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USING MULTI-POLE MODELING AND INTELLIGENT SIMULATION IN DESIGN OF A HYDRAULIC DRIVE WITH TWO–DIRECTIONAL FLOW REGULATING VALVE (PART 2)

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Abstract: *The paper presents simulation of a hydraulic drive containing pilot operated pressure relief valve and two-directional flow regulating valve. Multi-pole mathematical models of components were presented in Part 1 of the paper. An intelligent simulation environment CoCoViLa with feature of automatic synthesis of calculation algorithms is used as a tool. Simulation examples for calculating steady state conditions and dynamic transient responses are presented and discussed.*

Key words: *hydraulic drive, pilot operated pressure relief valve, two-directional flow regulating valve, intelligent programming environment, simulation.*

1. INTRODUCTION

An overview and a brief analysis of existing simulation tools for fluid power systems are presented in [1]. Most of tools are based on composing and solving differential equation systems.

When simulating non-elementary fluid power systems it is difficult to ensure that all the dependences are present and described correctly.

In the current paper an approach is used, which is based on using multi-pole models with different oriented causalities [1]. Multi-pole models enable to describe fluid power systems more adequately, taking into account straight and backward impacts of flow and potential variables.

It is feasible to build up model of a fluid power system in structural way, defining significant inner and outer variables of system components and relationships between variables.

During simulations calculations are performed at the level of components considering structure of the entire system. In such a way solving large equation systems can be avoided.

A visual simulation environment CoCoViLa with feature of automatic synthesis of calculation algorithms is used as a tool.

Proposed methodology is used in modeling and simulation of a hydraulic drive containing a two-directional flow regulating valve in cylinder outlet.

2. SIMULATION ENVIRONMENT

CoCoViLa is a software tool for model-based software development with a visual language support that performs automatic synthesis of programs from logical specifications [2, 3].

CoCoViLa consists of two components, Class Editor and Scheme Editor. The Class Editor is used for implementing visual languages for different problem domains. This is done by defining models of language components as well as their visual and interactive aspects. The Scheme Editor is a tool for drawing schemes, compiling and running tasks defined by a scheme and a goal.

Using a visual simulation environment CoCoViLa enables to describe multi-pole models graphically which facilitates the model developing. When simulating automatic synthesis of calculation algorithms provided by CoCoViLa is used. This allows focus on designing models of fluid power systems instead of constructing and solving simulation algorithms.

3. SIMULATION PROCESS ORGANIZATION

Typically two kinds of simulations, static and/or steady state conditions and dynamic transient responses are considered.

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.

Integrations in multi-pole models are performed only in component models to calculate their outer variables. For integrations during dynamic calculations the fourth-order classical Runge-Kutta method is used.

As component models contain limited number of output variables possible equation systems are mostly no more than of 2nd-3rd order.

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

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

A special method has been used that allows perform simulations on models containing feedbacks between components. The method is based on hierarchical structure of entire model and encapsulation of calculations in models of subsystems and components. One variable in a feedback loop is split into two variables, evaluated by an initial approximate value and iteratively recomputed. Recomputing algorithm is automatically synthesized by CoCoViLa. 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. In all the simulation tasks input variables must be evaluated by Source classes. Time is controlled by Clock.

Initial values of state variables and variables requiring iterations characterize the model in the beginning of the 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 fluid power systems under discussion time step $\Delta t = 1e\text{-}6$ 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 at each simulation step depending on average of input and output pressure in the component. In all the simulations below hydraulic fluid HLP46 is used. The initial values of physical properties of airless fluid HLP46 at zero pressure and at temperature 40°C are: ρ = 873 kg/m^3, ν = 46e-6 m^2/s, β = 6.1e-10 1/Pa. Air content in fluid: vol = 0.02.

4. SIMULATION OF STEADY STATE CONDITIONS

Simulation task of steady state conditions of a hydraulic drive with two-directional flow regulating valve in cylinder outlet is shown in Fig. 1.

