SIMULATION ANALYSIS OF INFLUENTIAL PARAMETERS EFFECTING THE HYDRAULIC PRESS BEHAVIOUR

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The most common purpose of hydraulic press is to withdraw the desired movement and to generate the force to overcome the movement restrictions. The dilemma is whether changes in component characteristics affect the hydraulic press's behaviour without exception. For this reason, this study investigates how alteration in influencing parameters affect the hydraulic press behaviour. Furthermore, the study considers change in characteristic parameters, such as: valve response time, proportional and integral control parameters, degree of control edge overlap or spool wear in hydraulic valve, extend of the dead volume in cylinder chambers, system pressure and friction characteristics in frame guides. Using a simulation modelling approach, an effective method is proposed to investigate the source of the change and its effect on the discrete hydraulic component. By monitoring the behaviour of the hydraulic press using virtual sensors, the most reflective behaviour of each hydraulic component is described using simulation analysis.

Keywords:
hydraulic press, simulation modelling, influential parameters, condition-based analysis, virtual sensors

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1 Intruduction

The hydraulic presses are widely used for numerous industrial as well as experimental and testing applications. Knowing the hydraulic system behaviour under different circumstances it is crucial to achieve optimal performance of the system. Thus, performing the real-time data analysis and control actions is needed. On the other hand, more and more digital models are used to perform the simulation and optimization of the real systems and processes. Moreover, the measured characteristics in simulation model are performed by the virtual sensors, that are required to develop a digital twins of hydraulic systems (DTHS). Such digital models represent the quasi-replicas of the real systems with the same functionality and behaviour and allows us to perform different working scenarios in advance or in parallel to the real processes. In this way a change in the components characteristics and their effects on the hydraulic system behaviour can be analysed more efficiently.

There are several authors investigating the influence of various causes on hydraulic system behaviour. The authors in [1] proposed acoustic signal-based fault detection of hydraulic piston pump. Researchers in [2] presented hydraulic control valve wear consequently affecting the hydraulic valve response time dynamics. Also, the trend of modern studies concentrates on fault recognition of the systems and leakage occurrence as presented in [3] and [4]. Additionally, researchers expose the importance of considering the nonlinearities in the simulation model resulting from friction dynamics [5], [6], hydraulic oil characteristics [7], hydraulic valve flow and pressure dynamics [8].

Therefore, the paper represents a thorough analysis of the chosen influential parameters of hydraulic components, which can affect the system behaviour. The focus of the investigation is to perform what-if scenarios by using the simulation approach, which allows us to validate and confirm the influential parameters and their minor or major effects on hydraulic system behaviour.

2 The concept of hydraulic system

The concept of servo hydraulic press is proposed as the base for modelling the system (Figure 1). The detail description of the system is presented in previous study [9], the components and their parameters are presented in [10] while the component parameters of the model are highlighted in section 3. The main components of the hydraulic press are: 1) hydraulic power unit; 2) servo valve MOOG D765; 3) Hanchen servo cylinder (series 320); 4) press frame (Kern Tool Technology, 1113 ISO 3-plate tooling frame); 5) PC and HMI (NI LabVIEW); 6) Moog P-I servo amplifier, G122-829A; 7) position transducer (Messotron Henning GmbH, type: DLH300); 8) pressure sensors (Turck, PT400R).

