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Multi-pole modeling and simulation of an electro-hydraulic servo-system in an intelligent programming environment

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ABSTRACT

The paper presents composing of models and simulation of an electro-hydraulic servo-system, including hydraulic servo-drive with feedback regulator, electro-hydraulic servo-valve and constant pressure feeding system with variable displacement pump. For composing mathematical models of the fluid power system multi-pole models with different oriented causalities are used. Using multi-pole models allows describe models of required complexity for each component. Using a Java based intelligent programming environment CoCoViLa as a tool, enables one graphically describe multi-pole models of the system and perform simulations in a user-friendly manner. Solving large equation systems during simulations can be avoided. Models of four-way sliding spool throttling slot pairs describe open slot and overlapped slot characteristics of various causalities. For correcting the control signal to the electro-hydraulic servo-valve a non-linear differential regulator is used. An intelligent simulation environment CoCoViLa supporting declarative programming in a high-level language and automatic program synthesis is shortly described. It is convenient to describe simulation tasks visually using visual images of multi-pole models. The designer does not need to focus on programming, but instead can use the models with the generated code. Simulations of subsystems of electro-hydraulic servo-system and the entire system are considered and simulation results are presented and discussed.

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Electro-hydraulic servo-system; multi-pole model; intelligent simulation

1. Introduction

Electro-hydraulic servo-systems are used in various applications such as aircraft flight controls, satellite positioning controls, injection molding machines for the plastics markets, robots and manipulators, synchronized drives, power generating turbines, simulators used to train pilots, etc. Computer modeling and simulation is a substantial step in the design of such systems.

The history of significant references in the area of electro-hydraulic servo-systems is given by Maskrey and Thayer (1978) and Gordić *et al.* (2004). In analysis and system synthesis frequently simplified, 3rd to 5th order transfer functions are used as models of electro-hydraulic servo-systems (Thayer 1965, Merritt 1967, Jones 1997, Sohl and Borrow 1999, Jelali and Kroll 2003, Noah 2005, Johnson 2008). Often the models are linear. When using such models, dynamics of all components can't be taken into account adequately.

Servo-system dynamic responses are mostly described in terms of the logarithmical amplitude ratio and phase angle lag of output in response to sinusoidal input of varying frequency.

Using two-pole models for mechanical and hydraulic systems is not correct, as components of such systems exert feedback actions. This concerns also the Bond graphs method, often used in simulations (Xu *et al.* 2010).

Modeling and simulation tools in existence such as MATLAB/Simulink, SimHydraulics™, ITI SimulationX, DSHplus, Dymola, HOPSAN, VisSim, AmeSim, 20-Sim, DYNAST, MS1™, shortly characterized in (Grossschmidt and Harf 2009a), HYVOS 7.0 (Bosch Rexroth 2010), etc. used for simulation of fluid power systems, are object-oriented (systems are described as functional or component schemes) using equations with fixed causality or equations in non-causal form for each object. The obtained equation systems usually need checking and correcting to guarantee solvability. It is very complicated to debug and solve large differential equation systems with a great number of variables. Special integration procedures must be used in case of large stiff differential equation systems.

In the current paper an approach is proposed, which is based on using multi-pole models with different oriented causalities and oriented graphs of functional elements (Grossschmidt and Harf 2009a, 2009b, 2010). An intelligent simulation environment CoCoViLa (Kotkas

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et al. 2011) supporting programming in a high-level language, automatic program synthesis and visual programming is used as a tool for modeling and simulation. It is convenient to describe simulation tasks visually, using prepared images of multi-pole models. The designer does not need to focus on programming, but instead can use the models with the generated code.

2. Multi-pole models

Multi-pole models represent mathematical relations between several input and output variables (poles) (Grossschmidt and Harf 2009a, 2010).

Multi-pole models used for describing fluid power systems enable to express both direct actions and feed-backs as they occur in hydraulic and mechanical systems. Each component of the system is represented as a multi-pole model having its own structure including inner and outer variables and relations between variables. Usually variables in fluid power systems are considered in pairs (effort and flow variable), one of them is an input and the other is an output. Therefore multi-pole models contain at least two pairs of outer variables and more than one relation.

For example, hydraulic resistors, volume elasticities and tubes are represented as four-pole models with pressures and volumetric flows at both ends as poles. Models of four types (G, H, Y, Z as in electrical engineering) are used for resistors and tubes describing different causalities between input and output poles. Multi-pole models of more poles are used for describing more complex components and devices. For example hydraulic cylinder is described by a nine-pole model and piston in cylinder by a twelve-pole model. In multi-pole models flow direction from left to right is taken as default.

Using multi-pole models allows describe models of required complexity for each component. A component model can enclose equations, inner iterations, logic functions and calculation programs as relations.

Multi-pole models of components can be connected together only by using poles. Using multi-pole models enables methodical, graphical representation of mathematical models of large and complicated systems. In this way we can be convinced of the correct composition of models and we don't need to check the solvability. It is possible to directly simulate the statics or steady state conditions without using differential equation systems.

Implementing multi-pole models for each component gives us the possibility to use distributed calculations. Integrations are performed in each model separately. In case of loop dependences between component models a special iteration method is used.

3. Electro-hydraulic servo-system

In following chapters an electro-hydraulic servo-system is considered that consists of three main subsystems: hydraulic servo-drive with feedback regulator, electro-hydraulic servo-valve and constant pressure feeding system with variable displacement pump.

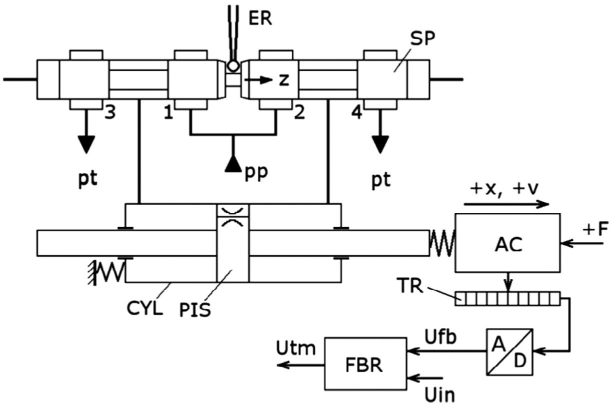

Figure 1. Functional scheme of a hydraulic servo-drive.

3.1. Hydraulic servo-drive

Functional scheme of a hydraulic servo-drive is shown in Figure 1. Servo-drive contains sliding spool SP giving feedback to servo-valve by elastic rod ER, hydraulic cylinder CYL, piston PIS, actuator AC, position sensor TR, digital-analog transformer A/D and feedback regulator FBR. Feeding pressure pp is as feeding system output. Outlet pressure is denoted by pt.