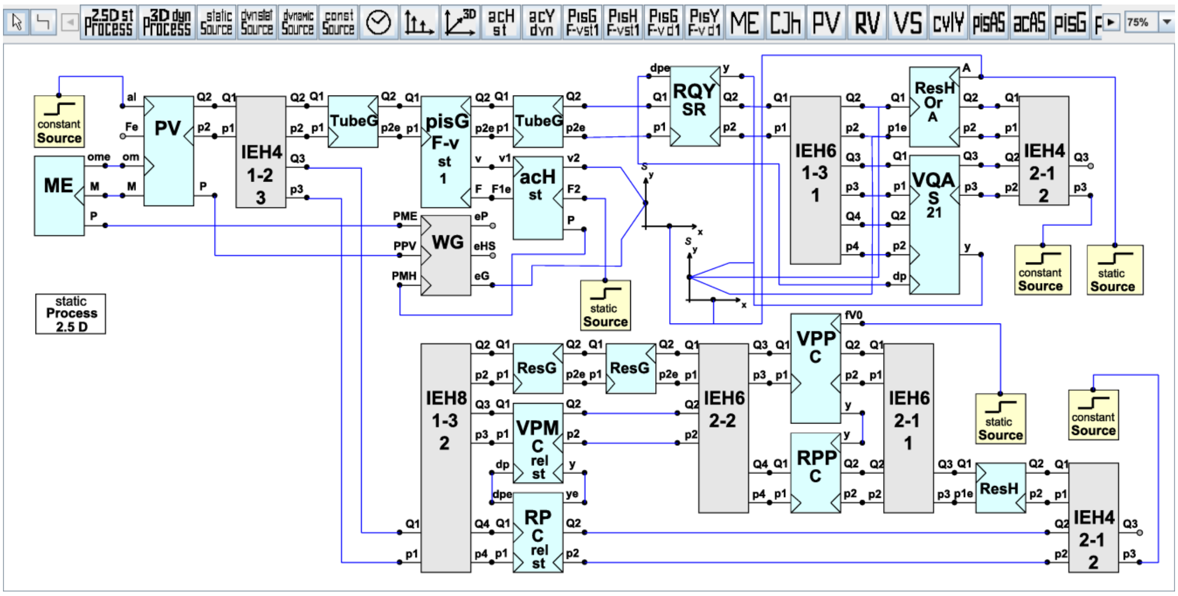

Fig.1. Simulation task of a hydraulic drive with two-directional flow regulating valve in cylinder outlet for steady state conditions

Multi-pole models: **ME-** electric motor, **PV -** hydraulic pump, **TubeG** – tubes, **pisG_F-v_st1** – piston, **acH_st** – actuator, **RQY_SR** – pressure compensator throttle opening consisting of a number of separate slots of triangular form, **ResH_Or_A –** regulating throttle orifice, **VQA_S_21** – pressure compensator spool with spring, **VPM_C_rel_st** – main poppet valve with spring, **RP_C_rel_st** – main poppet valve slot, **VPP_C** – pilot poppet valve with spring, **RPP_C** – pilot poppet valve slot, **ResG, ResH** – hydraulic resistors, **IEH –** interface elements, **WG** – efficiency coefficient calculator [4, 5, 6].

Inputs: load force F2, regulating throttle orifice area *A*, constant position angle *al* of the pump regulating swash plate, spring preliminary deformation fV0, outlet pressures *p*3.

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

Simulation manager*:* static Process 2.5D.

Parameters of the system have been described in detail in Part 1 of the paper. Parameter values listed below are chosen on the basis of the constructive aspects and have been adjusted step by step as a result of simulations in order to ensure the fluid power system well functioning.

VPP_C: d1=0.0048 m, d2=0.005 m, β=15 deg, G=8e11 N/m, ds=0.001 m, Ds=0.008 m, n=8, m=0.02 kg, Ff0=0, kfr=0, h=0.

RPP_C: μ= 0.8, d1= 0.0048 m, d2= 0.005 m, β= 15 deg.

VPM_C_rel_st: d1=0.021 m, d2=0.022 m, β=45 deg, G=8e11 N/m, ds=0.0009 m, Ds=0.016 m, n=8, fV0=0.0007 m.

RP_C_rel_st: μ=0.8, β=45 deg, d1=0.021 m, d2=0.022 m.

pisG_F-v_st1: dpi= 0.10 m, dr1= 0 m, dr2= 0.056 m, Ffpi0= 100 N, Ffr20= 50 N, h= 100.

acH_st: Ffr0= 10 N, h= 3e4 Ns/m.

TubeG: d= 0.019 m, l= 2 m.

ME: rotation frequency om0= 154.46 rad/s.

PV: working volume V= 6.935e-6 m^3/rad.

RQY_SR: number of slots n= 3.

VQA_S_21: parameters are discussed and values are described in [5].

The first **ResG** in relief valve: l= 0.005 m, d= 0.001 m, μ= 0.7.

The second **ResG** in relief valve: l= 0.005 m, d= 0.0015 m, μ= 0.7.

ResH: l= 0.03 m, d= 0.0012 m, μ= 0.7.

