USING MULTI-POLE MODELING AND INTELLIGENT SIMULATION IN DESIGN OF A HYDRAULIC DRIVE

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ABSTRACT

An approach based on multi-pole modeling and intelligent simulation is proposed for design of fluid power systems. The method is explained on the example of modeling and simulation of a hydraulic drive with two-directional flow regulating valve. Multi-pole mathematical model of a hydraulic drive is presented. An intelligent visual simulation environment CoCoViLa supporting declarative programming in a high-level language and automatic program synthesis is used as a tool. Simulation examples of steady state conditions and dynamics of the hydraulic drive are presented and discussed.

Keywords: hydraulic drive, multi-pole model, intelligent programming environment, simulation.

1. INTRODUCTION

Most of modeling and simulation tools in existence such as MATLAB/Simulink, SimHydraulics™, ITI SimulationX, DSHplus, Dymola, HOPSAN, VisSim, AmeSim, 20-Sim, DYNAST, MS1™ referred in (Grossschmidt and Harf 2009a) and 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 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 analysis and system synthesis frequently simplified, 3rd…5th order differential equation systems are used.

In the current paper an approach is proposed, which is based on using multi-pole models with different oriented causalities (Grossschmidt and Harf 2009a) for describing components of different complexity. When modeling fluid power systems, elementary components are hydraulic resistors, tubes, hydraulic interface elements, valves, pumps, motors, pistons, etc. (Grossschmidt and Harf 2010). Hydraulic control valves of different types (Harf and Grossschmidt 2014) are described using elementary components.

During simulations calculations are performed in level of elementary components considering structure of the entire system. In such a way solving large equation systems can be avoided. Therefore, multi-pole models of large systems do not need considerable simplification.

Modeling and simulation of a hydraulic drive including a two-directional flow regulating valve is considered as an example of applying proposed methodology.

2. MULTI-POLE MODELS

In general a multi-pole model represents mathematical relations between several input and output variables (poles). The nearest to physical nature of various technical systems is using multi-pole mathematical models of their components and subsystems.

In hydraulic and mechanical systems variables are usually considered in pairs (effort and flow variable). Multi-pole models enable to express both direct actions and feedbacks.

Each component of the system is represented as a multi-pole model having its own structure including inner variables, outer variables (poles) and relations between variables.

Using multi-pole models allows describe models of required complexity for each component. For example, a component model can enclose nonlinear dependences, inner iterations, logic functions and own integration procedures. Multi-pole models of system components can be connected together using only poles. It is possible directly simulate statics or steady state conditions without using differential equation systems.

The multi-pole model concept enables us to describe mathematical models graphically which facilitates the model developing.

3. 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, Ojamaa, Grigorenko, Maigre, Harf and Tyugu 2011). The environment supports visual and model-based software development and uses structural synthesis of programs (Matskin and Tyugu 2001) for translating declarative specifications of simulation problems into executable code.

CoCoViLa (Figure 1) 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 automatically generated from the visual language definition. When a visual scheme is composed by the user, the following steps – parsing, planning and code generation – are fully automatic. The compiled program then provides a solution for the problem specified in the scheme, and the results it provides can be feedback into the scheme, thus providing interactive properties.

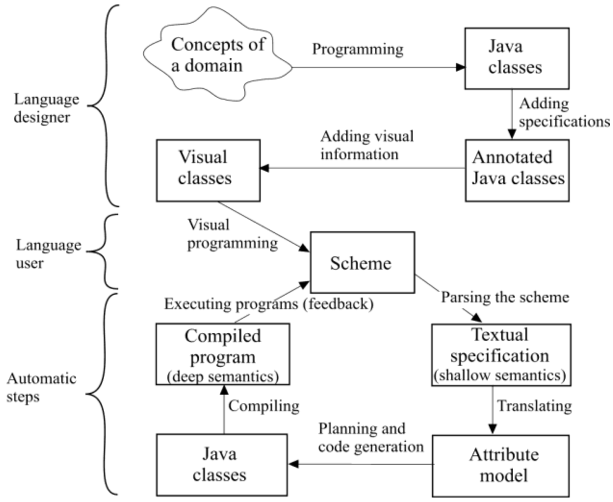

Figure 1. Technology of visual programming in CoCoViLa

Structural synthesis of programs is a technique for the automatic construction of programs from the knowledge available in specifications. The method is based on proof search in intuitionistic propositional logic.

