} I_2}{2m_2} \\ 0 \\ 0 \\ 0 \end{bmatrix} \tag{26}$$

$$C_2 = [1 \quad 0 \quad 0 \quad 0 \quad 0] \tag{27}$$

$$D_2 = 0 \tag{28}$$

4 Simulation of the EHS

In order to control the modelled system, the electrohydraulic subsystems were developed in Simulink environment using a fuzzy-PID controller. The structure of the self-tuning fuzzy PID controller and the EHS is presented in Figure 5. In order to obtain the critical gain and ultimate period of the two subsystems, their step response was obtained without a controller. The critical gain and period were used to obtain the gains of the PID controller using Nichole Ziegler tuning method (Matousek et al., 2011). The objective of the simulation is to control and synchronise the positions of all identical actuators in the subsystems such that a slight displacement of the valve spool will be captured in order to obtain its effect on the displacements of the actuators. In order to achieve this objective, the Mamdani model is applied as the structure of the fuzzy inference system. Seven linguistics variables are used to represent the input and output characteristics of the valve spool and the hydraulic actuators respectively. The range of values for the input and output membership functions are obtained from initial values of the PID gains. The proportional, integral and derivative gains are the outputs of the fuzzy inference system while the error and derivative of the error are the inputs (Adnan et al., 2011; Çetin and Akkaya, 2010; Kalyoncu and Haydim, 2009). The seven linguistic variable levels used to represent the membership functions are large negative (LN), medium negative (MN), small negative (SN), zero (ZE), small positive (SP), medium positive (MP) and large positive (LP). These variables are used for both inputs and outputs of the fuzzy inference system. The fuzzy rules for the inference system are presented in Table 1. The variables and rules are adopted for both electrohydraulic subsystems. However, the values of the variables used in each subsystem differs. The Simulink model for electrohydraulic subsystem one and two are presented in Figures 6 and 7, respectively. In essence, the controller is expected to improve the overall system performance characteristics and response of the actuators by tracking the reference input. It is also anticipated that the response of all identical actuators in the same subsystem should be uniform in order to obtain position synchronisation.

Table 1 Fuzzy inference rules for the controller outputs

e/de	*LN*	*MN*	*SN*	*ZE*	*SP*	*MP*	*LP*
LN	LN LP LN	LN LP LN	LN LP LN	LN LP LN	MN MN SP	SN SN MP	ZE LN LP
MN	LN LP LN	MN MP MN	MN SP MN	MN ZE MN	MN SN SN	SN MN ZE	ZE LN LP
SN	LN MP LN	MN SP MN	SN SP SN	SN SP SN	SN ZE ZE	ZE SN MP	ZE MN LP
ZE	LN LP LN	MN MP MN	SN SP SN	ZE ZE ZE	SP SN SP	MP MN MP	LP LN LP
SP	MN ZE MN	SN SP MN	SP SP SN	SP SN SP	SP SN SP	MP MN MP	LP LN LP
MP	ZE ZE MN	SP MP SN	SP MP ZE	MP MN SP	MP MN MP	MP LN MP	LP LN LP
LP	ZE ZE ZE	SP MN MP	MP SN SP	LP LN LP	LP LN LP	LP LN LP	LP LN LP

Note: KP
KI KD

Figure 5 Structure of the controller and the EHS (see online version for colours)

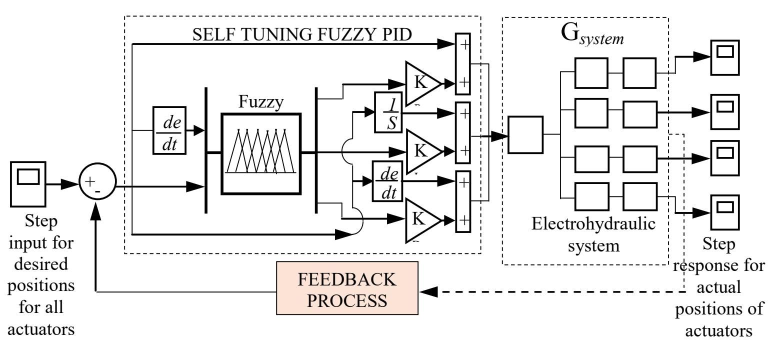


Figure 6 Simulink model for subsystem one (see online version for colours)