Results of simulation of steady state conditions depending on the load force value from -5e4 N to 10.5e4 N for three different values of the regulating orifice area A =(12, 8, 4) e-6 m^2 are shown in Fig.2 and Fig.3.

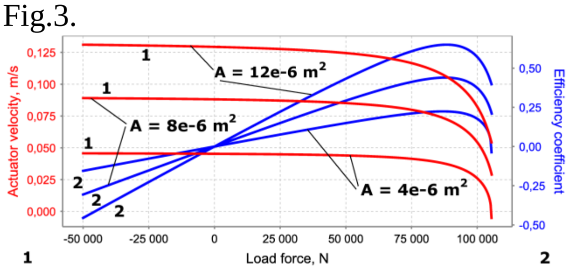

Fig.2. Graphs of simulations

In Fig.2 actuator velocity (graphs 1) decreases slightly until F < 7.5e4 N, after that it drops down fast. Increasing the load force causes efficiency coefficient (graphs 2) to rise. Efficiency coefficient reaches maximum at load force ~9e4 N, after it drops down fast.

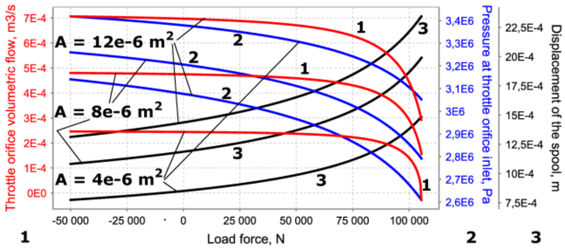

Fig.3. Graphs of simulations

In Fig. 3 throttle orifice volumetric flow (graphs 1) determines actuator velocity behavior (see graphs 1 in Fig. 2). Increasing the load force causes pressure at throttle orifice inlet (graphs 2) decrease and flow regulating valve spool displacement (graphs 3) increase.

Results of simulation of steady state conditions depending on the throttle orifice area from 4e-6 m^2 to 12e-6 m^2 for two different values of load force F = (1.0, 10.5) e4 N are shown in Fig.4 and Fig.5.

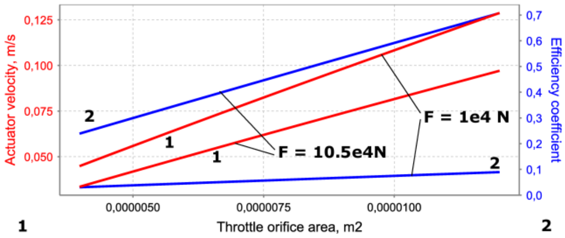

Fig.4. Graphs of simulations

In Fig.4 both actuator velocity (graphs 1) and efficiency coefficient (graphs 2) increase linearly if throttle orifice area increases.

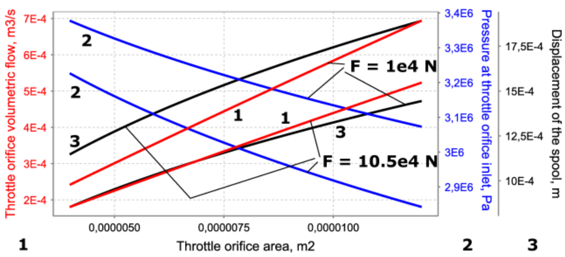

Fig.5. Graphs of simulations

In Fig.5 throttle orifice volumetric flow (graphs 1), and flow regulating valve spool displacement (graphs 3) increase almost linearly. Throttle orifice inlet pressure (graphs 2) decreases almost linearly.

5. SIMULATION OF DYNAMICS

Simulation task of a hydraulic drive with two-directional flow regulating valve in outlet for dynamics is shown in Fig.4.

Additional multi-pole models to models of steady state conditions: **CJh** – clutch, **TubeY**, **TubeG –** cylinder inlet and outlet tubes, **pisY –** piston, **cylY** – cylinder, **veZ1**, **veZ2 –** volume elasticities of cylinder chambers, **acY**– actuator [4, 5, 6].

Inputs: load force Fac2, regulating orifice area A, constant position angle al of the pump regulating swash plate, constant outlet pressures p3.

Outputs: actuator velocity v2, outlet volumetric flows Q3, cylinder position xfi.

Simulation manager*:* dynamic Process3D.

The following parameter values are chosen and have been adjusted as a result of dynamic simulations in addition to those used in simulations of steady state conditions.

VPM_C: d1=0.021 m, d2=0.022 m, β=45 deg, G=8e11 N/m, ds=0.0009 m, Ds=0.016 m, n=8, fV0=0.0007 m, m=0.06 kg, Ff0=0, kfr=0, h=0.