The four-way sliding spool SP has overlapped slots 1–4 (slots 1 and 2 for hydraulic cylinder inlet and slots 3 and 4 for hydraulic cylinder outlet). Sliding spool with overlapped slots enables to make volumetric flows minimal in central position of the spool, i.e. to act as lock of piston in cylinder.

Displacement of the sliding spool is denoted by z. The fixing of hydraulic cylinder CYL is at left end. Piston PIS has two piston rods and there is an orifice in piston for flow through. Piston rod drives the actuator AC (the positive directions of the displacement x and the velocity v of piston and the load force F are indicated). The position feedback from right piston rod is indicated as feedback voltage U_{fb}. The feedback regulator FBR calculates the torque motor feeding voltage U_{tm} from input voltage U_{in} and feedback voltage U_{fb}.

3.2. Electro-hydraulic servo-valve

In an electro-hydraulic servo-valve with mechanical feedback a torque motor is used as electro-mechanical transducer. A flapper is moved between nozzles by the torque motor. Feedback from sliding spool to flapper acts by elastic rod.

Electro-hydraulic servo-valve (Figure 2) consists of the following functional elements and subsystems: torque motor TM with flapper, flexure tube FT, nozzle-and-flapper valve NF with nozzles **N1** and **N2**, hydraulic resistors R1–R8, interface elements (tee

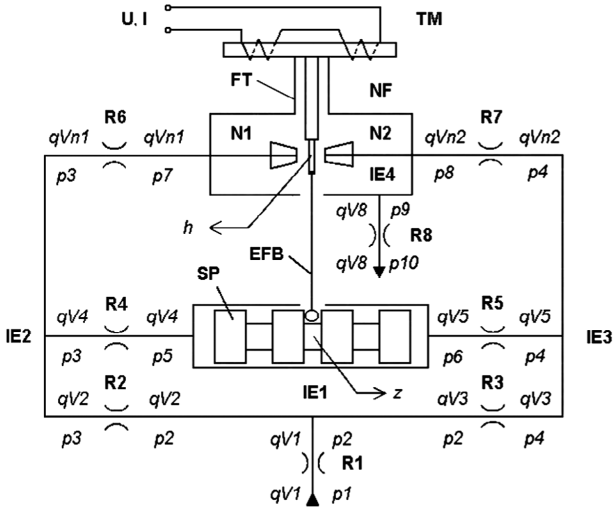

Figure 2. Functional scheme of an electro-hydraulic servo-valve.

couplings) IE1–IE4, sliding spool SP and elastic feedback EFB (as elastic conic rod) from spool to flapper.

Input variables are voltage U for the torque motor, feeding pressure $p1$ and outlet pressure $p10$.

Output variables are current I to the TM, position of the flapper h, position of the sliding spool z and volumetric flow rates $qV1$ and $qV8$.

The anchor of the torque motor is fixed on the flexure tube. The anchor turn angle $th1$ transmits to the stiff rod, which gives flapper the moving $h1$ between the nozzles. Difference of pressures at the ends of the sliding spool causes the spool to shift z. Elastic feedback rod bends, the flapper moves in opposite direction on position $h2$ and the anchor turns in opposite direction on the angle $th2$. Flapper takes position $h = h1 - h2$ and the anchor takes angle $th = th1 - th2$. The force acting to the sliding spool is Fz and the hydrodynamic force of fluid jets of nozzles is Fhd.

3.3. Constant pressure feeding system

Variable displacement pump PV with control system is used for feeding the electro-hydraulic servo-system. Under the steady state conditions the feeding system enables to provide the servo-system with feeding pressure, that changes in small range (3–4 bar), when the pump volumetric flow changes from minimum to maximum.

Functional scheme of a constant pressure feeding system with variable displacement pump is shown in Figure 3.

The pump PV is driven by electric motor ME through clutch CJh and controlled by a positioning cylinder CV. The right chamber of the cylinder CV is operated by the pump's outlet pressure. Pressure in the left chamber of cylinder CV is provided by the control system, which consists of a control valve VP containing a meter-in throttle edge RVS, a meter-out throttle edge RVT and a hydraulic resistor R5. The outlet of the pump is provided with volume elasticity VE. The outlet tube T1 of the feeding system is provided with a silencer (resistor R1 and dead-end tube T2) and a safety valve (spool VS and throttle edge RV). Hydraulic resistors R2–R7 ensure dynamic quality of the feeding system. IE1–IE4 are as interface elements (branchings) of hydraulic channels.

4. Modeling principles

To build up a model a fluid power system is decomposed into subsystems. Models of subsystems are composed using components from a library of multi-pole models. For each component models of required causalities must be available. This enables to compose multi-pole models for systems of different configuration to follow signal propagation in correspondence to physical or computational behavior.

Multi-pole mathematical models of basic hydraulic components such as hydraulic resistors, tubes, volume elasticities and hydraulic interface elements used in fluid power systems have been pre-composed and included into library. A library containing descriptions of physical properties of different hydraulic fluids is in use. Multi-pole mathematical models of basic hydraulic components and hydraulic fluids have been described in (Grossschmidt and Harf 2009a, 2009b, 2010).

Processes of two types are considered. Static and steady state processes are caused by static inputs. Dynamic transient responses are caused by inputs depending on time. Multi-pole models of different causalities are used for static and dynamic calculations.

When simulating dynamics multi-pole models of hydraulic servo-drive, electro-hydraulic servo-valve and constant pressure feeding system are used. The entire model is considered in the Chapter 11 of the paper.

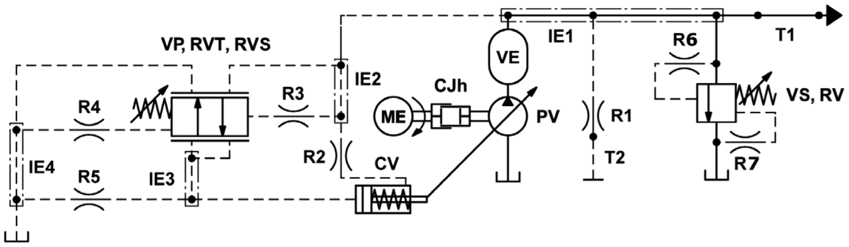

Figure 3. Functional scheme of a constant pressure feeding system with variable displacement pump.

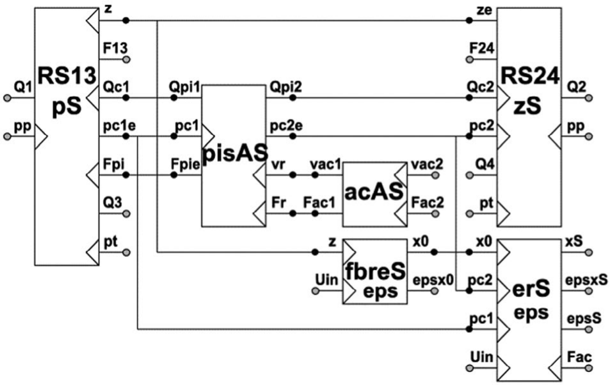

Figure 4. Multi-pole model of a servo-drive for statics and steady state conditions.

