MULTI-POLE MODELING AND INTELLIGENT SIMULATION OF CONTROL VALVES OF FLUID POWER SYSTEMS (PART 1)

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Abstract: Composing of multi-pole models and simulation of pressure control valves and flow regulating valves used in fluid power systems is considered in the paper. Part 1 of the paper discusses methodology of modeling and simulation. Multi-pole mathematical models of pressure control valves are presented. An intelligent simulation environment CoCoViLa supporting declarative programming in a high-level language and automatic program synthesis is used as a tool. Simulation examples of pressure control valves are presented and discussed. In Part 2 multi-pole mathematical models of flow regulating valves are described. Simulation examples of flow regulating valves are presented and discussed. Key words: multi-pole model, pressure control valve, intelligent programming environment, simulation.

1. INTRODUCTION

When composing models of complex fluid power systems usually models of components of different levels are used. Components of the lowest level are hydraulic resistors, tubes, hydraulic interface elements, main valves, pilot valves, etc. Hydraulic control valves of different types are used as components of the medium level. In this way models of complex systems can be built up hierarchically. Hydraulic control valves \(^{[1-5]}\) are quite complex devices containing low level components, including internal feedbacks and having possibilities for adjusting.

Modeling and simulation tools in existence, their characteristics and disadvantages have been discussed in \(^{[6-8]}\). Using mainly two-pole models for describing hydraulic control valves \(^{[3-5]}\) is not adequate, as components of such systems exert feedback actions. Obtained large equation systems usually need checking and correcting to guarantee solvability.

In the current paper an approach is proposed, which is based on using multi-pole models with different oriented causalities \(^{[4]}\). A special technique is used that allows avoid solving large equation systems during simulations \(^{[7]}\). Therefore, multi-pole models of large systems do not need considerable simplification.

2. MULTI-POLE MODELS

In general a multi-pole model represents mathematical relations between several input and output values (poles). 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.

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 [9]. The environment supports visual and model-based software development and uses structural synthesis of programs [10] for translating declarative specifications of simulation problems into executable code. 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. PRESSURE CONTROL VALVES

Here we consider pressure control valves that are types of valves used to limit (safety valve) or control (relief valve) pressure in a fluid power system. Two types of pressure control valves: direct operated and pilot operated are under consideration.

4.1 Direct Operated Pressure Control Valve

Direct operated pressure control valve under consideration is shown in Fig.1.

Fig. 1. Direct operated pressure control valve (Mannesmann Rexroth)

The valve consists of housing 1, sleeve 2, spring 3, advance mechanism 4, poppet with cushioning spool 5, hardened seat 6, spool sleeve 7 and spring retainer of a special shape 8 (used to compensate fluid jet force).

If the valve is used as safety valve, pressure is considered as input. If the valve is used as relief valve, volumetric flow is considered as input.

Simulation task for dynamics on a direct operated safety valve model is shown in Fig.2.

Fig.2. Simulation task of a direct operated safety valve dynamics


Inputs: pressure p1, spring preliminary compressibility fV0, outlet pressures p2, p3.

Outputs: volumetric flows Q1 and Q2.

Simulation manager: dynamic Process 3D.

4.1.1 Mathematical Models

Poppet with spool and spring VPCplsaf

Inputs: pressures p1, p2, p3, spring preliminary compressibility fV0, pressure drop in flow-through slot of poppet valve dp.

Outputs: volumetric flows Q1, Q2, displacement of poppet valve y.

Force acting to spring retainer:

\[ Fr = \pi \times \frac{(d_r^2 - d_1^2)}{4} \times \frac{(p_2 + p_3)}{2}, \]

where

- \( d_r \) diameter of spring retainer,
- \( d_1 \) diameter of spool sleeve,
- \( p_2 \) pressure after poppet valve,
- \( p_3 \) outlet pressure.

Lifting force to the poppet valve:

\[ F_1 = \pi \times \frac{d_1^2}{4} \times \frac{p_1}{4} + \pi \times \frac{(d_2^2 - d_1^2)}{(p_1 - p_2)} \times \sin \left( \frac{\beta \times \pi}{180} \right) + Fr, \]

where

- \( d_1 \) diameter of spool sleeve,
- \( d_2 \) outer diameter of poppet seat,
- \( p_1 \) input pressure,
p2 pressure after poppet valve,
β half of cone angle,
Fr force acting to spring retainer.

Opposite force to the poppet valve:
\[ F_2 = \pi * dr^2 * p_3 / 4, \]
where
dr diameter of spring retainer,
p3 outlet pressure.

Stiffness of the spring:
\[ c = G * ds^4 / (Ds^3 * n * 8), \]
where
G shear modulus,
ds diameter of spring wire,
Ds diameter of spring,
n number of turns of the spring.