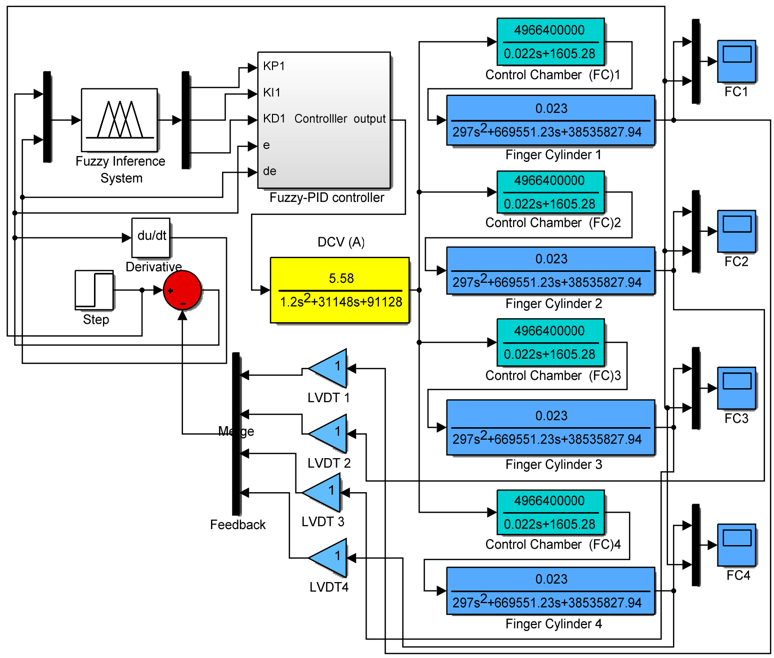
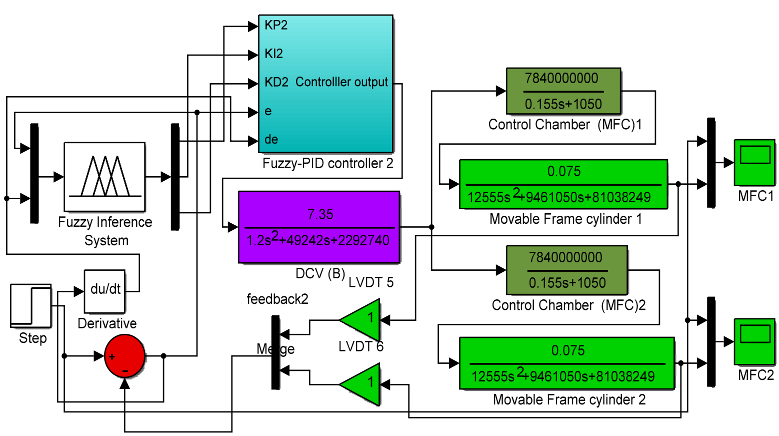


Figure 7 Simulink model for subsystem two (see online version for colours)



5 Simulation results and discussion

The results obtained from the simulation are presented in this section. The controller gains and system performance characteristics for both subsystems are presented in Table 2. As stated earlier, the goal of the controller is to ensure that the actuators track the reference or desired position. In order to achieve this, it is necessary to subject the modelled system to a step input and the output of all the actuators in each subsystem will be tracked with the input. The output response of the actuators in subsystems one and two when subjected to a step input, are presented in Figures 8 and 9, respectively. As presented in Table 2, it can be observed that the controller gains in subsystem one is higher than subsystem two for both controllers. The reason for the increased value of controller gains in subsystem one may be due to the large number of hydraulic actuators and control ports compare to subsystem two. In addition, the large number of hydraulic components in subsystem one may have caused the increase in percentage overshoot and peak amplitude of the subsystem compare to subsystem two where the percentage overshoot and peak amplitude is lesser. In essence, it can be deduced that the system with more hydraulic components will have large controller gains which will tentatively increase its peak amplitude and percentage overshoot. This is possible because increase in proportional gain increases the peak value of the system. Conversely, despite the increased percentage overshoot and peak value in subsystem one, the controller was able to achieve lesser peak time, rise time and settling time compared to subsystem two. This depicts that subsystem one has a fast response compare to subsystem two. The difference in response time may be due to large size of the hydraulic actuators in subsystem two and also the large weight pushed by the actuators. Furthermore, the gains obtained for the PID controller are higher than the gains obtained for the fuzzy-PID controller. This is an indication that the fuzzy-PID controller gives better performance than the PID controller comparing the values obtained for the system characteristics performance using the two controllers. In essence, it can be deduced that, increase in number of actuators in a multiple hydraulic actuating system, will cause the values of the percentage overshoot and peak amplitude to increase. Also, if the size of the actuators in the system is large or if the weight pushed is large, there is an indication that the peak time, rise time and settling time will be high.