The synthesizer (planner) determines computational paths from initial variables to required goal variables (i.e., tries to solve a given computational problem "find values of V from given values of U", where U and V are sets of input and output variables).

Designer do not need to deal with programming, he can use the models with prepared calculating codes. It is convenient to describe simulation tasks visually, using prepared images of multi-pole models with their input and output poles.

4. 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 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 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.). Time step length and number of steps are to be specified. For integrations in dynamic calculations the fourth-order classical Runge-Kutta method is used in component models.

Static, steady state and dynamic computing processes are organized by corresponding process classes (static Process, dynamic Process). To follow the system behavior, the concept of state is invoked. State variables are introduced for each component to characterize its behavior at the current simulation step.

A simulation task requires sequential computing states until some satisfying final state is reached. A final state can be computed from a given initial state if a function exists that calculates the next state from known previous states. This function is to be synthesized automatically by CoCoViLa planner.

A special technique is used for calculating variables in loop dependences that may appear when multi-pole models of components are connected together. One variable in each loop is split and iteratively recomputed to find it value satisfying the loop dependence.

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 used automatically.

5. HYDRAULIC DRIVE WITH TWO–DIRECTIONAL FLOW REGULATING VALVE

Functional scheme of a hydraulic drive with three-directional flow regulating valve in cylinder inlet is shown in Figure 2.

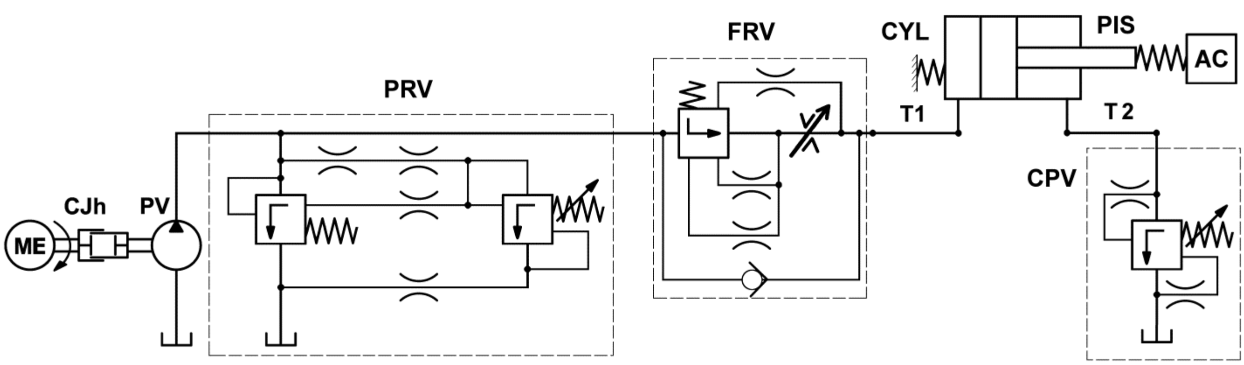

Figure 2: Functional scheme of a hydraulic drive with two-directional flow regulating valve

The pump **PV** is driven by electric motor **ME** through clutch **CJh**. The outlet of the pump is provided with pilot operated pressure relief valve **PRV** and two-directional flow regulating valve **FRV**.
Tubes **T1** and **T2** are located in inlet and outlet of hydraulic cylinder **CYL**. Piston and actuator are denoted respectively as **PIS** and **AC**. Constant pressure in outlet of the cylinder is ensured by pressure valve **CPV**.
In the next section two-directional flow regulating valve FRV is considered in detail to demonstrate how it is described by multi-pole and mathematical models.