5. Multi-pole mathematical models

In this chapter multi-pole model of a servo-drive is described. Mathematical models of sliding spool slot pairs are described. Models of servo-valve have been described in (Harf and Grossschmidt 2012).

5.1. Model for statics and steady state conditions of a servo-drive

Multi-pole model of a servo-drive (servo-system with simple servo-valve model and with constant pressure feeding) for statics and steady state conditions is shown in Figure 4.

The model contains the following component models: sliding spool throttling slot pairs RS13pS and RS24zS, piston with rod pisAS, actuator acAS, system elasticities erSeps and servo-valve with feedback and with regulator fbreSeps.

Output variables of multi-pole models of components that require calculating variables in loop dependences (see Chapter 7) are denoted with suffix “e”.

Simulation results of the servo-valve statics show that displacement of sliding spool linearly depends on input voltage. So it is reasonable to use a linear transfer coefficient as a simple servo-valve model.

Simulation results of the variable displacement pump feeding system statics show that output pressure changes marginally if volumetric flow changes from minimum to maximum. It is reasonable to use a constant pressure as feeding pressure model.

Multi-pole models of two components, slot pairs RS13pS and RS24zS are described as follows.

Each sliding spool slot is characterized by displacement z, volumetric flow Q and pressure drop Δp. By known two of them the third can be computed.

Mathematical models of volumetric flows are different for overlapped and open slots. In case of open slots the volumetric flow depends on pressure drop square root. In case of overlapped slots the volumetric flow is proportional on pressure drop.

In a particular case slots are considered in pairs (1–3 and 2–4). For example the model of slot pair 1–3 includes computational procedures for different states of slots (1 open and 3 overlapped, 1 overlapped and 3 open, both 1 and 3 overlapped).

The multi-pole mathematical models RS13pS and RS24zS of sliding spool slot pairs have the following outer variables:

z displacement of servo-valve from initial position, m

$F13$, $F24$ sum of hydrodynamic forces of jets through slots 1, 3 and through slots 2, 4 respectively, N

$Q1$, $Q2$, $Q3$, $Q4$ volumetric flows through slots 1, 2, 3 and 4, m^3/s

$Qc1$, $Qc2$ volumetric flows to hydraulic cylinder left and right chamber, m^3/s

Fpi piston force, N

$pc1$ pressure in the left chamber of hydraulic cylinder, Pa

$pc2$ pressure in the right chamber of hydraulic cylinder, Pa

pp, pt supply pressure and outlet pressure, Pa.

The multi-pole mathematical model RS13pS has input variables z, $Qc1$, Fpi, pp and pt. Output variables are $F13$, $Q1$, $pc1$ and $Q3$.

In the model RS13pS slot pair 1–3 is considered in the way that all the possible combinations of open and overlapped cases of slots 1 and 3 are described.

Here mathematical models for sliding spool slot 1 (open and overlapped case) are presented.

For open slot 1 [$z >= (-\,z1 - 2{*}r)$]

Fluid jet velocity $v1$

$$v1 = \left[2/\rho{*}\mathrm{abs}(pp{-}pc1)\right]^{1/2}{*}\mathrm{sign}(pp{-}pc1)$$

Volumetric flow Q1

$$Q1 = Qc1 + Q3$$

Fluid jet force $F1$

$$F1 = -\,\rho{*}Q1{*}v1{*}\cos\theta$$

Pressure $pc1$

$$pc1 = pp - \left\{1\,/\,\left[K1^2{*}fzq(z,z1,zm,r,s)^2\right]\right\}{*}Q1^2{*}\rho\,/2$$

where

$$K1 = \mu{*}Pi{*}d{*}k,$$

$$\text{if}\left[(z >= \text{cond})\&\,(z < zm)\right]$$

$$\left[fzq\; = \;\left(y^2 + \det^2\right)^{1/2} - 2{*}r\right]$$

else

$$\left[fzq\; = \;\left(ym^2 + \det^2\right)^{1/2} - 2{*}r\right],$$

$$\text{cond} = -(z1\; + \;2{*}r),$$

$$y = z1 + z + \ 2{*}r,$$

$$ym = z1 + zm + 2{*}r,$$

$$\det = s + 2{*}r.$$

For overlapped slot 1 [$z < (-\ z1\ -\ 2{*}r)$]

Constant values:

$$K1 = \mu{*}Pi{*}d{*}k,$$

$$K3 = (12{*}\nu{*}\rho)/\left(Pi{*}s^3{*}d{*}kE{*}k\right),$$

$$K4 = Pi{*}d{*}s{*}kE{*}k.$$

Fluid jet velocity $v1$

$$v1 = Q1/K4.$$

Volumetric flow Q1

$$Q1 = Qc1 + Q3.$$

Fluid jet force $F1$

$$F1 = -\rho{*}Q1{*}v1{*}nk.$$

Coefficient $K2$ (for connecting the characteristics of overlapped and open slots):

$$K2 = K3{*}(-z{-}z1{-}2{*}r + ltr1),$$

where

$$ltr1 = \left(pp{-}pc1\right)/K3/Q1tr,$$

$$Q1tr = v1tr{*}K1{*}fzq(-za1,\ z1,\ zm,\ r,\ s),$$

$$v1tr = \left[2/\rho{*}\mathrm{abs}\left(pp{-}pc1\right)\right]^{1/2}{*}\mathrm{sign}\left(pp{-}pc1\right),$$

$$za1 = z1 + 2{*}r,$$

Pressure $pc1$ in the left chamber of hydraulic cylinder

$$pc1 = pp{-}K2{*}Q1{-}\left\{1/\left[K1^2{*}fzq(za1,\ z1,\ zm,\ r,\ s)^2\right]\right\}{*}$$

$$Q1^2{*}\rho/2.$$

Notations used above:

d diameter of sliding spool, m

s value of radial slot of sliding spool, m

k coefficient of slot length

kE eccentricity coefficient of slot ($kE = 1 \ldots 2.5$)

$ltr1$ length of transition of overlapped slot 1, m

nk coefficient in the equation for hydrodynamic force of the jet (for overlapped slot)

r radius of sliding spool edge, m

$z1$ initial value of slot 1 displacement from initial position (+ initially open, - initially closed), m

zm maximum opening value of valve slot, m

ν kinematic viscosity of fluid, m^2/s

μ discharge coefficient of valve slot

ρ fluid density, kg/m^3

θ angle between jet and valve axis (open slot), rad.