Displacement of the poppet valve:
\[ y = (F_1 - F_2) / c - fV_0, \]
where
F1 lifting force to the poppet valve,
F2 opposite force to the poppet valve,
fV0 spring preliminary compressibility.

Volumetric flows:
\[ Q_1 = 0, \quad Q_2 = 0. \]

Difference of valve velocity used in Runge-Kutta method for integration for dynamics:
\[ dv = (\Delta t / m)*(F_1 - F_2 - (y + fV_0)*c - (Ff_0 + kfr*(p_1 + p_2)/2)*sign(v,0.001)-hv*v), \]
where
Δt time step,
m mass,
Ff0 constant part of friction force,
kfr coefficient of friction force,
v velocity of valve,
hv damping coefficient.

Difference of valve displacement:
\[ dy = \Delta t * v. \]

Effective area of valve:
\[ A = \pi * d_1^2/4. \]

Volumetric flows:
\[ Q_1 = A * v, \quad Q_2 = Q_1. \]

Poppet valve slot RPCsaf

Inputs: pressures p1, p2, displacement of poppet valve y.

Outputs: volumetric flows Q1, Q2, pressure drop in poppet valve slot dp.

Area of the poppet valve slot:
\[ Ad = \mu * \pi * (d_1 + d_2)/2 * y * \sin(\beta * \pi / 180), \]
where
\[ \mu \] discharge coefficient,
d1 diameter of spool sleeve,
d2 outer diameter of poppet seat,
y displacement of the poppet valve,
\[ \beta \] half of cone angle.

Pressure drop in poppet valve slot:
\[ dp = p_1 - p_2. \]

Volumetric flows:
\[ Q_1 = Ad * (2 * \abs(dp) / \rho)^{1/2}, \quad Q_2 = Q_1, \]
where
\[ \rho \] fluid density.

4.1.2 Simulations

The following parameter values are used in the simulations under consideration.

Basic parameters for the fluid HLP46:
kinematic viscosity at temperature 40°C \( \nu_{40} = 46e-6 \text{ m}^2/\text{s} \), density at temperature 15°C \( \rho_{15} = 875 \text{ kg/m}^3 \), basic compressibility factor of fluid at temperature 20°C \( \beta_{F20} = 1/18.4e8 \text{ m}^2/\text{N} \), relative content of undissolvable air in fluid \( \text{vol} = 0.02 \) and temperature \( \theta = 40^\circ \text{C} \).

For VPCp1saf:
d1=0.0048 m, d2=0.005 m,
dr=0.014 m, \beta=15 deg,
ds=0.0025 m, Ds=0.016 m, n=6,
G=8e11 N/m, m=0.04 kg,
kfr=0 N/Pa, Ff0=0 N, hv=0 Ns/m.

For RPCsaf:
d1=0.0048 m, d2=0.005 m,
\mu=0.8, \beta=15 deg.

For ResHrad:
dr=0.014 m, d1= 0.0048 m.

For ResGCh:
diameter d=0.0005 m, length l=0.02 m.

Results of direct operated safety valve statics simulation are shown in Fig.3.

Fig.3. Direct operated safety valve statics

Graphs of volumetric flow through the valve depending on the input pressure for 5 different values of spring preliminary compressibility fV0 from 0.0005 m to 0.0025 m are presented. The valve opens correspondingly at pressures 5e6 Pa to 22e6 Pa depending on fV0. The value of
volumetric flow is defined by restricted displacement value $1 \times 10^{-3}$ m of the poppet valve.

Results of direct operated safety valve dynamics simulation are shown in Fig.4.

Fig.4. Direct operated safety valve dynamics

A step disturbance $4 \times 10^6$ Pa of pressure $p_1$ at the left port during 0.001 s is applied as input (graph 2). The volumetric flow (1) follows the change of the input pressure, oscillations damp in 0.008 s.

Results of direct operated relief valve statics simulation are shown in Fig.5.

Fig.5. Direct operated relief valve statics

Graphs of output pressure (1) and displacement of the valve (2) depending on the input volumetric flow for three different values of spring preliminary compressibility $fV0$ from 0.0005 m to 0.003 m are presented. Some dependence of output pressure from input volumetric flow can be observed.

Results of direct operated relief valve dynamics simulation are shown in Fig.6. A step disturbance $1 \times 10^{-4}$ m$^3$/s of volumetric flow during 0.001 s at the left port is applied as input (2). Displacement of the valve (3) almost follows change of the input volumetric flow. Damped oscillating output pressure (1) increases from initial to new higher level.

4.2 Pilot Operated Pressure Control Valve

Pilot operated pressure control valve under consideration is shown in Fig.7.