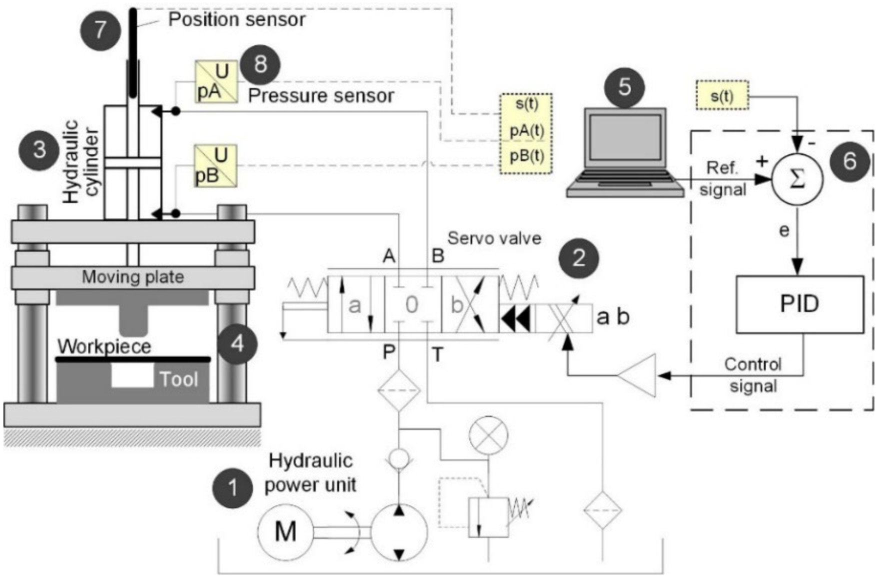


Figure 1: The concept of hydraulic press.
Source: own.

3 Modelling and simulation of hydraulic system

3.1 Modelling of hydraulic system

For the given concept of a hydraulic system (Figure 1), a development of the simulation model is carried out by using the simulation tool DSH^{plus} (Figure 2) representing the quasi-replica of the real hydraulic system [11].

The simulation model structure contains hydraulic, control and mechanical components in order to achieve the same functionality as the real hydraulic system. Hydraulic power unit is simplified involving only the hydraulic pump (01) and pressure relief valve (02) to achieve proper flow rate and working pressure. Servo valve 4/3 PT1 (03) is used to simulate first order system behaviour in opening and closing regime. Double-rod, double-acting hydraulic cylinder is used (04) with output signal activated to monitor the cylinder velocity (v04) and cylinder stroke (s04) used in closed-loop position control. Mechanical system with spring-damper component (05) and moving mass (06) is added to simulate the stiffness of hydraulic press frame and the mass inertia of the moving parts. Function generator (07) is used to set the position reference (s07). Proportional amplifier (13) is used to convert the reference stroke (s07) from millimetres to reference stroke (s13) as electrical value in Volts. The SUM component (08) and the PI module (09) are used as real Moog P-I servo amplifier G122-829A. The signal limiter (10) is used since Moog servo vale has integrated control electronic that limit the control signal to maximum and minimum signal ±10 V. *P* transfer function module (11) is used to convert and amplify cylinder stroke (s04) in millimetres to output voltage signal (s11) in Volts according to the characteristics of real position transducer amplifier. The detail characteristics of simulation components used from DSH^{plus} library, are presented in Table 1. To generate an external force (s12, system disturbance) the function generator (12) has been added to the model if required in future investigation.

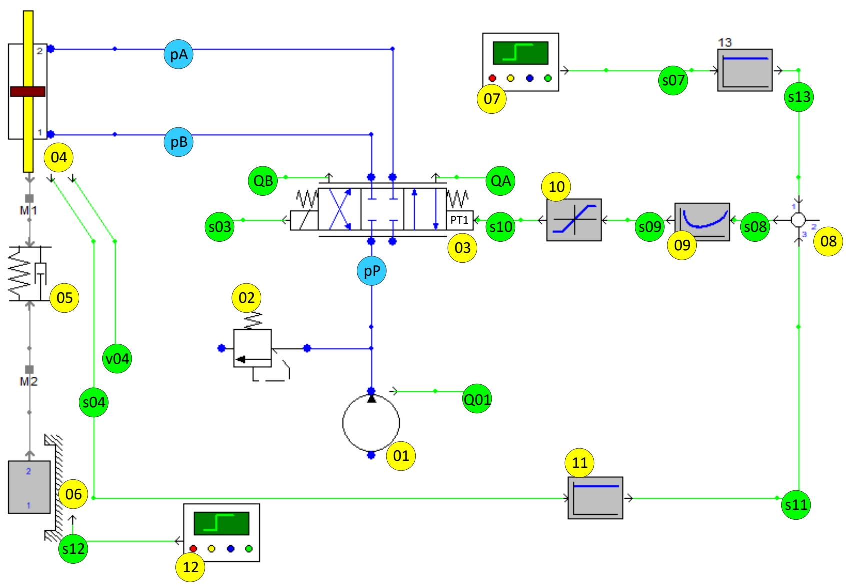


Figure 2: Simulation model of hydraulic closed-loop position control system.