Table 2 Results obtained from the simulation

Electrohydraulic subsystems	*Controller gains and system characteristics performance*	*PID controller*	*Fuzzy-PID controller*
Subsystem 1	K_{P_1}	340,591,850	155,306,240
	K_{I_1}	1,900,341,423	905,628,755
	K_{D_1}	2,032,545	1,716,300
	Peak amplitude	1.16	1.06
	% overshoot	16.1	5.61
	Peak time (s)	0.0319	0.0276
	Rise time (s)	0.0144	0.0111
	Settling time (s)	0.0556	0.0542

Table 2 Results obtained from the simulation (continued)

Electrohydraulic subsystems	*Controller gains and system characteristics performance*	*PID controller*	*Fuzzy-PID controller*
Subsystem 2	K_{P_2}	10,205,747	10,143,869
	K_{I_2}	55,166,202	33,056,992
	K_{D_2}	472,016	471,123.5
	Peak amplitude	1.15	1.04
	% overshoot	14.6	4.49
	Peak time (s)	0.273	0.285
	Rise time (s)	0.121	0.138
	Settling time (s)	0.487	0.396

Figure 8 Step response of actuators in subsystem one (see online version for colours)

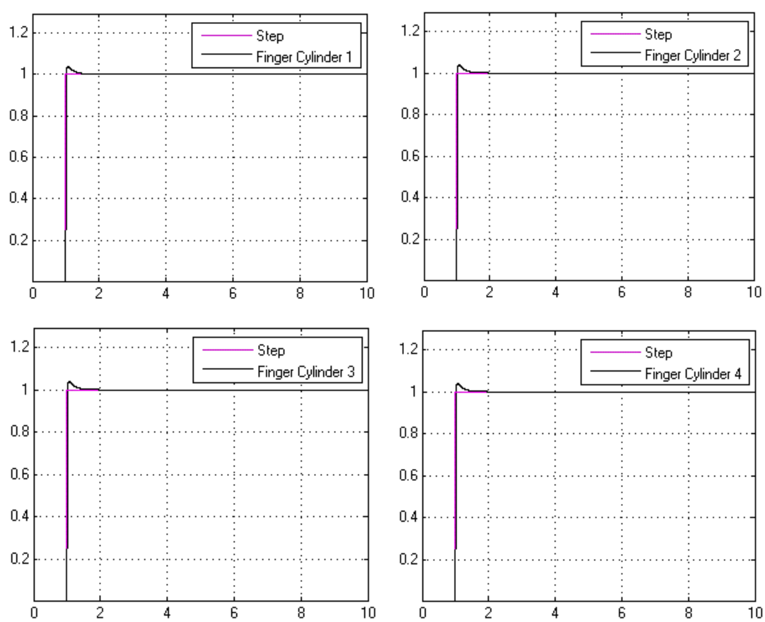
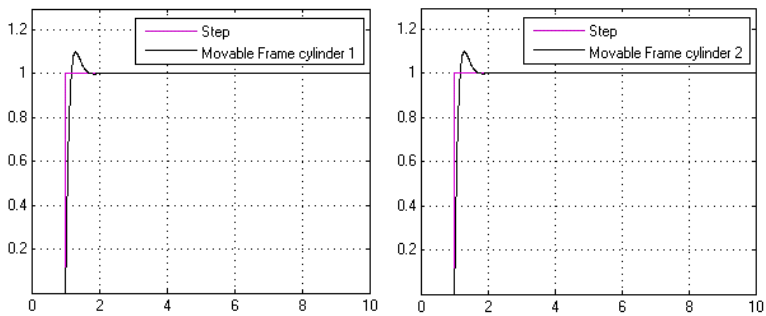


Figure 9 Step response of actuators in subsystem two (see online version for colours)



6 Conclusions

This paper describes the control of EHS for a RAF. It presents a study on position control of multi-hydraulic actuators using a distributive approach modelling and a fuzzy-PID controller. The EHS comprised of two hydraulic subsystems. Each electrohydraulic subsystem has more than one hydraulic actuators. These actuators are controlled by one directional control valve, three-way flow control valves and pressure reducing valves. A mathematical model is presented using a distributive approach. In order to obtain position tracking of the actuators with a single directional control valve, the modelled system is simulated in order to obtain its step response and characteristics. The controller achieved position control of all identical actuators in each subsystem, because it provides minimal settling time, good transient response and eliminate excessive overshoot. Also, the output response of all actuators in the same subsystem were able to track the reference signal and are uniform. In essence, the area of contribution of this article is that, position control of large number of hydraulic actuators connected to one directional control valve will give rise to increased values of percentage overshoot and peak amplitude. Also, the size of the actuators and the weight pushed indicates the response time of the system.

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