The multi-pole mathematical model RS24zS has input variables $Qc2$, $pc2$, pp and pt. Output variables are z, $F24$, Q2 and Q4.

Model RS24zS concerning slot pair 2–4 is similar to RS13pS. Method of gradual concretization is used when displacement of servo-valve ze is calculated. The concretization takes place according to absolute error of volumetric flow $Qc2$ in iteration procedure.

Parameter values both RS13pS and RS24zS:

$$d\ =\ 0.008\,\mathrm{m},\ s\ = 3.5\text{E-}6\,\mathrm{m},\ k$$
$$=\ 0.667,\ kE\ =\ 1,\ \mu\ =\ 0.7,$$

$nk\ =\ 0.51,\ r\ = 4\text{E-}6\,\mathrm{m},\ z1, \ldots,\ z4\ = -3\text{E-}5\,\mathrm{m},\ zm =$ 6.8E-4 m.

5.2. Model for dynamics of servo-drive

Multi-pole model of a servo-drive (servo-system with simple servo-valve model and with constant pressure feeding) for dynamics is shown in Figure 5. Simple servo-valve model is as a delay element.

The model of dynamics differs from the model of statics. It contains the following multi-pole models: sliding spool slot pairs RS13q and RS24q, volume elasticities of fluid veZ1 and veZ2 of cylinder chambers, piston pisY, cylinder cylY, actuator acY, torque motor feeding voltage calculator fbrex and simple servo-valve model tmnf.

Models of sliding spool slot pairs RS13q and RS24q for dynamics differ from models RS13pS and RS24zS for statics. The mathematical dependences are the same but the inputs and outputs are different.

6. Simulation environment

CoCoViLa is a flexible Java-based simulation environment that includes both continuous-time and discrete event simulation engines and is intended for applications in a variety of domains (Kotkas *et al.* 2011). The environment supports visual and model-based software development and uses automatic synthesis of programs for translating declarative specifications of simulation problems into executable code.

CoCoViLa supports a language designer in the definition of visual languages, including the specification of graphical objects, syntax and semantics of the language. CoCoViLa provides the user with a visual programming environment, which is generated from the visual language definition. When a visual scheme is composed by the user, the following steps – automatic synthesis of problem solving algorithm and generation of calculation program code – are fully automatic. The compiled program then provides a solution for the problem specified in the scheme.

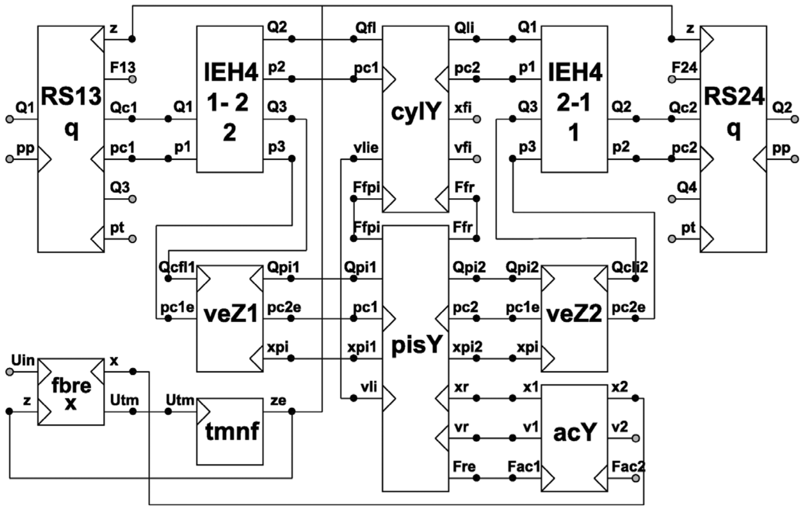

Figure 5. Multi-pole model of a servo-drive for dynamics.

Automatic synthesis of programs is a technique for the construction of programs from the knowledge available in specifications (Tyugu 1991, Matskin and Tyugu 2001). The method is based on proof search in intuitionistic propositional logic.

From a user's point of view the CoCoViLa framework consists of two components: Class Editor and Scheme Editor. The Class Editor is used for defining models of components of schemes as well as their visual and interactive aspects. The Scheme Editor is a tool for the language user. It is intended for developing schemes and for compiling programs from the schemes according to the specified semantics of a particular domain.

The environment is developed as an open-source software, its extensions can be written in Java and included into simulation packages. CoCoViLa is implemented in the Institute of Cybernetics at the Tallinn University of Technology. The CoCoViLa environment is platform-independent.

7. Simulation process organization

Using visual specifications of described multi-pole models of fluid power system components one can graphically compose models of various fluid power systems for simulating statics, steady state conditions and dynamic responses.

When simulating statics or steady state conditions fluid power system behavior is simulated depending on different constant values of input variables. Number of calculation points must be specified.

When simulating dynamic behavior, transient responses in certain points of the fluid power system caused by applied dynamic disturbances are calculated. Disturbances are considered as changes of input variables of the fluid power system (pressures, volumetric flows, load forces or moments, control signals, etc.). Typically disturbances of step, pulse or sine form can be used.

To perform integrations and differentiations in calculations the system behavior in time must be followed. Therefore, the concept of state is invoked. State variables are introduced for components to characterize the elements behavior at the current simulation step. The simulation process starts from the given initial state and includes calculation of following state (*nextstate*) from previous states (from *oldstate* and *state*). Final state (*finalstate*) is computed as a result of simulation. Program for calculating *nextstate* from *oldstate* and *state* is generated automatically by CoCoViLa.

For integrations in dynamic calculations the fourth-order classical Runge-Kutta method is used in component models.

In multi-pole mathematical models of components relations between input and output poles can be expressed by equations or by programs (Java methods) defining calculations of whatever complexity.

Integrations in multi-pole models are performed only in component models to calculate their outer variables. As component models contain limited number of output variables, integrations for solving these possible equation systems (usually no more than of 2nd-3rd order) are not very large. In such a way time consuming solving large differential equation systems can be avoided.

Time step length and number of simulation steps are to be specified individually for each specific simulation task.

We have been not concentrated to use variable time step as equation systems to be solved are not very large.

Static, steady state and dynamic computing processes are organized by corresponding process classes (static Process, dynamic Process).

When calculating next state from previous states loop dependences between outer variables (poles) of component models may occur. A special technique is used for calculating variables in loop dependences. One variable in each loop is split and iteration is used for re-calculating variables in loops.