Source: own.

Table 1: Simulation parameters of hydraulic system.

Component	Simulation component and parameters
01–Hydraulic pump	Flow rate: Q= 44 l/min
02–Pressure relief valve	Opening pressure, working pressure: p=100, 150, 200, 250 bar; nominal size: Q=60 l/min (Δp=5 bar).
03–Hydraulic servo valve Moog D765	Type: servo valve 4/3 PT1; nominal size: Q=38 l/min (Δp=70 bar); spool overlap: 1-3%; response time: t=10 ms (100%); input control signal: U=± 10 V; spool stroke: ± 100%; monitoring the spool position s03 and flow rate QA and QB.
04–Hydraulic cylinder, Hanchen, series 320	Type: double-acting, double rod; dimensions: 45/30/200 (piston/rod/stroke); mass of movable parts: m=1 kg; orientation, vertical: 90°; static friction: Ftr_s=10 N; mixed friction: Ftr_m=1 N; dynamic friction: Ftr_d=0,1 m/s; damping: d=10 Ns/mm.
05, 06–mass-spring-damper, to simulate the hydraulic press frame, Kern Tool Technology	Spring stiffness: 287 kN/mm; damping: 1 kNs/m; moving mass: m_1=50 kg, m_2=100 kg (m_3=300 kg); static friction: Ftr_s=20 N; mixed friction: Ftr_m=1 N; dynamic friction: Ftr_d=0,1 m/s; damping: 10 Ns/mm; orientation, vertical: 90°.
07–Function generator	Generation of press cycle, reference position $s(t)$.
08–Sum point	Comparator, factor 1: 1; factor 2: 1; factor 3: -1
09 - PI controller	KP: 1...1000 (P_2=10 set as initial value) KI: 0 (initial setting, analysed during the simulation)
10–Signal limiter	Maximal signal: 10 V; minimal signal: -10 V.
11–Position transducer	Amplifier: proportional gain Kp: 0,05 V/mm.
12–External force generator	Not considered in the analysis, simulation of forming force acting as axial force on hydraulic system.
13–Signal amplifier	Reference signal conversion [mm] - [V].
Hydraulic nodes	pP: 10 l, initial pressure 0 bar; pA: 0,1 l, initial pressure 0 bar; pB: 0,1 l, initial pressure 0 bar.

3.2 Simulation scenarios

The simulation involves detailed analysis of main influencing parameter, which can affect the hydraulic system behaviour (the system response). The influential parameters are:

I. **Servo valve response time (t):** Several time constants are analysed to cover different hydraulic valves: t_1= 5 ms, t_2 = 10 ms and t_3= 20 ms.

II. ***P* (proportional gain) and *I* (integral gain):** P_1=5, P_2=10, P_3=15 and P_4=20 are considered for analysis.

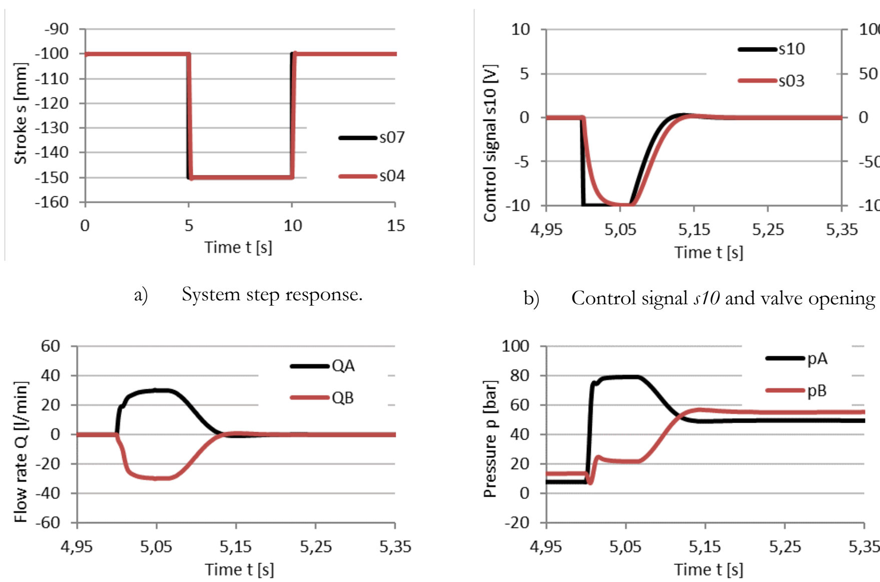
III. **Valve spool overlap (z):** Three sizes of the positive valve overlap are considered: z_1=0% (servo valves), z_2=+3% (high-response direct drive servo valves), z_3=+15% (proportional valves).

