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Modelling of a Hydraulic Actuator with Variable Displacement Pump for Front Shovels

[1]Juma Yousuf Alaydi and [2]P.M. Gladky

[1]Mechanical Engineering Dept., IUG, Palestine
[2]Hydro-Pneu-Automatic Dept., KPU, Ukraine

Abstract: In this paper an electro-hydraulic actuator with a variable displacement pump was modeled. The control algorithm of the actuator was itemized and analyzed. The items of the electro-hydraulic control system were presented and modeled. The results of the simulation were also presented. The results confirmed that the proposed control algorithm was successful and had an excellent and robust behavior. The results also showed a very good performance of the overall control algorithm and validated the practical application of the proposed control algorithm.

Key words: Hydraulic system · Hydraulic excavator · Variable load sensing system · Electrohydraulic actuator

INTRODUCTION

It is recommended to develop systems which are characterized by high efficiency coefficients. Conventional hydraulic supply systems typically provide a constant supply pressure or a constant supply volume flow, independent of the actual demands of the load. Thus, the worst energetic efficiency occurs in the case when no power is needed by the load. The increasing demands on the energetic efficiency require the implementation of hydraulic supply systems which can be adjusted to the actual requirements of the load (load-sensing). Basically, two approaches do exist to control the supply volume flow [1].

One way to increase the efficiency is to use a pump characterized by constant volume and changeable speed. In this case it is useful to use a combination of asynchronous motor and frequency changer. This is a smart and easy solution, because the frequency changer allows comfortable control of the speed as well as reduces the consumed power. In many applications, fixed displacement pumps are driven by electric motors which allow an easy control of the speed. The dynamics, however, are very limited such that in general the demands on the dynamical performance cannot be met with this concept. This is even more obvious if the pump is driven by a combustion engine [2].

The second possibility to control the volume flow of a pump is to change the displacement of the pump. In this context, variable-displacement axial piston pumps are often used, whereby the displacement of the pump (i.e. the volume flow) can be changed by tilting a swash plate. This can be done fast enough to meet the dynamical demands of many loads [3].

The present paper deals with the supply pressure control of electrohydraulic systems comprising a variable displacement axial piston pump and a variable load. Typically, linear control strategies are used in such applications, see, e.g., [2] and [3]. Since electrohydraulic systems exhibit a significant nonlinear behavior the performance of the closed-loop system is normally rather limited. Furthermore, a rigorous stability proof is lacking in most cases and the tuning of the controller parameters turns out to be very time-consuming. In this work, a new model-based nonlinear control algorithm is derived, which, on the one hand, takes into account the essential nonlinearities of the system and, on the other hand, can be easily adjusted to pumps of different installation sizes in the same model range.

A general problem in designing a load-sensing system is to find out the actual demands of the load, since in most cases the load is neither known nor can it be measured. This problem also occurs in the application considered in this paper where the load not only is

Corresponding Author: Juma Yousuf Alaydi, Mechanical Engineering Dept., IUG, Palestine.

unknown but may also change in a very fast manner. In order to deal with this fact, the nonlinear control algorithm has to be augmented by a load estimator. This is a challenging task since it is well known that the separation principle of the controller and the estimator design does not hold for nonlinear systems [12].

In order to meet the high demands both on the tracking behavior and the non-linearity of the closed-loop system a two degrees-of-freedom control structure, comprising a PI and a PID part, is proposed in the controller design. Thereby, the design of the control algorithm becomes very challenging in the considered application due to the fact that the axial piston pump is self-supplied, i.e. the volume flow which is necessary to control the pump is taken from the output volume flow of the pump. This, as will be outlined in detail in the next section, yields a switching mathematical model of the system [13].

In this paper an electro- hydraulic actuator with a variable displacement pump was modeled, this actuator deals with pressure control self-supplied variable displacement axial piston pump. The control algorithm of this actuator was itemized and analyzed. The items of the control system were presented as well as the solution of control circuits. The results of the simulation were presented.

Mathematical Modeling and Control System

Dynamic Modeling of Excavator Boom Cylinder: The shovel kinematics models are obtained for the front shovel assembly. The purpose of introducing the kinematic models of the shovel is to transfer bucket tip's reference trajectories to the corresponding and required reference angles sequence of each joint and to get the motion sequences of the hydraulic cylinders. The dynamic models use the results of the kinematics as input to develop the models for active and resistive forces, moments and resistive moments.