6. TWO-DIRECTIONAL FLOW REGULATING VALVE

Flow regulating valves (Murrenhoff 2005, Gebhardt, Will and Nollau 2011) are used when the working speed of hydraulic drive should remain almost constant in case of different loads at the user.
Two-directional flow regulating valve FRV in Figure 2 contains adjustable throttle and connected in sequence pressure compensator ensuring constant pressure drop in the throttle.

In Figure 3 two-directional flow regulating valve of Mannesmann Rexroth is shown.

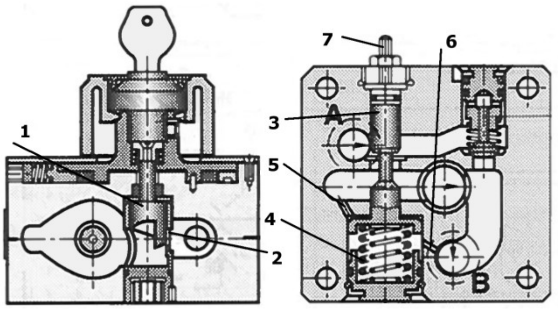

Figure 3: Two-directional flow regulating valve

The valve consists of throttle pin 1 with orifice 2, normally open pressure compensator spool 3 with two springs 4, bores 5, 6 to the spool surfaces and stroke limiter 7.
Multi-pole model for dynamics of a two-directional flow regulating valve FRV is shown in Figure 4.

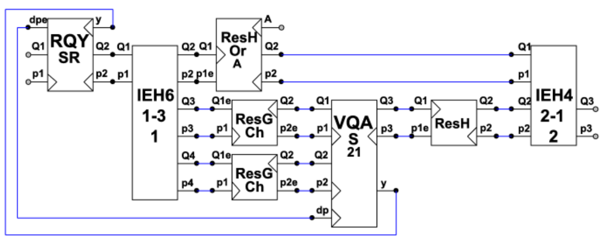

Figure 4: Multi-pole model of a two-directional flow regulating valve

Multi-pole models: **RQYSR** – pressure compensator slots, **ResHOrA** – regulating spool slot, **VQAS21** – pressure compensator spool, **ResGCh, ResH** – cushioning resistors, **IEH** – interface elements.

6.1. Mathematical Models

6.1.1. Regulating throttle orifice ResHOrA

Inputs: pressure $p2$, volumetric flow $Q1$, area of the regulating throttle orifice A.
Outputs: pressure $p1$e, volumetric flow $Q2$.

Output pressure

$$p1e = p2 + RT * abs(Q1) * Q1.$$

Resistance at turbulent flow

$$RT = \rho / (2 * \mu^2 * A^2),$$

where
ρ fluid density,
μ discharge coefficient.

Output volumetric flow

$$Q2 = Q1.$$

6.1.2. Pressure compensator spool VQAS21

Inputs: pressures $p1$, $p2$, $p3$, pressure drop dp in valve flow-through notches.
Outputs: volumetric flows $Q1$, $Q2$, $Q3$, displacement of valve y.

Pressure compensator spool areas:

$$A1 = \pi * d1^2 / 4,$$
$$A2 = \pi * (d2^2 - d1^2) / 4,$$
$$A3 = \pi * d2^2 / 4,$$

where
d1 diameter of the spool,
d2 diameter of the spool head.

Stiffness of springs:

$$c1 = G * ds1^4 / (Ds1^3 * n1 * 8),$$
$$c2 = G * ds2^4 / (Ds2^3 * n2 * 8),$$

where
G shear modulus,
ds1, ds2 diameters of spring wires,
Ds1, Ds2 diameters of springs,
n1, n2 numbers of turns of springs.

Sum of spring stiffness

$$c = c1 + c2.$$

Force to pressure compensator spool

$$F = A1*p1 + A2*p2 - A3*p3.$$

Displacement of the pressure compensator spool:

$$y1 = 1 / (F / c - fV0),$$

where
fV0 preliminary compressibility of spring.

Output pressure compensator spool slot width

$$y = y0 - y1,$$

where
y0 initial width of spool notches.