IV. **Control volumes – nodes *pA* in *pB*:** Cylinder chambers, internal channels of the mounting plate or the pipes connected between control valve and the cylinder. The smallest volume pA_1=pB_1=0.01 l is considered to simulate the servo valve and mounting plate in the cylinder; pA_2=pB_2=0.1 l and pA_3=pB_3=1 l volume is chosen to simulate the control valve - cylinder connection via pipes (different diameters and lengths).

V. **Working pressure (*pP*):** For the analysis, we considered: pP_1=100 bar, pP_2=150 bar, pP_3=200 bar and pP_4=250 bar.

VI. **Mass of the moving parts (m):** Proposed scenarios: m_1=50 kg (no tooling), and m_2=100 kg (with possible tooling).

VII. **Change in friction – guiding system:** The damage of the guiding elements results in friction change (static friction, mixed friction, dynamic friction). Several static and mixed frictions are analysed: F_1=10 N, F_2=100 N and F_3=1000 N.



a) System step response.

b) Control signal *s10* and valve opening *s03*.

c) Flow rate *QA* and *QB*.

d) Pressure response *pA* and *pB*.

Figure 3: Analysis of hydraulic system behaviour.
Source: own.

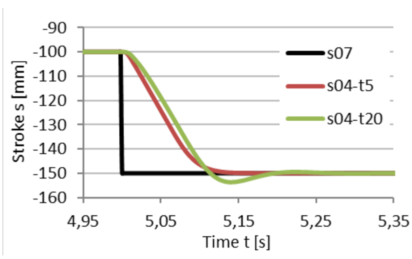
In the analysis, we consider the reference step signal, s_0=-100 mm (initial cylinder position) and s_1=-150 mm (end position). The analysis is focused on system response (Figure 3a) where *s07* represents the reference cylinder position (step position signal) and *s04* the cylinder position (simulation). For a better understanding, additionally the valve response presented in Figure 3b (s10 – valve control signal and *s03* – valve opening), the valve flow rates *QA* and *QB* (Figure 3c) and the pressure responses *pA* and *pB* (Figure 3d) are analysed.

4 Results and discussion

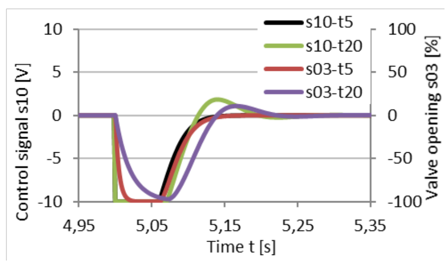
The results presented in this section show how the change of different parameters of hydraulic components influence the hydraulic system behaviour.

4.1 Influence of servo valve response time

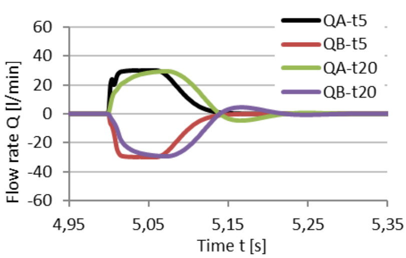
The results show better step response of the system while using high response valve. Response time t_1=5 ms results in stable system response without overshoot, while response time t_3 =20 ms results in overshoot of the system (O=3,6 mm) and the settling time t_{stl}=0,17 s.