Fig.7. Pilot operated pressure control valve (Mannesmann Rexroth)

The valve contains pilot poppet valve 1, pilot valve spring 2, pilot control line resistors 3 and 4, main valve 5, main valve spring 6, main valve cushioning resistor 7 and advance mechanism 8.

If the valve is used as safety valve, pressure is considered as input. If the valve is used as relief valve, volumetric flow is considered as input.

Simulation task for dynamics on a pilot operated safety valve model is shown in Fig.8.

Fig.8. Simulation task of a pilot operated safety valve dynamics
**Multi-pole models:** VPPC – pilot poppet valve with spring, RPPC – pilot poppet valve slot, VPMC – main poppet valve with spring, RPMC – main poppet valve slot, ResGRor – pilot control line resistors, ResHRor – main poppet valve cushioning resistor, ResHCh – outlet resistor, IEH8-1-3-1, IEH6-2-2, IEH6-2-1, IEH4-2-1-2 – interface elements.

**Inputs:** pressure p1, spring preliminary compressibility fV0, outlet pressure p3.

**Outputs:** volumetric flows Q1 and Q2.

**Simulation manager:** dynamic Process 3D.

The following parameter values are used in the simulations.

- For VPPC: d1=0.0048 m, d2=0.005 m, dr=0.014 m, β=15 deg, ds=0.015 m, Ds=0.008 m, n=8, G=8e11 N/m, m=0.02 kg, kfr=2e-9 N/Pa, Ff0=0 N, hv=50 Ns/m.

- For RPPC: d1=0.0048 m, d2=0.005 m, μ=0.8, β=15 deg.

- For VPMC: d1=0.021 m, d2=0.022 m, β=45 deg, ds=0.0008 m, Ds=0.016 m, n=8, G=8e11 N/m, m=0.05 kg, kfr=2e-9 N/Pa, Ff0=2 N, hv=5 Ns/m.

- For RPMC: d1=0.021 m, d2=0.022 m, μ=0.8, β=45 deg.

- For ResGRor: d=0.001 m and 0.002 m, l=0.005 m.

- For ResHRor: d=0.001 m, l=0.005 m.

- For ResHCh: d=0.002 m, l=0.03 m.

Results of pilot operated safety valve statics simulation are shown in Fig.9.

![Fig.9. Pilot operated safety valve statics](image)

Graphs of main volumetric flow (1), pilot volumetric flow (2) and pilot pressure (3) depending on the input pressure for two different values of preliminary compressibility fV0 = 0.0016 m and 0.0018 m of the pilot valve spring are presented. If fV0 = 0.0016 m pilot valve opens rapidly at pressure 11e6 Pa then closes a bit and opens more later. If fV0 = 0.0018 m pilot valve opens rapidly at pressure 12.3e6 Pa then closes and reopens later.

The main valve reacts slightly to opening of the pilot valve and opens rapidly correspondingly at pressures 12.2e6 Pa and 13.6e6 Pa depending on the value of fV0.

Results of pilot operated safety valve dynamics simulation are shown in Fig.10.

![Fig.10. Pilot operated safety valve dynamics](image)

A step disturbance 3e6 Pa of pressure p1 during 0.1 s at the left port is applied as input (1). Pilot valve (3) begins to open at input pressure 12.3e6 Pa. At input pressure 13.4e6 Pa main valve (4) begins to open. At the same time output volumetric flow (2) rapidly increases to maximum. Main valve shift (4) to maximum 20e-4 m causes additional flow through pilot valve and its additional shift (3). When the main valve stops at the maximum displacement the additional flow through pilot valve disappears and pilot valve partly closes.

Results of pilot operated relief valve statics simulation are shown in Fig.11.

![Fig.11. Pilot operated relief valve statics](image)

Graphs of output pressure (1) and displacement of the main valve (2) depending on the input volumetric flow for three values of preliminary compressibility fV0 from 0.0006 m to 0.0014 m of pilot valve spring are presented. Slight dependence of output pressure on input
volumetric flow can be observed. Results of pilot operated relief valve dynamics simulation are shown in Fig.12.

Fig.12. Pilot operated relief valve dynamics

A step disturbance 1e-4 m³/s of volumetric flow during 0.001 s at the left port is applied as input (2). Displacement of the main valve (3) follows the input volumetric flow with a little delay.

Both the pilot valve (4) and the output pressure (1) oscillate and reach the new level during 0.003 s.

5. CONCLUSION

In the paper modeling and simulation of hydraulic pressure valves of fluid power systems were considered. Multi-pole models are used that enable adequately describe physical processes in hydraulic systems. Both direct actions and feedbacks can be expressed in component models.

Simulations are performed using an intelligent programming environment CoCoViLa with feature of automatic synthesis of programs from the knowledge available in visual model.

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7. REFERENCES

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