Difference of actuator velocity dv for integration used in Runge-Kutta method is calculated by formula

$$dv=(\Delta t /m)*(F - (y1+ fV0)*c - (Ff0 + kfr*(p1 + p2) /2)*sign(v) - h*v),$$

where
Δt time step,

m mass,
Ff0 constant part of friction force,
kfr coefficient of friction force,
v velocity of valve,
h damping coefficient.

Difference of spool displacement

$$dy = \Delta t * v.$$

Output volumetric flows:

$$Q1 = A1*v, \; Q2 = A2*v, \; Q3 = A3*v.$$

6.1.3. Pressure compensator slots RQYSR

Inputs: pressures *p*1, *p*2e, displacement *y* of the pressure compensator spool.
Outputs: volumetric flows *Q*1, *Q*2.

Initially, pressure compensator model strictly followed the build-up of the flow regulating valve (Fig. 3). Simulation results showed some instability in behaviour of the valve. To achieve more precise and smoother control over flow, pressure compensator slot of changed shape was invoked. The conical poppet of the compensator spool was replaced by inclined triangular notches.

Pressure compensator slots area

$$A \approx n * (y * \sin(\beta * \pi / 180) - h/2)*y,$$

$$h = [d/2 - (d^2/4 - y^2/4)^{1/2}] *\cos(\beta * \pi / 180),$$

where
n number of notches.
β inclined angle of triangular notches,
h height of segment-shaped portion of slot,
d diameter of spool.

Output volumetric flows:

$$Q1 = \mu *A* (2 * \mathrm{abs}(p1 - p2e) / \rho) * \mathrm{sign}(p1 - p2e),$$

$$Q2 = Q1,$$

where
μ discharge coefficient,
ρ fluid density.

7. SIMULATION OF STEADY STATE CONDITIONS

Simulation task for steady state conditions of a hydraulic drive with two-directional flow regulating valve is shown in Figure 5.

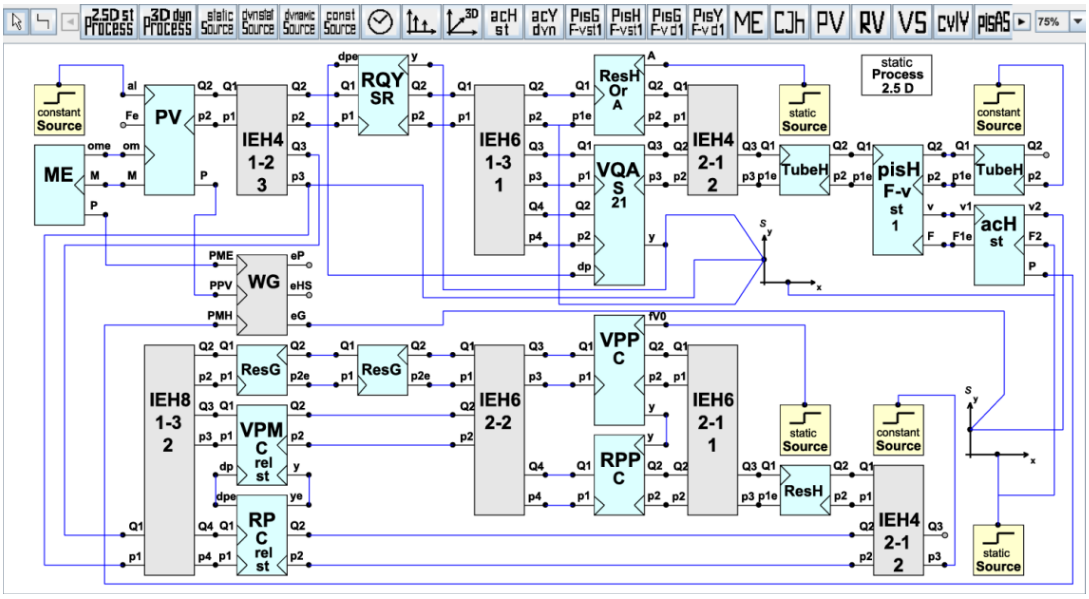

Figure 5: Simulation task of a hydraulic drive with two-directional flow regulating valve for steady state conditions