The mathematical modeling of hydraulic excavator was the introduced in many different researches, as in, [4], [5] and [6], detailed works are available for the mathematical modeling variable displacement pumps [6], [7], [8], [11], [12] and [15]. Since the detailed mathematical models seizing all the dynamical properties are in general relatively complicated they are not suitable for a model based controller design. Consequently, an analysis of the dynamics of the system based on the singular perturbation theory was performed in [2] to steadily reduce the general complication of the mathematical model. Analysis of hydraulic servo valve actuator system is presented in [9] [10] and [17]. The model, which will be used in the following control design, is derived from the boom cylinder kinematics and hydraulic system shown was presented and analyzed in [5] is shown in Fig. 2. and Fig. 3.

This system is characterized by variable parameters depending on the element positions [18] and [19]. It is clear that any change in the digging mechanism element position will result in extensive changes in the inertia and the equivalent mass as well as the total contained oil volume. Thus, the control system with dynamic characteristics require to design a control algorithm structure (dynamic controller), which is efficient to compensate the dynamic variation of the controlled object parameters. The dynamic parameters changing range of the boom mechanism can be modeled from Fig. 2 as given in [5]:

- The movement equation of the mass m, which is actuated by cylinder HC.

$$m\frac{dv}{dt} = A \cdot P_1 - R \tag{1}$$

- Continuity equation of the oil in the cylinders chambers

$$\frac{V_1}{E}\frac{dp_1}{dt} = K \cdot y_p - C_e \cdot P_1 - A \cdot v \tag{2}$$

where: m, mass, v, piston rod velocity. A is the effective area of the piston. R, external load, V_1 is the volume of hydraulic cylinder chamber, E is the oil bulk modulus.

Combining equation 1, 2 and 3 the mathematical model will have the following form:

$$m_{eq}\left(\vec{h}\right)\frac{dv}{dt} = A_1 P_1 - A_2 P_2 - R\left(\vec{h}\right),$$

$$\frac{V\left(h_1\right)}{E}\frac{dP_1}{dt} = k\psi - (C_e + C_i)(P_1 - P_2) - A_1 v, \tag{3}$$

where:

m_{eq} is the equivalent mass of the element structures acting on the cylinder piston rod, kg

A_1, A_2 are the areas of the piston and rod chambers, .m^2

P_1, P_2 are the pressure at the piston and rod chambers, Pa.

k is the pump control element (swash plate) gain coefficient $m^3/rad/deg$.

V is the piston rod velocity, m/s.

V is the total contained volume including the chambers and the connecting hoses and manifolds. E is the liquid bulk modulus of elasticity (constant).

φ is the swash plate angle ($0 \leq \varphi \leq 1$)