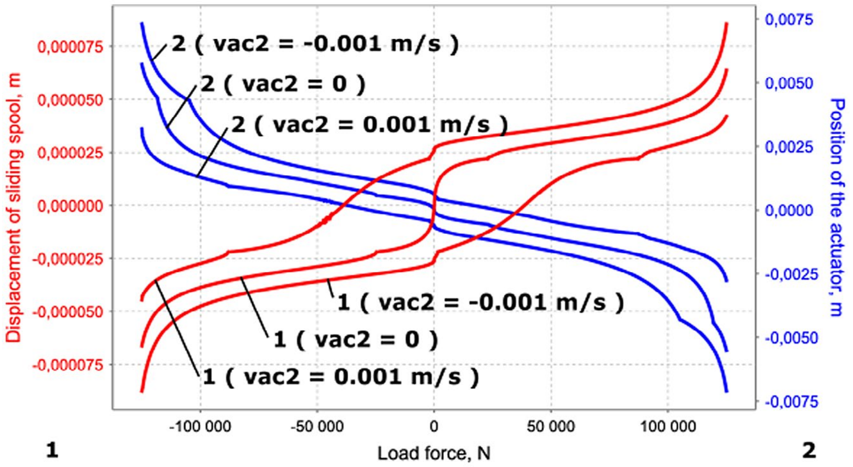

Figure 6. Simulation results of servo-drive statics and steady state conditions.

State variables and split variables must be described in component models. When building a particular simulation task model and performing simulations state variables and split variables are handled and used automatically.

When building up a simulation task scheme all the parameters of components must be provided with values using special properties windows. In all the simulation tasks input variables are given by Source classes. Time is given by Clock.

Initial values of state variables and variables requiring iterations characterize the model in the beginning of the simulation. Specifying precise initial values is not critical for statics and steady state conditions. Approximate initial values must be set. For dynamics setting initial conditions is critical to guarantee reliable transient responses calculated. Trustful initial conditions can be obtained only as a result of iterative calculations of transient responses in the case of zero disturbances which must be performed as a separate simulation.

Dynamic simulation time step must be chosen short enough in order to calculate transient responses of higher frequencies and rapid transitions. In the simulation examples concerning electro-hydraulic system under discussion time step $\Delta t = 1E - 6$ s is used.

Maximum number if iterations, adjusting factor for iterations, allowed absolute and relative errors are to be specified for calculating variables in loops.

Physical properties of working fluid (density ρ, kinematic viscosity ν and compressibility factor β) are calculated for each component at each simulation step depending on average of input and output pressure in the component. In all the simulation examples below hydraulic fluid HLP15 is used. The initial values of physical properties of fluid HLP15 at temperature 40 °C are: ρ = 865 kg/m^3, ν = 15E-6 m^2/s, β = 6.7E-10 1/Pa. Air content in fluid: vol = 0.02.

8. Simulation of the servo-drive with simple servo-valve and with constant feeding pressure

The model of the servo-drive for statics and steady state conditions does not contain the feeding system and the servo-valve is represented as a transfer coefficient.

Simulated displacement of the sliding spool z (graphs 1) and position of the actuator xS (graphs 2) are shown in Figure 6 in dependence of:

- load force $Fac2$ in range ± 1.25 E5 N,
- velocity $vac2$ of actuator in range ± 0.001 m/s.

Near to load force $Fac2$ = 0 the graphs of the sliding spool displacement (graphs 1) have transitions, subject to overlapped throttling slots of sliding spool.

For reducing non-linearity of the servo-system characteristics in the area of all overlapped slots an amplification coefficient ka of torque motor feeding voltage (designed to be a broken linear dependence) depending on the sliding spool displacement is invoked. Due the coefficient ka the graphs of position of the actuator xS (graphs 2) have approximately linear shapes in range of load force $Fac2$ = ± 7.5E4 N. In the range of force (abs($Fac2$) > 7.5E4 N the actuator position abs(xS) increases rapidly.

9. Simulation of the servo-valve

The servo-system is set up in a way that input voltage U change from −10 V to +10 V causes position of actuator to change from −0.1 to +0.1 m from the middle position.

Simulation of statics shows that the dependence of sliding spool position from the input voltage is linear.

Simulation results of dynamics of the servo-valve (see model in Figure 10) are shown in Figure 7.

The graphs are shown, which are as consequences of three different input disturbance steps of U = 1, 5 and 9 V

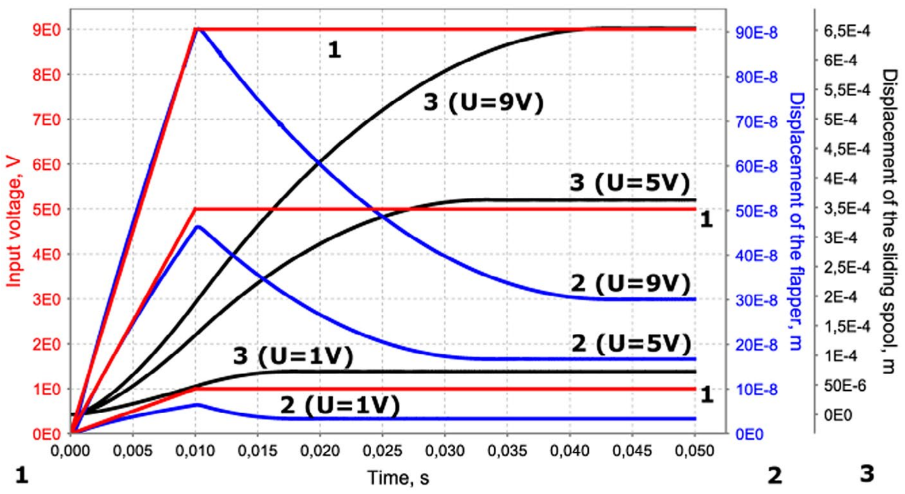

Figure 7. Simulated graphs of the electro-hydraulic servo-valve for dynamics.

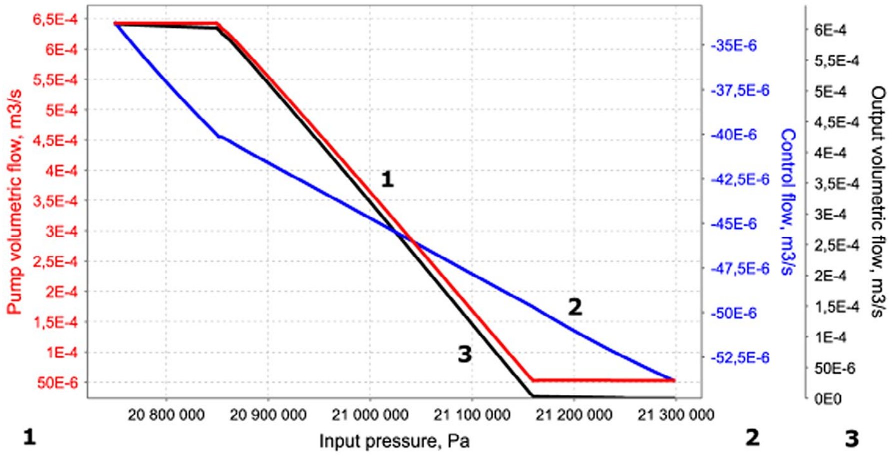

Figure 8. Simulated graphs of steady state volumetric flows of the feeding system.

