MODELLING AND SIMULATION OF PROPORTIONAL MOBILE VALVES

Timo KÄPPI and Asko ELLMAN

Institute of Hydraulics and Automation
Tampere University of Technology
P.O. Box 589, FIN-33101 Tampere, Finland
(E-mail: tkappi@butler.cc.tut.fi)

ABSTRACT

A modern mobile valve is a complex entity which consist of several integrated valve functions such as pressure compensation, pump flow sharing (anti-saturation), pressure relief and anticavitation. Main spool is either electrically or hydraulically pilot operated with load sensing ability so the valve can be used with LS-pumps.

In this paper the methods of modelling multifunctional mobile valves are presented. The proposed modelling technique is numerically efficient since small internal oil volumes of the valve can be ignored and mathematical stiffness of the model is significantly decreased. The usability of the model is taken into account by means of easy parameterization. The model can be identified using data available in the manufacturers' catalogues.

KEY WORDS

Modelling, simulation, mobile valve, pressure compensation, flow sharing

NOMENCLATURE

A_C, A_S :	equivalent orifice flow cross section area	$Q_{1,2,3}$	flow parameters
A_{spool} :	spool end area	Q :	flow
b :	viscous friction coefficient	R_{cs} :	relative opening of the equivalent orifice
C_q :	discharge coefficient	R_s :	relative opening of the main spool
D :	compensator spool diameter	T :	auxiliary variable
m :	spool mass	x :	compensator spool position
F_f :	friction force	α :	pressure ratio
F_p :	pressure force	Δp :	pressure difference across the valve
F_{pre} :	spring pre-setting force	ρ :	fluid density
F_s :	mechanical spring force	τ :	time constant
F_μ :	friction force		
k_{hyd} :	hydraulic spring constant		
k_{mech}	mechanic spring constant		
$p_{1,2,3}$	pressure parameters		
p_C :	compensator pressure		
p_{LS} :	individual spool section load pressure		
$p_{LS MAX}$:	maximum (system) load pressure		
p_s :	supply pressure		
p_τ :	pressure level		

INTRODUCTION

Proportional directional control valves are the standard components employed in mobile hydraulics. They consist of various spool sections and several valve functions are typically integrated into the valve block.

However, simulation models presented usually concern with greatly simplified model of the valve. In order to be able to model and simulate the performance of mobile

machines such as forest machines or excavators, an accurate model of the valve becomes essential.

Figure 1. VOAC K170LS directional control valve supplied with two solenoid controlled spool sections [6]

Fig. 2 shows typical spool section of mobile valve. The spool is supplied with load drop check valve and pressure compensator. Pressure relief valves and anti-cavitation check valves are included in the motor ports A and B. In case of several spool sections the biggest load signal is gathered utilizing logic check valves.

Figure 2. Example of spool section

The previous modelling procedures of flow and pressure valves are presented in references [3] and [4]. Mobile valves typically have small internal oil volumes between load drop check valve and compensator and between compensator and main spool as well. The numerical difficulties occurring during simulation due to these small volumes are clarified in reference [2]. The major advantage of the method stated in this paper is how avoid the use of small volumes in the model by merging consecutive components into one equivalent orifice.

MODELLING OF MOBILE VALVES

This paper basically considers the modelling of pressure compensated spool with or without flow sharing (anti-saturation) function. The model is semi-empirical since it can be parameterized by measured or catalogue based values instead of analytical ones. The semi-empirical model is not dependent on valve's internal structure.

Pressure compensation

The function of pressure compensator is to maintain constant pressure across the main valve spool and therefore preserve constant actuator velocity.

Figure 3. Schematic picture of pressure compensated spool

According to fig. (3) the equation for the flow in valve port P can be defined as

$$\begin{aligned} Q_P &= C_q A_C \sqrt{\frac{2(p_s - p_c)}{\rho}} \\ &= R_s C_q A_S \sqrt{\frac{2(p_c - p_{LS})}{\rho}} \end{aligned} \quad (1)$$

where R_s is relative opening of the main spool (-1...1) that is a function of spool displacement and shape, spool laps, switching time and solenoid or (hydraulic) pilot circuit dynamics.