Multi-pole models: **ME-** electric motor, **PV -** axial-piston pump, **RQYSR** – pressure compensator slots, **ResHOrA –** regulating throttle orifice, **VQAS21** – pressure compensator spool, **VPPC –** pilot poppet valve with spring, **RPPC –** pilot poppet valve slot, **VPMCrelst –** main poppet valve with spring, **RPCrelst –** main poppet valve slot, **pisH_F-v_st1** – piston, **acHst** – actuator, **TubeH** – tubes, **ResG, ResH** – resistors, **IEH –** interface elements, **WG** – efficiency coefficient calculator (Grossschmidt and Harf 2009b, Grossschmidt and Harf 2014).

Inputs: outlet pressures *p*2, regulating orifice area *A*, constant position angle *al* of the pump regulating swash plate.

Outputs: actuator velocity *v*2, efficiency coefficient *eG* of the entire hydraulic drive.

Simulation manager*:* static Process 2.5D.

The following parameter values are used for steady state simulations.

For pilot operated pressure control valve the parameter values are shown in (Harf and Grossschmidt 2014).

For flow regulating valve:

for **ResHOrA:** μ=0.7;

for **VQAS21:** d1=0.01 m, d2=0.03 m, ds1=0.0024 m, Ds1=0.022 m, n1=5, ds2=0.0024 m, Ds2=0.014 m, n2=4, G=8e11 N/m, fV0=0.004 m, kfr=2e-9 N/Pa, Ff0=3 N;
for **RQYSR:** n=3, β=30 deg, d=0,01 m, μ=0.8.

For tubes, piston and actuator:

for **TubeH:** d=0.019 m, l=2 m;

for **pisH_F-v_st1:** piston diameter dpi = 0.10 m, diameters of rods dr1=0 m, dr2=0.056 m, piston friction force Ffpi=100 N, rod friction force Ffr=50 N;

for **acHst:** Ffr=100 N, h=3e4 Ns/m.

Results of simulation of steady state conditions depending on the pressure compensator slots area for three different values of the load force F = (0.1, 0.6, 1.1) e5 N are shown in Figure 6 and Figure 7.

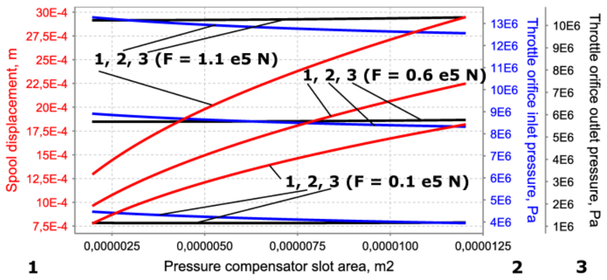

Figure 6: Graphs of simulations of steady state conditions

Pressure compensator spool displacement (graphs 1) is bigger in both cases pressure compensator slot area and load force values are bigger. Throttle orifice inlet and outlet pressures (graphs 2 and 3) depend only on the load force.

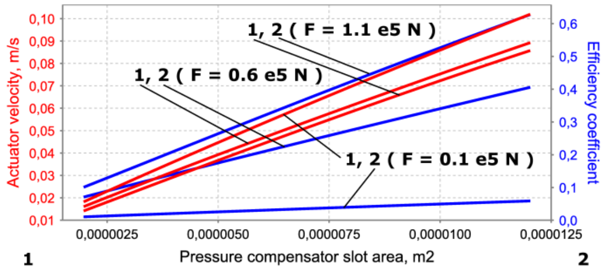

Figure 7: Graphs of simulations of steady state conditions

Actuator velocity (graphs 1) linearly depends on the pressure compensator slot area. Dependence on the load force is marginal. Efficiency coefficient (graphs 2) depends on both pressure compensator slot area and load force. Efficiency coefficient is higher in case of bigger actuator velocity and load force.

Results of simulation of steady state conditions depending on the load force for three different values of the regulating orifice area A = (12, 7, 2) e-6 m^2 are shown in Figure 8 and Figure 9.