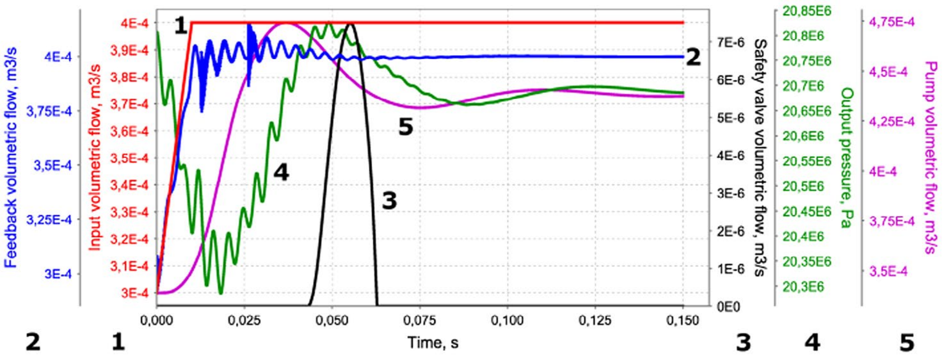

Figure 9. Simulated graphs of the feeding system for dynamics.

in time duration $t_{\min} = 0.01$ s (graphs 1). Displacements of the flapper h (graphs 2) in interval of 0–0.01 s follow the input disturbances. In the interval from 0.01 s to (0.02, 0.03, 0.04 s) the flapper takes a new position (4E-8, 17E-8, 30E-8 m) due to feedback. Displacement z of the sliding spool (graphs 3) increases from 0 to (0.75E-4, 3.7E-4, 6.5E-4 m) during (0.017, 0.034, 0.042 s).

The results are in accordance with MOOG catalog characteristics for servo-valve of such type (Moog, 1995).

10. Simulation of the feeding system

Simulation results of steady state conditions of the feeding system are shown in Figure 8.

Pump volumetric flow (graph 1) of the feeding system decreases almost linearly from maximum volumetric flow 6.4E-4 to 50E-6 m^3/s in pressure range from 20.85 to 21.16E6 Pa. Control volumetric flow (graph 2) decreases almost linearly from −34E-6 to −54E-6 m^3/s.

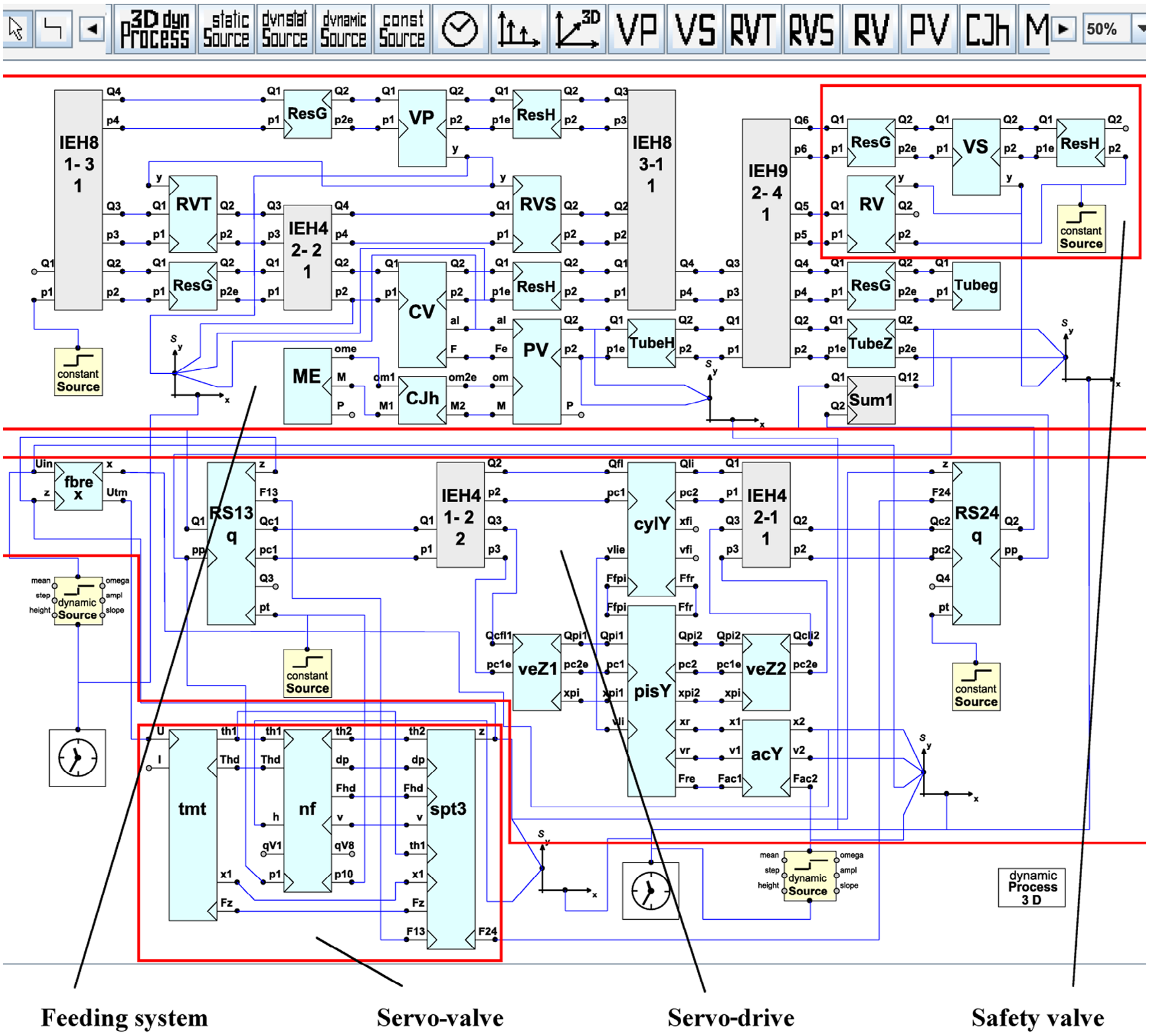

Figure 10. Simulation task description of entire servo-system for dynamics.