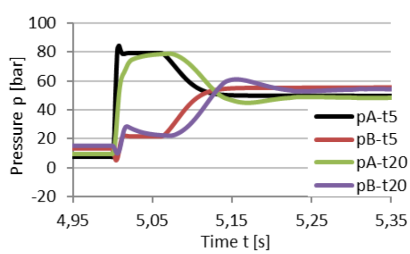
a) System step response.



b) Control signal *s10* and valve opening *s03*.



c) Flow rate *QA* and *QB*.



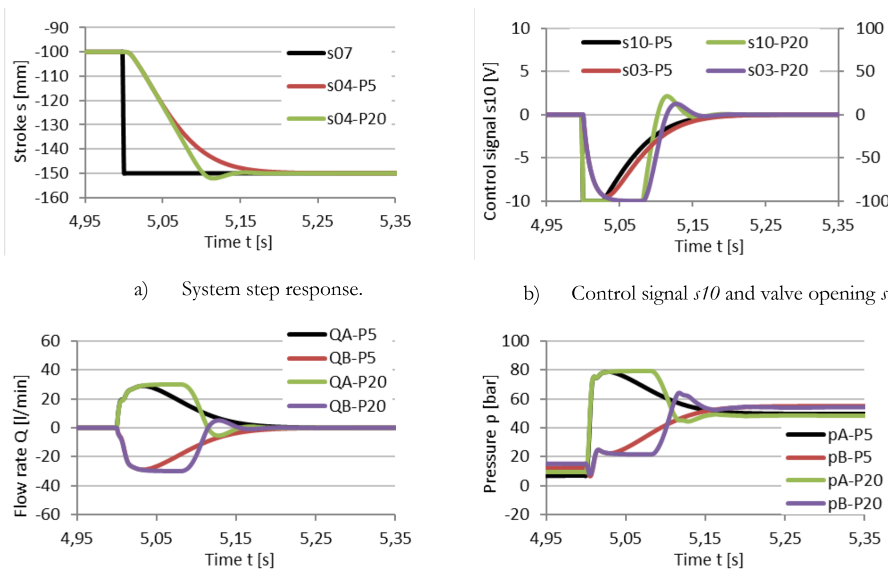
d) Pressure response *pA* and *pB*.

Figure 4: Influence of servo valve response time on system behaviour.
Source: own.

Low dynamic characteristic of the system is the results of low dynamic behaviour of the valve (Figure 4b) and consequently the low dynamics of the flow and pressure characteristics (Figure 4c, Figure 4d). The stable condition without overshoot of the system can be achieved with smaller *P* (approximately *P*=4).

4.2 Influence of *P* (proportional gain) and *I* (integral gain)

Figure 5a shows the response of the system at P_1=5 (red curve) and P_4=20 (green curve). It can be concluded that a small *P* results in a poor system response, while a high *P* results in a fast system response with small overshoot.



a) System step response.

b) Control signal *s10* and valve opening *s03*.

c) Flow rate *QA* and *QB*.

d) Pressure response *pA* and *pB*.

Figure 5: Influence of control parameters P and I on system behaviour.
Source: own.

We can also conclude that we need to find the proper *P* in order to achieve fast and stable response. By using the valve with time constant t_2=10 ms, the optimal *P*=8. While using the valve with time constant t_1=5 ms, the optimal *P*=17 resulting in higher response of the system (settling time is reduced by 0,03 s), no overshoot recognized (Figure 6).

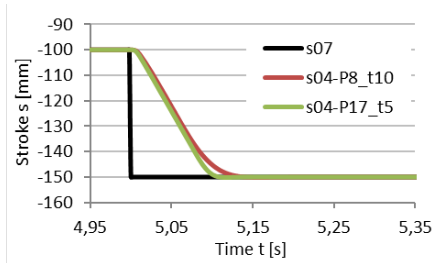


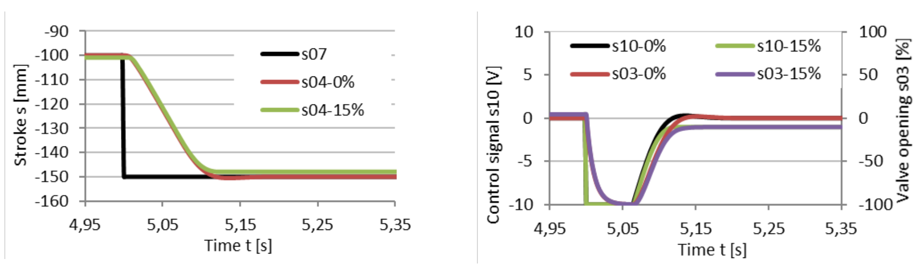
Figure 6: Response of the system at *t*=5 ms, *P*=17 and *t*=10 ms, *P*=8.