In Figure 8 pressure compensator spool displacement (graphs 1) is bigger in both cases pressure compensator slot area and load force values are bigger. Feeding pressure (graphs 2) is slightly lower in case of higher load force. Throttle orifice inlet pressure (graphs 3) linearly depends on the load force.

In Figure 9 actuator velocity (graphs 1) is kept almost constant on load forces lower 80 kN. Efficiency coefficient (graphs 2) is maximal on load force 110 kN.

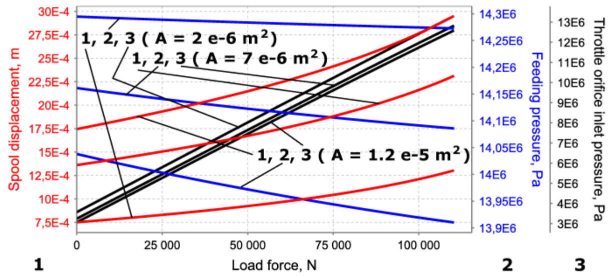

Figure 8: Graphs of simulations of steady state conditions

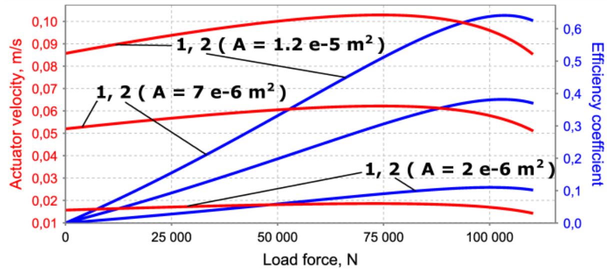

Figure 9: Graphs of simulations of steady state conditions

8. SIMULATION OF DYNAMICS

Simulation task of a hydraulic drive with three-directional flow regulating valve for dynamics is shown in Figure 10.

Additional and different multi-pole models from steady state conditions: **CJh** – clutch, **VPMC –** main poppet valve with spring, **RPCreldyn –** main poppet valve slot, **TubeH**, **TubeY –** inlet and outlet tube, **pisY–** piston, **cylY** – cylinder, **veZ1**, **veZ2 –** volume elasticities of cylinder chambers, **acY** – actuator, **ResGCh** – resistors.

Inputs: constant outlet pressures *p*2, load force *Fac*2, regulating orifice area *A,* constant position angle *al* of the pump regulating swash plate.

Outputs: actuator velocity *v*2, outlet volumetric flows *Q*2, cylinder position *xfi*.

Simulation manager*:* dynamic Process3D.

The following additional parameter values are used in dynamic simulations.

For flow regulating valve:

for **VQAS21:** m=0.04 kg, h=20 Ns/m;
for **ResYOrA:** A=1e-5 m^2;
for **ResGCh1:** d= 0.0012 m, l=0.02 m;
for **ResGCh2:** d= 0.0015 m, l=0.01 m;
for **ResH:** d= 0.0012 m, l=0.03 m.
For tubes, cylinder, piston and actuator:

for **TubeH**, **TubeY:** d=0.019 m, l=2 m;
for **cylY:** fixing elasticity efi= 1e-8 m/N, m=20 kg, hfi=5e5 Ns/m, Fffi= 20 N;
for **veZ1**, **veZ2:** lengths of chambers l1=l2=0.2 m;
for **pisY:** elasticity of piston rod er2=1e-10 m/N;
For **acYdyn:** m=20 kg, h= 3e3 Ns/m.