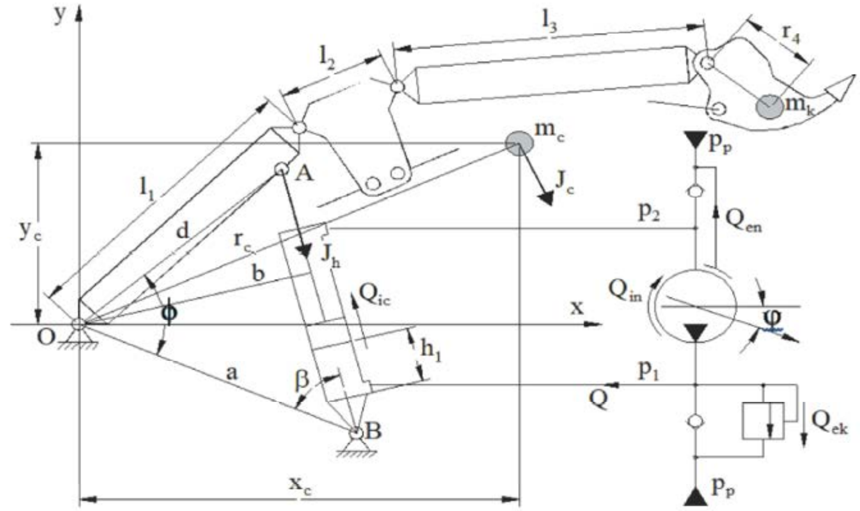

Fig. 2: The hydraulic control system of excavator boom cylinder

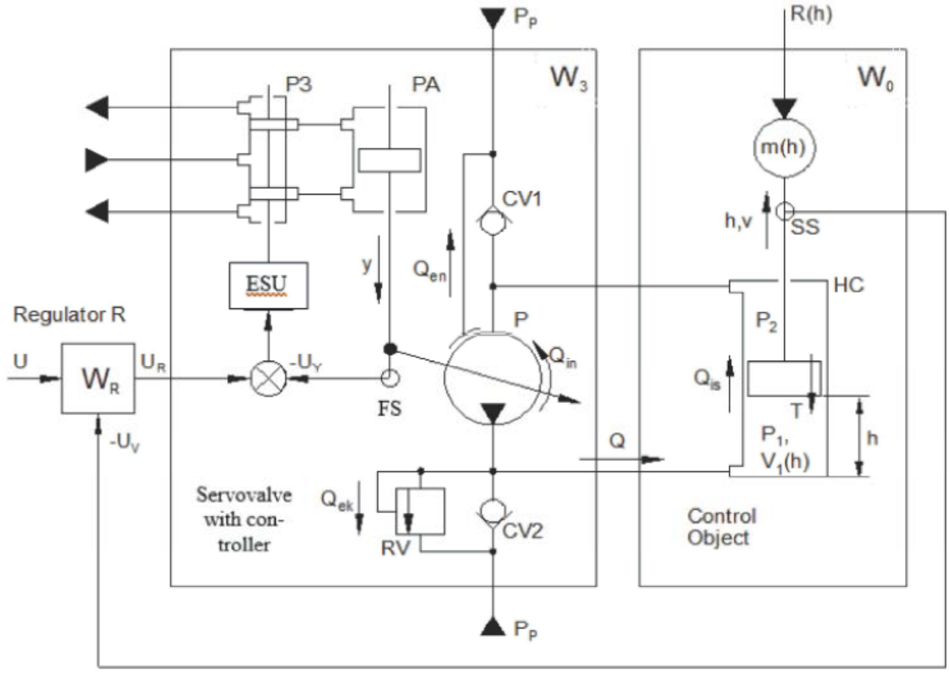

Fig. 3: Schematic diagram of the hydraulic servo valve actuator, the controller and the load *R(h)*

C_e, C_i are the external and internal leakage coefficients.
$\bar{h}$ is the victor of digging mechanism state position characterized by cylinder stroke, h_1 boom, h_2 insert, h_3 stick and h_4 bucket. T is the time

Equation 4 can be Laplace transformed to give.

$$\left(\frac{m_{eq}(\vec{h})\ V(h_1)}{EA_1^{\,2}}s^2+\frac{m_{eq}(\vec{h})(C_e+C_i)}{A_1^{\,2}}s+1\right)\cdot\delta v=k\cdot\delta\varphi \tag{4}$$

where:
δ is the variable deviation from steady state value.
s is Laplace operator.

The system characteristic equation can be analyzed to give:

$$\frac{m_{eq}(\vec{h})\ V(h_1)}{EA_1^{\,2}}=T^2 \tag{5}$$

$$\frac{m_{eq}\left(\vec{h}\right)\left(C_e + C_i\right)}{A_1^{\,2}} = 2T\xi \tag{6}$$

where:
T is the characteristic time, ξ is the damping ratio.
$M_{eq}(h)$ can be computed from the kinematics analysis of the system given in.

$$m_{eq}\left(\vec{h}\right)\, b\frac{d^2h_1}{dt^2} = m_c r_c\left(\vec{h}\right)\, w_\tau \tag{7}$$

where:
r_c, is the radius of rotation around hinge O,
w_τ is the angular acceleration of the center of mass,
m_c is the mass of the system

The servo mechanism of the hydraulic actuator in Fig. 3. controls a variable displacement pump as a final control element in the servo mechanism control. The pump controller controls the pump delivery according to the demanded flow, which depends on the actuator response sensor *ss*. The pressure difference between the two pressure lines (high pressure P_1 and low pressure P_2) is automatically given by the actuator sensing actuator. The low pressure is set to a constant value as the atmospheric pressure, which can also by employ for the supply of the pump control system. Therefore the hydraulic pump power is simultaneously adjusted to the required mechanical output power of the actuator. The servo mechanism of the hydraulic actuator has simple design and high response dynamic performance (a small mass of swash plate and control piston as well as the accurate flow control) the variable displacement axial piston pump would be usually the best choice for the servo pump.