Source: own.

The simulation results indicate that *I* gain does not significantly affect the steady-state error (Δs). Moreover, the added *I* deteriorates the stability of the system. For the gain I=0, the Δs at cylinder position s=-150 mm is Δs=0.003 mm. If the *I* is increased I>0.1, the Δs is equal to Δs>0.01 mm. Based on this, it is recommended to use only the *P* control for the given hydraulic system.

4.3 Influence of valve spool overlap (z)

The increase in the positive valve overlap has a bad effect on the responsiveness of the hydraulic system and at the same time affects the steady-state error. Figure 7a shows a comparison of the system response between a valve overlap 0% (red curve) and +15% (green curve). The steady-state error with a positive overlap of the valve (+15%) is up to 2 mm (s=148 mm in Figure 7a). In Figure 7b it is clearly seen that the positive overlapped valve remains open in the overlapped dead zone resulting no flow across the valve and stops the movement of the hydraulic cylinder.



a) System step response.

b) Control signal *s10* and valve opening *s03*.

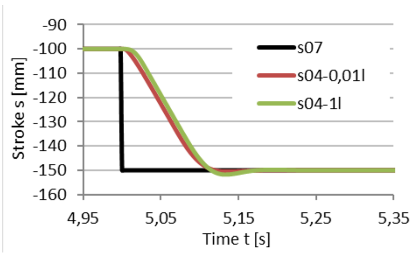
Figure 7: Influence of spool overlap on system behaviour.

Source: own.

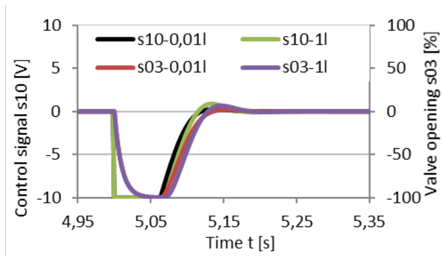
All this is also the result of the size of the proportional gain P and the small position error calculated in closed-loop control, which results in a very small valve control signal and small opening of the valve (within the size of valve overlap). The problem of steady-state error can be improved by increasing the proportional gain from P=10 to P=17.

4.4 Influence of control volumes – nodes *pA* and *pB*

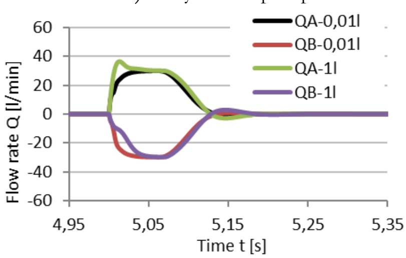
The results show that by increasing the volume of *pA* and *pB*, the hydraulic stiffness of the system decrease. As a result, the system response is decreased and the higher overshoot appears (Figure 8a). The main cause represents the pressure response (Figure 8d), which is the result of valve behaviour (Figure 8b) and the flow rate (Figure 8c).



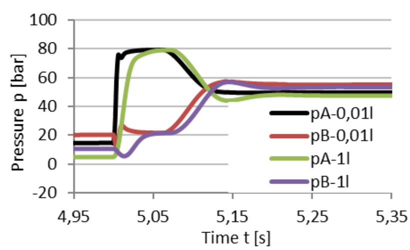
a) System step response.



b) Control signal *s10* and valve opening *s03*.



c) Flow rate *QA* and *QB*.