RP_C_rel_dyn: μ=0.8, β=45 deg, d1=0.021 m, d2=0.022 m.

pisY: dpi=0.10 m, dr1=0 m, dr2=0.056 m, Ffpi0=100 N, Ffr20=50 N, h=100 Ns/m, er=1e-10 m/N.

cyl1: dpi=0.10 m, dr1=0 m, dr2=0.056 m, efi=1e-9 m/N, ebu=1e-9 m/N, Fffi=20 N, h=5e5 Ns/m, m=20 kg.

veZ1, **veZ2:** length of piston stroke lch=0.4 m.

acY: Ffr0=10 N, h=3e4 Ns/m, m=20 kg.

TubeG, TubeY: d=0.019 m, l=2 m.

The first **ResH** in relief valve: l=0.005 m, d=0.0004 m, μ=0.7.

The first **ResG_Ch**: l=0.02 m, d=0.0012 m, μ=0.8.

The second **ResG_Ch**: l=0.01 m, d=0.0015 m, μ=0.8.

ResH in flow regulating valve: l=0.03 m, d=0.0012 m, μ=0.7.

Results of simulation of dynamic responses caused by applying the hydraulic drive step load force F2 = 5e3 N from mean value 0 (step time 0.01 s) as input disturbance are shown in Fig.7 and Fig.8. Throttle orifice area has constant value 5e-6 m^2.