The achievement of pump control is simple for both rotary actuators and linear actuators. Several industrial applications have been established during the last time. The kinematics and the whole machine design of earth moving equipment require the use of the servo mechanism of the hydraulic actuator.

Simulation of Electrohydraulic Control System

Load Modeling: The main clue of the load oriented control system is a system associated with the load. In order to test the system a load mechanism was designed, which was able to change the load during the work. The block diagram of hardware designed to control the single bucket excavator is shown in Fig. 2 and modeled in Eq. 4. The control system senses a reference low pressure setting a reference swash plate angle and the high pressure level controller computes reference angles corresponding to reference trajectories and transfers orders to low-level controller. Accordingly the low pressure controller acquires position control by measuring the swash plate angle and pressure of cylinders. That is, controlling the pilot valve to drive the cylinders according to the output current. These inputs can be formulated from eq. 7 and the data given at the end of this paper.

There exists hydraulic Load-Sensing system, which uses a pump characterized by constant angular velocity and variable volume. The main idea of a new structure Electrohydraulic Load-Sensing system is to use a pump characterized by constant capacity and variable angular velocity, which is realized by asynchronous motor powered by frequency controller.

Modeling of Electrohydraulic Load-Sensing System: The Schematic diagram of the hydraulic servo valve actuator is shown in Figure 3. To produce the mathematical model of this system, it is required to model the following subsystems.

The Controller Unit: Is concerned with the development of a nonlinear model based control algorithm for the electrohydraulic system. In this work, a two degrees-of-freedom control structure comprising a feed forward and a feedback part is used to solve the aforementioned control task.

Modified PI Controller: There are many ways to control the load. They are based on standard PID controllers, state controllers, fuzzy controllers etc. Radek MANASEK in [20] introduced and used a modified and PID controller, because the load consists of saturation non-linearity.

Fig. 4. Shows the modified PI controller. The main idea of the modified PI controller is to stop integration as soon as the saturation limit of the input signal is reached. If the system returns from the saturation state, the integration part will respond immediately after making step over the saturation limit. If we use a standard PI controller, the integration part will not stop after reaching the saturation state and will keep increasing during abidance of the system in the saturation area [20].

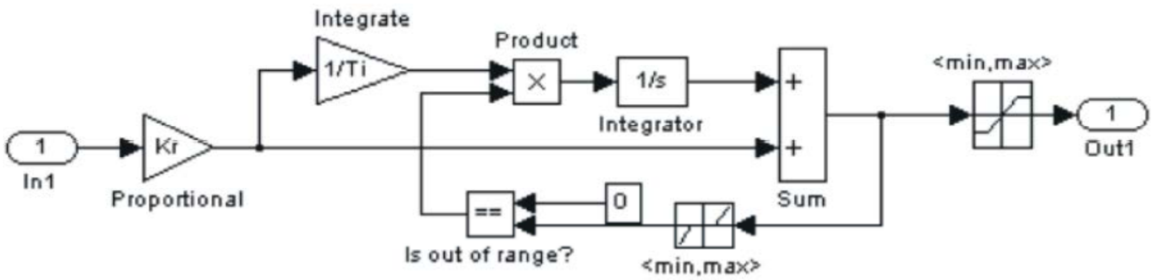

Fig. 4: Modified PI controller

Control of the Pressure Source P_1 (The Variable Displacement Pump): As it can be seen form Figure 5, the pressure source consists of: electrical drive, pump and connected pipeline, which have to be involved too. Following is a summary of the assumption that has been made in developing the model of a hydraulic cylinder:

- The proportional valve is a symmetrical 5/2 way valve. The dead band of valve is also symmetrical and the flow in it is turbulent.
- Possible dynamic behavior of the pressure in the transmission lines between valve and actuator is assumed to be negligible.
- Pressure is equal everywhere in one volume of hydraulic cylinder and the temperature and the bulk modulus are constants.
- The leakage of flows is laminar.