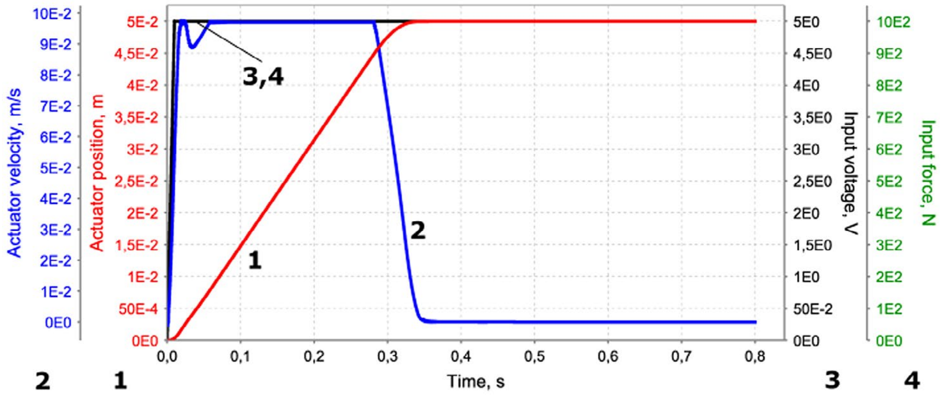

Figure 11. Simulated positioning and velocity of actuator.

Output volumetric flow (graph 3) depends on pump flow (graph 1).

Simulation results of dynamics of the feeding system (see model in Figure 10) are shown in Figure 9.

Step disturbance of input volumetric flow 1E-4 m^3/s (mean flow value 3E-4 m^3/s, step time 0.01 s) (graph 1) causes damped oscillations of the output pressure (graph 4) of two frequencies (~13 and ~260 Hz). Feedback

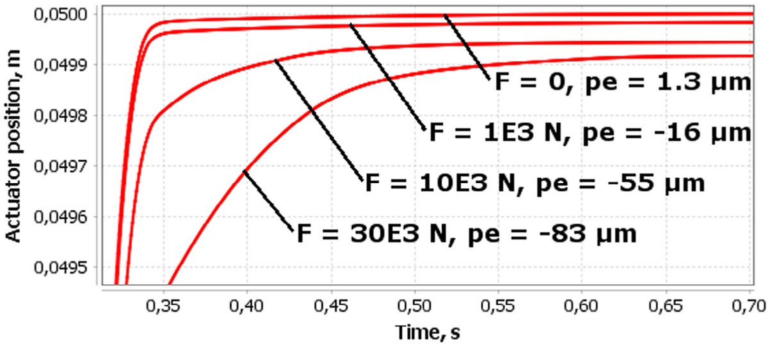

Figure 12. More precise positioning graphs of actuator.

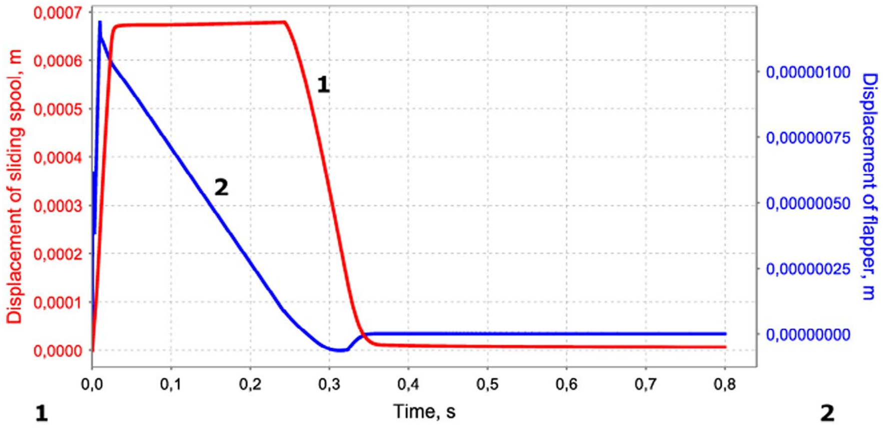

Figure 13. Simulated displacements of flapper and sliding spool of servo-valve.

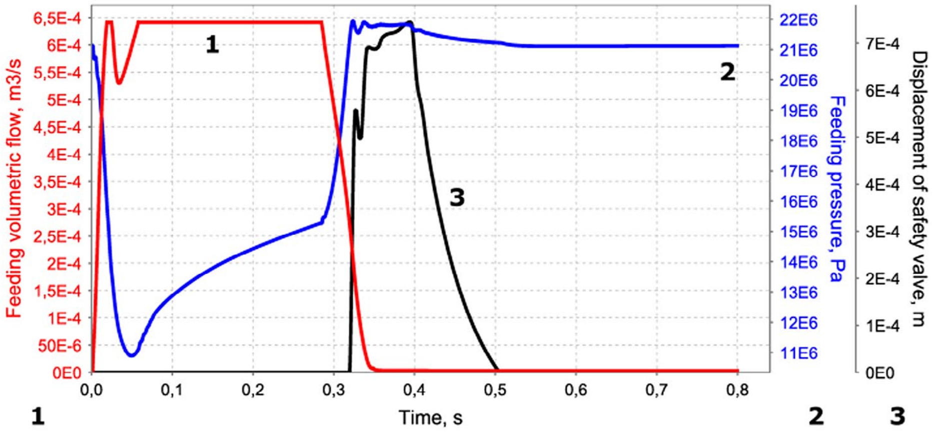

Figure 14. Simulated feeding volumetric flow, feeding pressure and displacement of safety valve.

volumetric flow at the left end of the output tube (graph 2) follows the input volumetric flow, carrying higher frequency oscillations till 0.05 s.

Output pressure (graph 4) growth to value 20.83E6 Pa causes safety valve to open and output pressure to drop down to 20.75E6 Pa which causes safety valve to close (graph 3). Pump volumetric flow (graph 5) follows step disturbance of the input volumetric flow with some delay and damped oscillation.

11. Simulation of the entire servo-system

The model of the entire servo-system contains models of all three subsystems. As a result of simplifications the model for statics of servo-drive with simple servo-valve and with constant feeding pressure (see Figure 4) can be considered as a model for the entire servo-system statics. Therefore, simulation of statics of the servo-drive can be considered as simulation for statics of the entire servo-system (see Chapter 8).

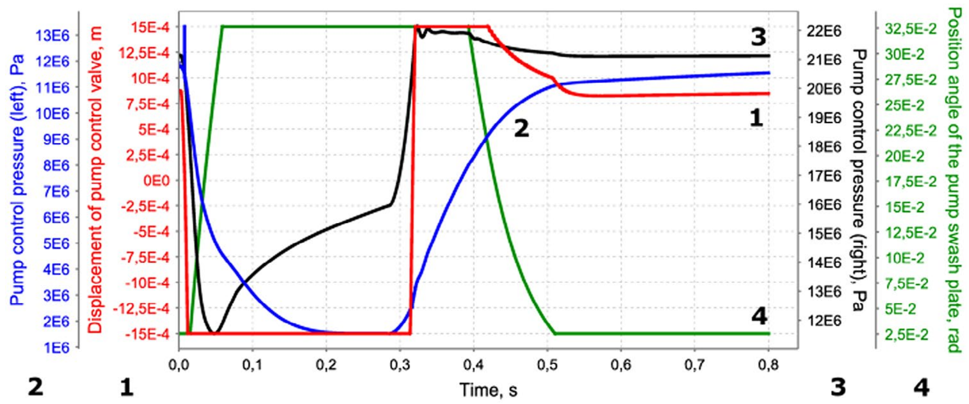

Figure 15. Simulated graphs of pump regulating system.