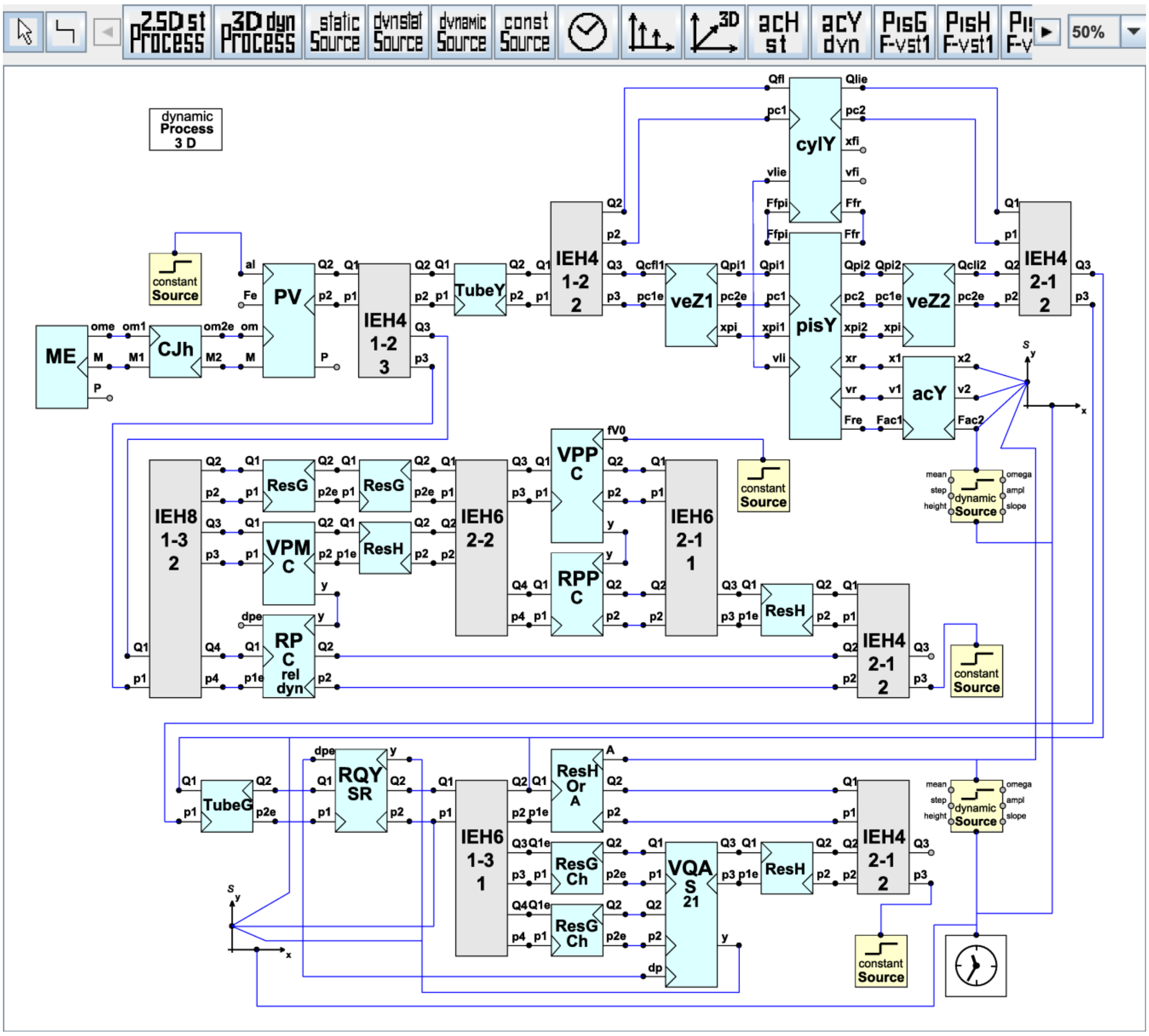

Fig.6. Simulation task of dynamics of a hydraulic drive

In Fig.7 input load force step change (graph 3) initially causes actuator velocity (graph 2) to drop. After the load force takes a new level, actuator velocity stabilizes with damped oscillations. Actuator moves almost linearly (graph 1).

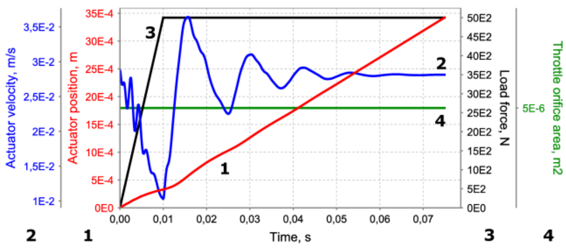

Fig.7. Graphs of actuator

In Fig.8 decreasing actuator velocity (graph 2 in Fig.7) causes throttle orifice volumetric flow (graph 1) and pressure at throttle orifice inlet (graph 2) to decrease with damped oscillations. Flow regulating valve spool moves to the new position (graph 3).

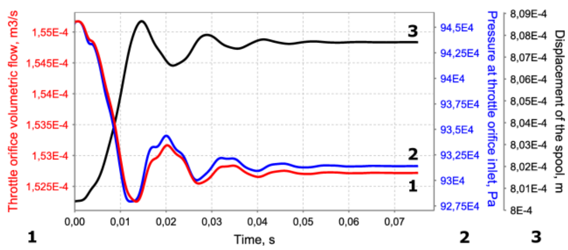

Fig.8. Graphs of flow regulating valve

Results of simulation of dynamic responses caused by applying the hydraulic drive step change A=1e-6 m^2 of throttle orifice area from mean value 5e-6 m^2 (step time 0.01 s) as input disturbance are shown in Fig.9 and Fig.10. Load force is of constant value 0. In Fig.9 input throttle orifice area step change (graph 4) initially causes actuator velocity (graph 2) to oscillate, to increase to the new level and to stabilize with damped oscillations. Actuator moves linearly (graph 1).

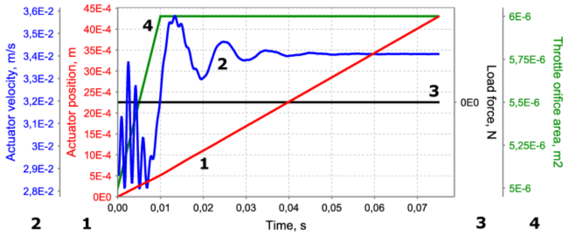

Fig.9. Graphs of actuator

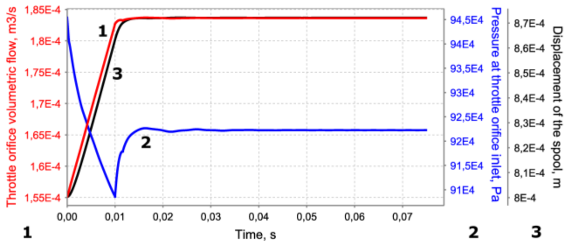

Fig.10. Graphs of flow regulating valve

In Fig.10 input disturbance (graph 4 in Fig.9) causes throttle orifice volumetric flow (graph 1) and flow regulating valve spool (graph 3) to follow the input. Opening throttle orifice causes pressure at throttle orifice inlet (graph 2) to drop.

6. CONCLUSION

In the paper simulation of a hydraulic drive with two-directional flow regulating valve in cylinder outlet has been considered.

As a result of number of step by step simulations of steady state conditions and dynamic transient responses a set of optimal parameters for the hydraulic drive was proposed.

Using methodology described here enables to try out different configurations and find optimal parameters in design and development of various fluid power systems.

7. ACKNOWLEDGEMENTS

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