The linearization of the system was accomplished and the parameters setting of the PID controller by various methods were tested by using modified PID controller. The best results were received by minimal regulation area method.

The feed forward control designed in the last subsection has to be augmented by a feedback control in order to stabilize the tracking error in case of parameter variations or model uncertainties. Therefore, the pressure error $e = P_1 - p_{1,d}$ and the error in the swash plate angle $e_\varphi = \varphi_p - \varphi_{p,d}$ are introduced [20].

Experimentation Measurement Results: The test stand was designed and built at the university hydraulic lab, see Fig. 3. The main components of this test stand are the variable displacement axial piston pump driven by an electric motor and controlled by the control valve, the load volume and the load orifice. The schematic diagram of the hydraulic circuit of the test stand is given in Fig. 2 and the parameters of the system are summarized below. The actuator for tilting the swash plate is controlled by a (5/2) proportional directional valve.

Simulation of Actual Parameters of the Pump and the Load: The given analysis allows computing the important parameters of the controlled object at any digging mechanism position given by victor $\bar{h}$, this calculation was accomplished for the following real dimensions collected from 15 m^3 single bucket excavator:

l_1=l_3=6.5m; l_2=1.9m; r_4=1.4m; a=3.0m; d=5.54m; h_{01}=3.3m; m_1=m_3=12.5*10^{3}kg; m_2=8.5*10^{3}kg; m_4=40*10^{3}kg bucket with full load; A_1=2*0.125m^2; E=1.2*10^{3}MPa;

Simulation of the system of equation (2) for T and ξ were accomplished in four dimensions net with step Δh_i=0.1H_i for each digging mechanism element h_1, h_2, h_3 and h_4 (H_i is the maximum stroke of the boom cylinder).

Leakage coefficients:

C_{ep}=2.24*10^{-11}m^5/(N•s); C_{ip}=5.22*10^{-11}m^5/(N•s);

The following coefficients collected from manufacturer catalogues. Relief valve leakage coefficient C_{er}=0.026*10^{-11}m^5/(N•s), Hydraulic cylinder leakage coefficient C_{ic}=0.137*10^{-11}m^5/(N.s).

The lumped leakage coefficients are:

$C_e = C_{ep} + C_{er}$; $C_i = C_{ip} + C_{ic}$

The circuit demonstrates usage of a load and pressure-limiting unit in a conventional reciprocal system with variable load on the forward stroke. The unit limits output pressure to 300 bar and maintains a preset pressure drop in 10 bar across the velocity control orifice OR. The unit is built of two 5-way, 2-position valves, two single-acting hydraulic valve actuators SA and SA1 and one double-acting hydraulic valve actuator DAA. We incorporated the model elements described above hydraulic actuator with load-sensing variable-displacement pump model using Simulink.