Auxiliary variable T can be set as

$$T = \left(\frac{1}{R_s C_q A_S} \right)^2 \frac{\rho}{2} \quad (2)$$

The pressure p_c between compensator and main spool can be solved from the eq. (1).

$$p_C = \left(\frac{Q}{R_S C_q A_S} \right)^2 \frac{\rho}{2} + p_{LS} \quad (3)$$

$$= T Q^2 + p_{LS}$$

The characteristics curve of pressure compensated valve can be divided into three parts shown in fig. 4. When the pressure difference across the valve is below the corresponding compensator spring pre-set, the valve performs as a restrictor. When the critical pressure difference is exceeded the compensator starts to operate. It keeps the flow nearly constant (the decreasing gradient is due to compensator flow forces) until the maximum operation pressure is reached. At this point the compensator spool movement is saturated due to the mechanical restrains of valve construction. However, the valve does not completely close but flow-through is allowed. After this point the valve performs as a restrictor again. In practise, compensator saturation indicates poor system design thus component selection should be done carefully in order to avoid discussed phenomenon.

Figure 4. Valve parametrization

Mobile valve spools are typically delivered based on customers' specifications. Therefore the spool is manufactured in a way that each flow path (P->A, P->B, A->T, B->T) gives a specified flow with certain pressure drop across the spool.

The valve model (flow paths P->A , P->B) can be identified based on the characteristics curve (fig. 4) using three Qp-points. The values depend on spring pre-set, spool specification and linearity of the valve performance.

The maximum $C_q A$ -value for the compensator and main spool combination is derived from pressure p_1 corresponding to spring pre-set and nominal flow Q_1 across the main spool with full opening.

$$\begin{cases} C_q A_{\max}^{PA} = \frac{Q_{1PA}}{\sqrt{\frac{2 p_{1PA}}{\rho}}} \\ C_q A_{\max}^{PB} = \frac{Q_{1PB}}{\sqrt{\frac{2 p_{1PB}}{\rho}}} \end{cases} \quad (4)$$

The second Qp-point (p_2, Q_2) is taken from the operating area of pressure compensator.

The third Qp-point (p_3, Q_3) indicates the pressure where the compensator saturation begins.

The other flow paths (A->T , B->T) can be set similarly when one Qp-point is known from the characteristics curve.

$$\begin{cases} C_q A_{\max}^{AT} = \frac{Q_{AT}}{\sqrt{\frac{2 p_{AT}}{\rho}}} \\ C_q A_{\max}^{BT} = \frac{Q_{BT}}{\sqrt{\frac{2 p_{BT}}{\rho}}} \end{cases} \quad (5)$$

The force equation for the compensator spool is expressed as

$$\begin{aligned} m\ddot{x} &= F_s + F_p + F_f + F_{pre} + F_{\mu} \\ &= -k_{mech} x - A_{spool} (p_C - p_{LS}) - \\ &\quad k_{hyd} x (p_S - p_C) + A_{spool} p_{pre} - b\dot{x} \end{aligned} \quad (6)$$

Assuming the mass of spool $m \approx 0$ and linear damping force, the originally second order system can be reduced into first order system. This also makes it considerably easier to parameterise the dynamics of the valve.

$$\dot{x} = \frac{1}{b} \left[-x [k_{mech} + k_{hyd} (p_S - p_C)] + A_{spool} [p_{pre} - (p_C - p_{LS})] \right] \quad (7)$$

The $C_q A$ -value for the spool valve can be defined as

$$C_q A_C = C_q \pi D x \quad (8)$$

Consequently the compensator spool position is

$$x = \frac{C_q A_C}{C_q \pi D} \quad (9)$$

Substituting eq.(9) into eq.(7) leads into following differential equation describing the opening of compesator orifice

$$\begin{cases} C_q \dot{A}_c = \frac{1}{b} [-C_q A_c [k_{mech} + k_{hyd}(p_S - p_C)] + \\ C_q \pi D A_{spool} [(p_{pre} - (p_C - p_{LS}))]] , \Delta p < p_3 \\ C_q \dot{A}_c = 0 , \Delta p > p_3 \end{cases} \quad (10)$$

Viscous friction parameter b defines the compensator dynamics. It can be derived from the motion equation utilizing Laplace transformation (derivation not presented here)

$$b = \tau (k_{mech} + k_{hyd} p_\tau) \quad (11)$$

where τ is the compensator time constant and p_τ is the corresponding pressure level where the time constant is defined.