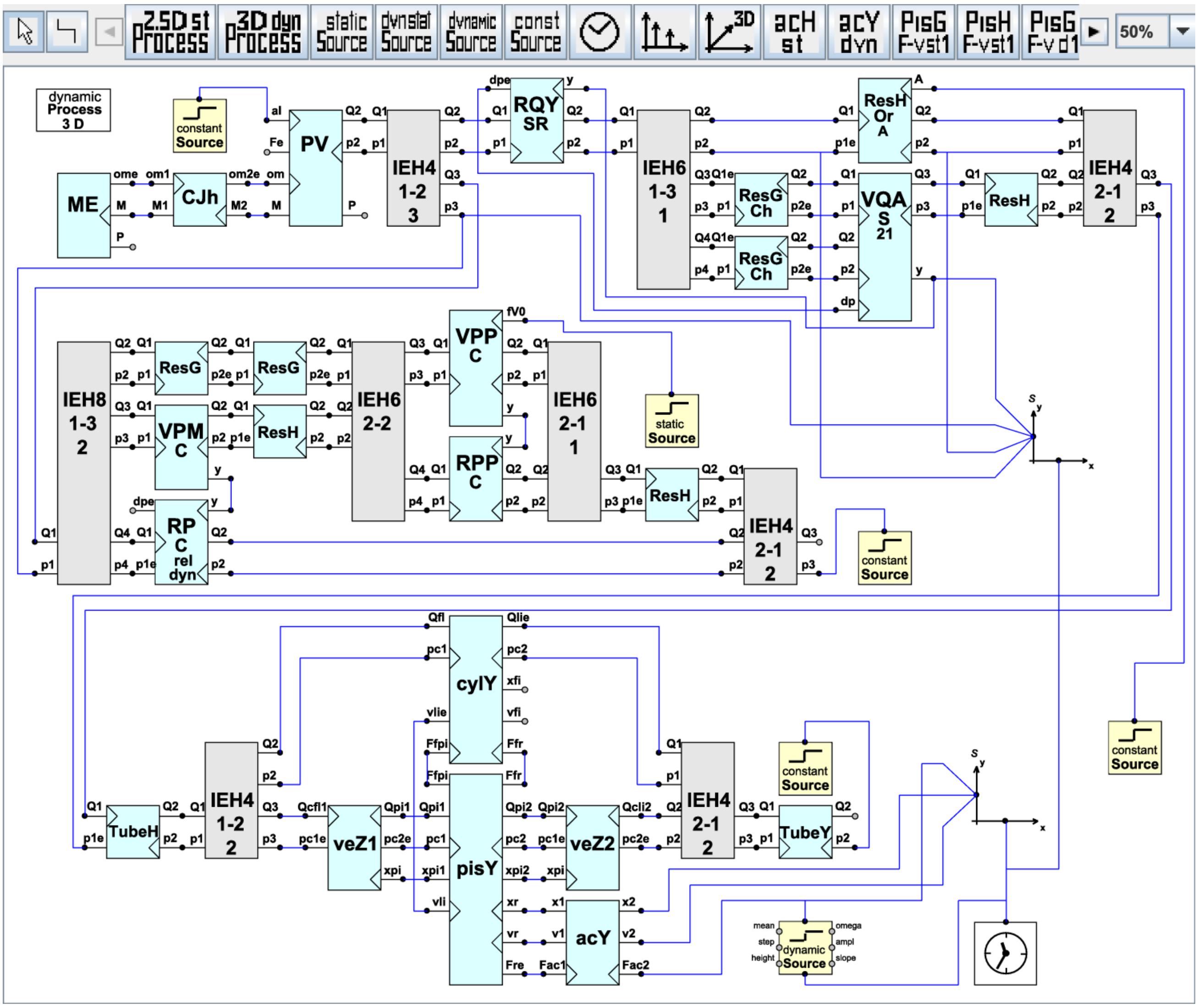

Figure 10: Simulation task of dynamics of a hydraulic drive

Results of simulation of dynamic responses caused by step change of regulating throttle orifice area *A* from 1e-5 m^2 to 1.1e-5 m^2 (step time 0.01 s) (graph 1 in Figure 11) as input disturbance are shown in Figure 11 and Figure 12. Load force *Fac2* is taken of constant value 1e4 N.