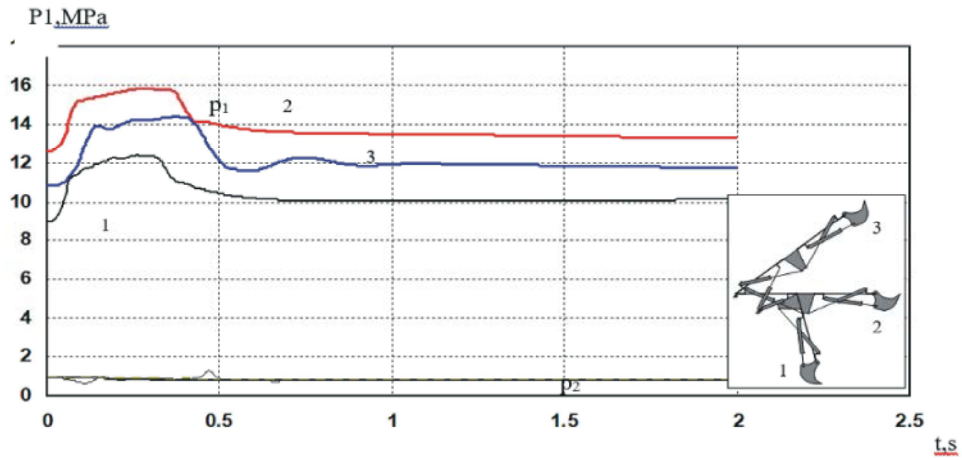

Fig. 58: The tracking behavior of the load pressure for three load positions

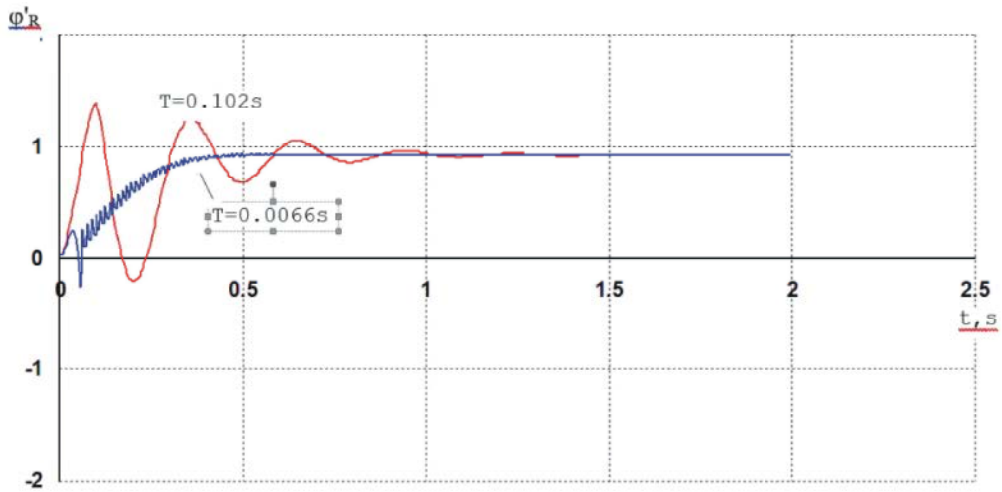

Fig. 69: Loads positions impact on the trajectories of the swash plate angle φ_R

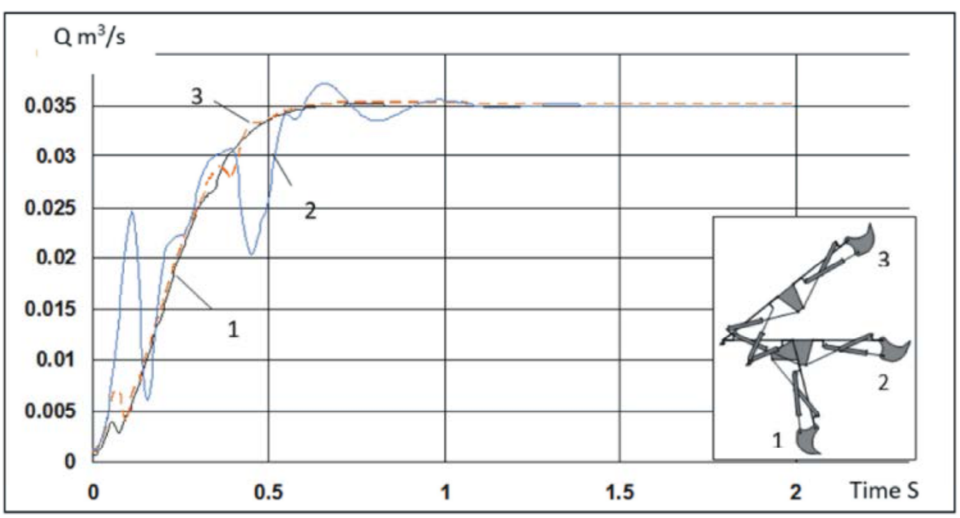

Fig. 710: The effect of the different load positions on actuator flow rate Q

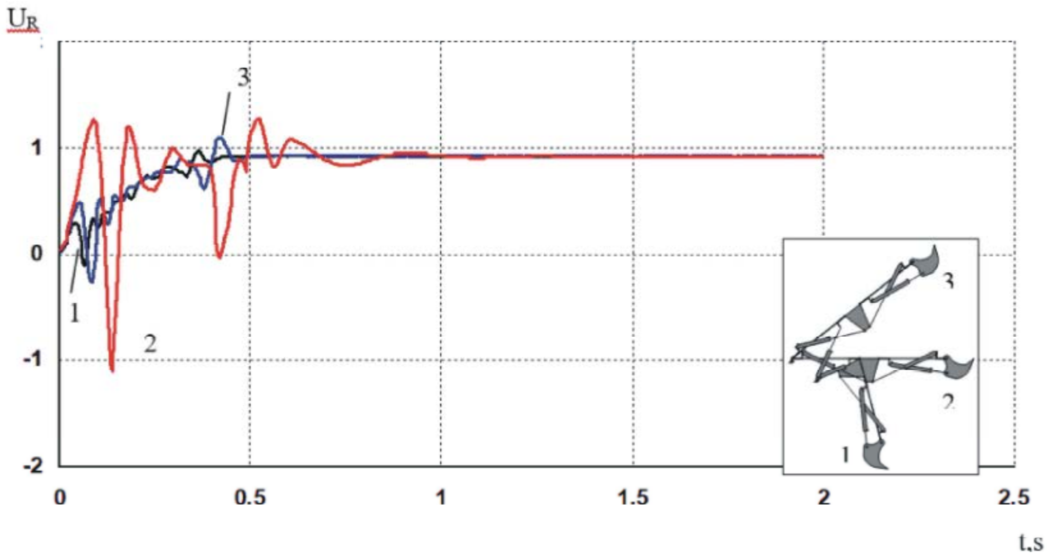

Fig. 811: The effect of the different load positions on input control signal U_R

RESULTS AND DISCUSSION

The tracking performance of the load pressure P_l is investigated. Three digging mechanism positions are analyzed. As can be seen from the time-evolution of the load pressure P_l an excellent tracking performance is achieved independent of the real value of the load. On the other hand, the different loads positions have a large impact on the trajectories of the swash plate angle φ_R. Evidently, this is due to the fact that only a small displacement Q of the pump is essential to provide the (small) load volume flow for the small load coefficient while quit much higher displacement flow Q is required for full loaded bucket. The effect of the different load positions can also be understood in the plots of the actuator flow rate Q and the actual input control signal U_R which are given in Fig. 8, this figure shows the performance of the system for fast changes of the load coefficient control signal with time. Here, the load servo valve orifice responds as fast as possible while a desired trajectory $P_l(t)$ in the load pressure is tracked. Of course, the rapid change of the load produces significant errors in the load pressure with high frequency of the signal changing but these errors are compensated and not appear at the load because of the damping and low response of the mechanical load compared to the electrical signal of the controller of the servo. From the results it is clear that stability exists of the overall closed-loop system, evidently, the case of rapidly changing loads is still in the stability margin. The dynamic behavior of the load coefficient, as given in Figures 5-8, show that the estimated digging mechanism position would have a rapid response of the load coefficient in an outstanding manner. It has to be mentioned that the large overshoot in the load coefficient will not affect the stability because it has low