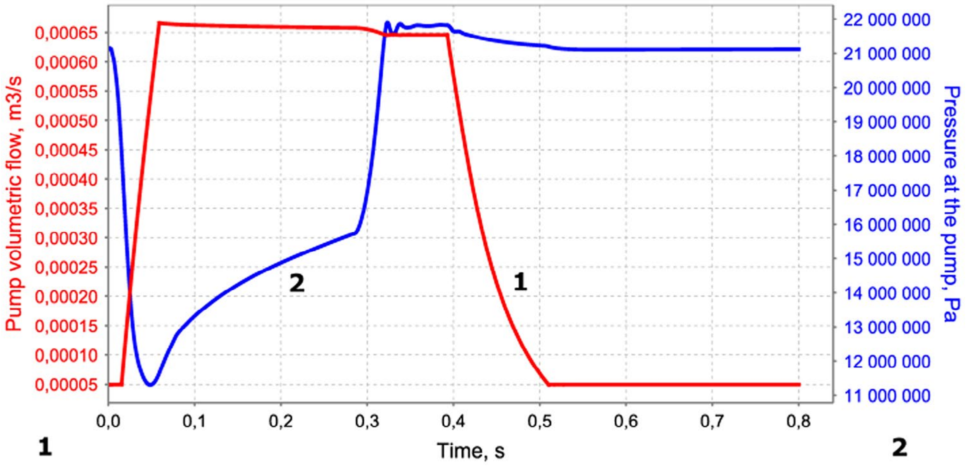

Figure 16. Simulated graphs of pump.

The model of dynamics of the entire servo-system contains three subsystems: servo-drive, servo-valve and feeding system. The simulation task of the whole servo-system is shown in Figure 10.

Results of simulation of dynamic responses caused by applying the servo-system simultaneously both step control voltage U_{in} = 5 V (step time 0.01 s) and actuator step load force *Fac*2 = 1E3 N (step time 0.01 s) as input disturbances are shown in Figures 11–16.

To assure faster and more precise positioning of the actuator a feedback correction by a non-linear differential regulator of the voltage of torque motor of servo-valve is used in dynamics.

In Figure 11 the actuator (graph 1) moves 0.05 m in time 0.35 s. At the beginning of the process the actuator moves with velocity (graph 2) defined by the maximal volumetric flow of the pump. Thus the fastest transient response is achieved. After 0.28 s the actuator slows down. Step voltage (graph 3) and step load force (graph 4) are as inputs.

The servo-system parameters have been chosen to ensure that the actuator position 0.05 m corresponds to the input voltage 5 V in the ideal case. Actuator positioning error is mainly caused by actuator load force. In Figure 12 simulated precise positioning of actuator (graph 1 in Figure 11) is shown for four load force step values.

Step force *F* = 0 causes positioning error *pe* = 1.3 μm, step force *F* = 30E3 N causes *pe* = −83 μm. Positioning takes up to 0.7 s, according to load force step value. Positioning error could be decreased by invoking feedback by load force.

In Figure 13 the sliding spool rapidly moves from 0 to maximum 680 μm (graph 1). Until time moment 0.25 s the sliding spool stays at the maximum position. Then it returns and takes final position 10 μm. The flapper moves fast from 0 to 1.2 μm (graph 2). Then it returns to initial zero position after a little under run.

In Figure 14 the feeding volumetric flow (graph 1) rapidly achieves maximum value 6.4E-4 m^3/s. After 0.29 s it decreases to zero at 0.35 s. Feeding pressure (graph 2) decreases from initial value 21E6 to 11E6 Pa during 0.05 s. Then it increases to maximum value determined by safety valve, at 0.32 s. Finally it returns to the initial value. At time moment 0.32 s when the pressure achieves safety valve setting pressure value 21.4E6 Pa the safety valve opens (graph 3). After few oscillations the safety valve closes at 0.50 s.

In Figure 15 initially opened pump control valve VP (graph 1) closes rapidly, then at 0.32 s it opens maximal due to pump control pressure right (graph 3). After 0.43 s it begins to close and achieves stable position at 0.53 s. Pump control pressure (left) (graph 2) drops from initial value to 1.6E6 Pa, then begins to grow at 0.3 s due to pump control pressure right (graph 3) and achieves stable value at ~11.3E6 Pa. Pump control pressure (right) (graph 3) drops from initial value to 11.5E6 Pa, then begins to grow and achieves value 22E6 Pa at time 0.32 s. Later it drops to stable value 21.1E6 Pa. Pump swash plate (graph 4) rapidly turns from minimal position to maximal. After 0.4 s it turns back to the initial minimal position at 0.51 s. The minimal position angle 2.73E-2 rad is set up to ensure sufficient feeding the pump control system.

In Figure 16 the pump volumetric flow (graph 1) almost follows the shape of position angle of the pump swash plate (graph 4 on Figure 15). Pressure at the pump (graph 2) almost follows the shape of feeding pressure (graph 2 on Figure 14) being slightly higher. Some dependence of pump volumetric flow from pressure at the pump can be observed.

Multi-pole model of the electro-hydraulic servo-system considered in the paper contains 37 components and 28 variables which must be calculated by integrations in multi-pole models of components. The number of variables in loop dependences which required special technique of iterative recalculating is 20.

The simulations performed take 15 min computing time on the computers of average processing performance.

12. Conclusions

In the paper multi-pole modeling and simulation of the electro-hydraulic servo-system has been considered.

Multi-pole mathematical models are used that enable adequately describe physical processes in hydraulic and mechanical systems. Both direct actions and feedbacks are expressed models. Using multi-pole modeling and visual representation of models gives a user a convenient and efficient tool for describing and manipulating models of large fluid power systems.

Simulations are performed using an intelligent programming environment CoCoViLa providing feature of automatic synthesis of programs from the knowledge available in visual model of a fluid power system.

Due to using multi-pole models solving large differential equation systems during simulations are avoided.

A method has been described that allows perform simulations on a model of a fluid power system containing loop dependences between components.

Results of simulations of three subsystems (hydraulic servo-drive with feedback and regulator, electro-hydraulic servo-valve and constant feeding with variable displacement axial piston pump) and the entire servo-system have been presented and discussed.

The modeling and simulation procedure proposed in the paper is an efficient and powerful tool for design of complicated fluid power systems and technical chain systems of any kind.

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