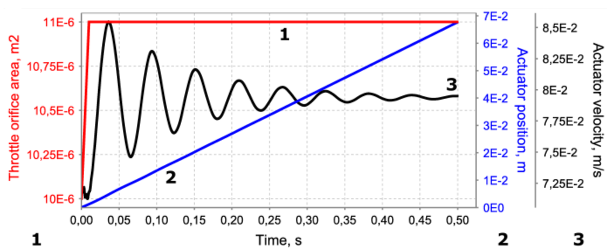

Figure 11: Graphs of actuator

Actuator moves linearly (graph 2), actuator velocity (graph 3) reacts by damped oscillation. The process lasts 0.5 s.

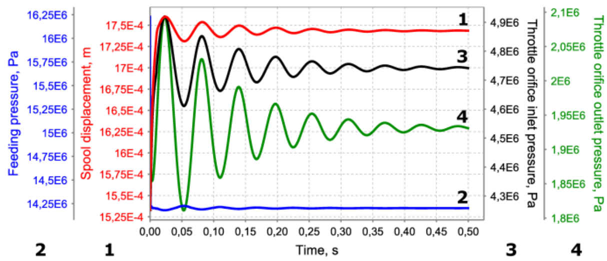

Figure 12: Graphs of flow regulating valve

Pressure compensator spool (graph 1) takes a new position after damped oscillations. Feeding pressure (graph 2) remains almost constant. Throttle orifice inlet and outlet pressures (graphs 3 and 4) oscillate synchronously and are damped in 0.5 s. Pressure drop in throttle orifice remains almost constant.

Results of simulation of dynamic responses caused by applying the hydraulic drive actuator step load force *Fac2* from 0 to 5E3 N (step time 0.01 s) (graph 1 in Figure 13) as input disturbance are shown in Figure 13 and Figure 14. Regulating throttle orifice area *A* is taken of constant value 1e-5 m^2.

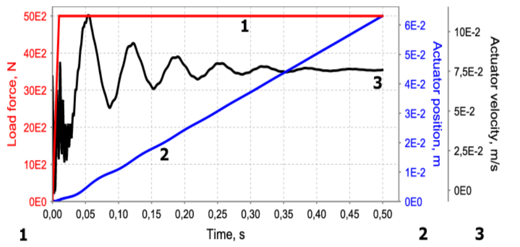

Figure 13: Graphs of actuator

Actuator moves linearly (graph 2), actuator velocity (graph 3) reacts by damped oscillation of two different frequencies. The process lasts 0.5 s.

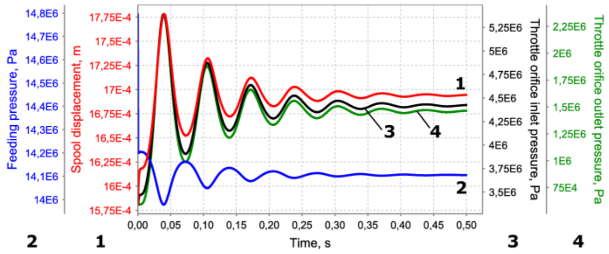

Figure 14: Graphs of flow regulating valve

Pressure compensator spool (graph 1) takes a new position after damped oscillations. Feeding pressure (graph 2) remains almost constant. Throttle orifice inlet and outlet pressures (graphs 3 and 4) oscillate synchronously and are damped in 0.5 s. Pressure drop in throttle orifice remains almost constant.

CONCLUSION

A simulation methodology for design of fluid power systems based on multi-pole modeling and intelligent simulation has been discussed in the paper. Modeling and simulation of a hydraulic drive with two-directional flow regulating valve was considered as an example.

As a result of the experiments initially used two-directional flow regulating valve of Mannesmann Rexroth was modified. The conical poppet of the compensator spool was replaced by several inclined triangular notches. Also, it was noticed that parameters of control valves such as stiffness and preliminary compressibility of springs, values of hydraulic resistors and damping coefficients required precise adjustment for each particular case to attain the best performance of the hydraulic drive.

Control valve models e.g. those we described and used in the paper can be used when composing models of fluid power systems whatever type.

The methodology described and applied for modeling and simulation of hydraulic drive is meant to be used at the first stage of design of fluid power systems. This enables to try out different configurations and find first approximate parameters in development of fluid power systems. Results of simulations are meant to be as basis for the further experimental stages of the design process.

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