frequency compared to the servo valve control signal which has much higher frequency enable it to overcome these limited time overshooting intervals. However, the main emphasis of the control algorithm is good tracking of the load pressure P_l. For this task, the selected parameters of the controller have proven stability and feasible in practical application during real excavation performance of the load pressure and a good estimation of the load coefficient. To sum it up, the measurement results show a very good performance of the overall control algorithm and proves the practical feasibility of the proposed control algorithm comprising the feed forward control, the feedback control and the extended control unit.

CONCLUSION

Modeling and simulation of hydraulic actuator with variable load displacement pump deals with the pressure control of self-supplied variable displacement axial piston pumps. The basic setup of the hydraulic system and its mathematical model was analyzed. A new control concept for the pressure control of self-supplied variable displacement axial piston pumps with variable load was designed and to solve this control task, a two degrees-of-freedom control structure comprising a PI and a PID controller in combination with a load estimator was proposed.

The system was itemized and the models of basic elements were presented. The attention went in control circuits consequentially.

The advantages of this approach are the model-based design, which allows an easy implementation of the control concept to other installation sizes in the same model range and the simple parameterization by means of a few controller parameters. The feasibility of the control algorithm was shown by measurement results, whereby an excellent tracking performance could be achieved.

In this paper, the results exhibit good tracking performance for boom cylinder under the controller developed. The peak error is less than 4 degrees. To sum it up, the measurement results show a very good performance of the overall control algorithm and thus prove the practical feasibility of the proposed control algorithm comprising the feed forward control, the feedback control. The extended control unit and Control algorithm satisfies required transient response performance in large range of load and variation.

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