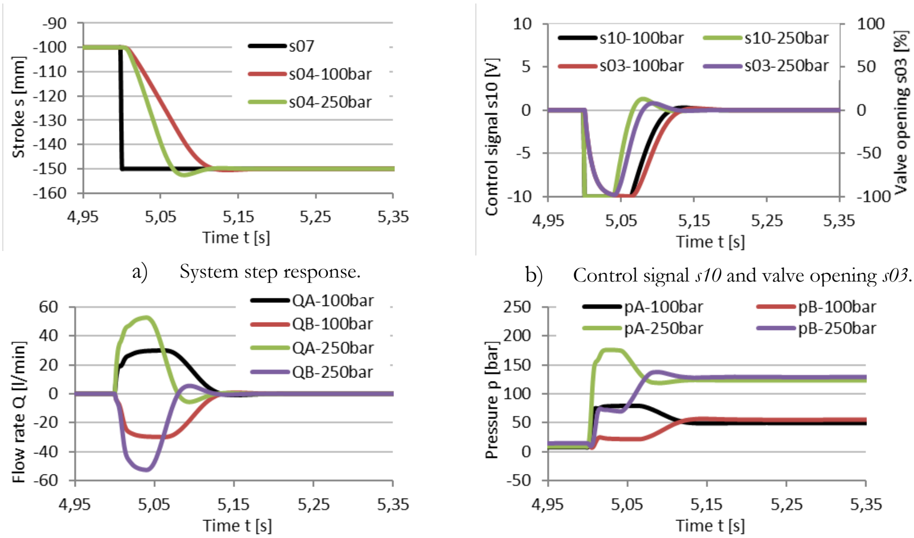


d) Pressure response *pA* and *pB*.

Figure 8: Influence of control volumes on system behaviour.
Source: own.

4.5 Influence of working pressure (*pP*)

Two working pressures (*pP*) are taken into consideration, pP_{min}=100 bar and pP_{max}=250 bar. The increase in *pP* influence on increase of hydraulic stiffness of the system and therefore on change in pressure distribution at *pA* and *pB* (Figure 9d) and better response of the system (Figure 9a). Change in pressure *pP* also effects on the flow rate *QA* and *QB* (Figure 9c) and thus hydraulic cylinder velocity, consequently the hydraulic system overshoot. Overshoot of the system can be improved by using higher response valves (*t*=5 ms) or reducing the *P* gain (*P*=5) while response of the valve remains the same (*t*=10 ms).



a) System step response.

b) Control signal *s10* and valve opening *s03*.

c) Flow rate *QA* and *QB*.

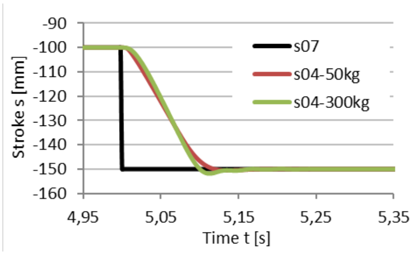
d) Pressure response *pA* and *pB*.

Figure 9: Influence of working pressure on system behaviour.

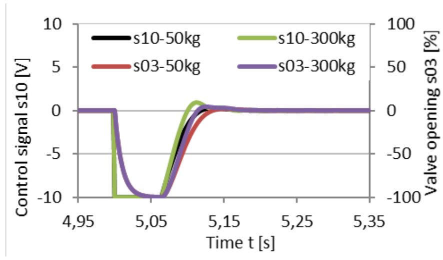
Source: own.

4.6 Influence of moving mass (m)

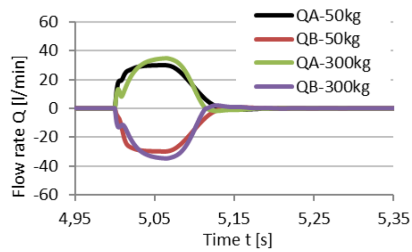
Preliminary simulation shows that increasing the mass from *m*=50 kg to *m*=100 kg does not significantly affect the behaviour of the system. The change in system behaviour is seen when the mass exceeds 200 kg (Figure 10a). One example is given by Figure 10, where mass of 300 kg is used and compared with initial system (*m*=50 kg). The main effect can be seen in settle time and system overshoot. The unstable system response can be corrected by changing the working pressure from *pP*=100 bar to at least *pP*=200 bar and a reduction of the *P* gain from *P*=10 to *P*=5 for the chosen response time *t*=10 ms.



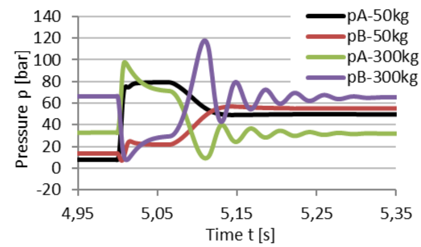
a) System step response.



b) Control signal *s10* and valve opening *s03*.



c) Flow rate *QA* and *QB*.



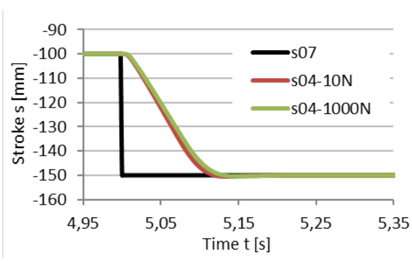
d) Pressure response *pA* and *pB*.

Figure 10: Influence of moving mass (m=300 kg) on system behaviour.

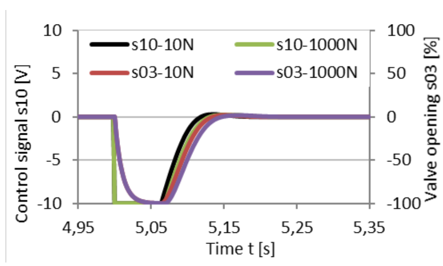
Source: own.

4.7 Influence of change in friction – guiding system

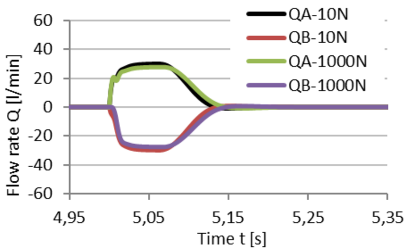
The results from Figure 11 show a small influence of the friction force on change of system behaviour. Nevertheless, higher friction force (*Ftr*=1000 N) leads to lower velocity of hydraulic cylinder (Δv<0,01 mm/s) and small overshoot (Δs<1 mm) of the system (Figure11a). The main difference appears in pressure distribution (Figure 11d) where the pressure difference (*pP-pA* and *pB-pT*) is increased to overcome the additional friction of the system.



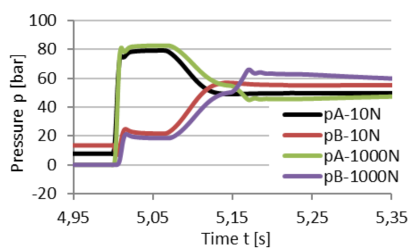
a) System step response.



b) Control signal *s10* and valve opening *s03*.



c) Flow rate *QA* and *QB*.



d) Pressure response *pA* and *pB*.

Figure 11: Influence of change in friction on system behaviour.

Source: own.

5 Conclusions

The paper presents a simulation approach to characterise and recognize the most influential parameters of proposed hydraulic system that effects the hydraulic system behaviour. The main contribution of this research is the modelling of the hydraulic system, characterization of the influential parameters for hydraulic system, in depth monitoring and analysis of measured variables, simulation scenarios presentation and simulation analysis performance evaluation with discussion.

- The results of this study show that the control valve response time has major influence on system behaviour such as system response, size of system overshoot and settling time. The higher the response of valve is the better the system response is achieved, which is logical. In this context we have to take into consideration the proper control parameters, i.e. the proportional gain, to achieve stable condition.
- The valve spool overlap has minor influence on system response, which can be improved by using larger proportional gain of control unit.
- The control volume or the dead volume has impact on hydraulic stiffness of the system. The small volume results in higher stiffness of the system and therefore on better system response. The same can be said for the higher working pressure of the hydraulic system. As noted in theory the pressure above 200 bar is recommended to eliminate the effect of hydraulic stiffness.

– The moving mass increase slightly effects on system overshoot and settle time increase. It depends on working pressure, the response of the control valve and control parameters set-up.
– The increase of friction in the guiding elements has minor effect on system behaviour. The major concern is the increase in pressure and thus increase in hydraulic power needed for the given press cycle.
– In our opinion the most influential parameters are the control valve response in combination with the control parameters and working pressure. For these parameters the proper values should be set to achieve fast and stable response of hydraulic system.

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