During simulation eq.(10) is numerically integrated and the relative opening R_{cs} of the equivalent orifice (0...1) corresponding to compensator and main spool can be expressed as

$$R_{cs} = |R_s| \left(\frac{C_q A_c}{C_q A_{max}} \right) \quad (12)$$

The flow over the valve can be calculated using orifice model presented in reference [1].

$$\begin{cases} Q_P = f(R_{cs}, C_q A_{max}) \text{ when } p_S - p_{LS} > 0 \\ Q_P = 0 \text{ when } p_S - p_{LS} < 0 \end{cases} \quad (13)$$

Load drop check valve is typically installed in spool section before compensator (fig. 3). It ensures that the flow through the port P backwards is prevented if load pressure exceeds the supply pressure. This may occur in LS-systems if the valve is opened fast and pump response is relatively slow. The second definition in eq.(13) corresponds an ideal load drop check valve where the dynamics of the check valve are neglected. This simplification is justified because check valves operate very fast thus their dynamics has no significant influence on the system behaviour. It also considerably decreases the mathematical stiffness of the simulation model and reduces the computational effort.

The stationary performance of the valve can be solved from the motion equation (10) when velocity dependent factor $C_q \dot{A}_c = 0$.

Solved from the eq.(2) and (10)

$$C_q A_c = \frac{C_q \pi D A_{spool} (p_{pre} - T Q^2)}{k_{mech} + k_{hyd} [(p_S - p_{LS}) - T Q^2]} \quad (14)$$

Based on the given parameters (fig. (4)) the following system of equations can be set

$$\begin{cases} Q_2 = \frac{C_q \pi D A_{spool} (p_1 - T Q_2^2)}{k_{mech} + k_{hyd} (p_2 - T Q_2^2)} \sqrt{\frac{2}{\rho} (p_2 - T Q_2^2)} \\ Q_3 = \frac{C_q \pi D A_{spool} (p_1 - T Q_3^2)}{k_{mech} + k_{hyd} (p_3 - T Q_3^2)} \sqrt{\frac{2}{\rho} (p_3 - T Q_3^2)} \end{cases} \quad (15)$$

The mechanic and hydraulic spring constants k_{mech} and k_{hyd} , respectively, can be calculated using symbolic mathematics software such as Mathematica or Maple.

Pressure compensation with anti-saturation feature

In case when the compensator is provided with anti-saturation feature there are extra pistons added to the compensator construction (fig. 5). The supply pressure as well as the maximum load pressure of the system affects on the compensator spool displacement. When the pump output flow saturates due to simultaneous use of actuators, the pump is disabled to keep the desired Δp . At that time individual compensators tend to close and all motions are slowed down independently on individual actuator loads. Otherwise the lightest operations would go on and actuators of heaviest loads would either slow down or totally stop.

Figure 5. Schematic picture of pressure compensated spool with anti-saturation feature

Based on fig. (5) eq. (10) can be improved into following form

$$C_q \dot{A}_c = \frac{1}{b} [-C_q A_c k_{hyd} (p_S - p_C) + C_q \pi D A_{spool} (\alpha p_S + p_{LS} - p_C - \alpha p_{LS MAX})] \quad (16)$$

The value of α depends on the following ratio

$$\alpha = \frac{P_C - P_{LS}}{P_S - P_{LSMAX}} \quad (17)$$

Pressure relief valve function

The model of pressure relief valve is similar as expressed in reference [5]. However, similarly as in case of compensator spool, the dynamic model is reduced down to first order system in order to make the model parametrization easier. The model equals typical well-damped single stage pressure relief valve.

$$C_q \dot{A}_{prv} = \frac{1}{b} \left[-C_q A_{prv} (k_{mech} + k_{hyd} P_{IN}) + \frac{(P_{IN} - P_{REF} - P_{OUT})}{\sqrt{\frac{2}{\rho}}} \right] \quad (18)$$

Anti-cavitation valve function

The check valve model considers cracking pressure P_{CRACK} and one point from Qp-curve (p_1, Q_1). The switching time of the valve is assumed to be constant.

$$\begin{cases} Q_{OUT} = 0, & \Delta p < P_{CRACK} \\ Q_{OUT} = K_1 \Delta p, & P_{CRACK} < \Delta p < p_1 \\ Q_{OUT} = K_2 \Delta p, & \Delta p \geq p_1 \end{cases} \quad (19)$$

SIMULATION EXAMPLES

The models presented in the previous chapter are clarified by two examples. In the first one the pressure compensated valve is described by using parameters in table 1.

Table 1. Valve parameters

p_1	6 bar
p_2	100 bar
p_3	300 bar
Q_1	180 l/min
Q_2	175 l/min
Q_3	170 l/min
τ	50 ms
p_t	100 bar

Figure 6. Simulation results of the valve using 100 %, 50 % and 20 % openings of main spool

Three simulations are run using different openings of the main spool. The pressure across the valve is increased slowly in order to minimize the effect of valve dynamics. The flow responses as a function of pressure difference can be seen in fig. (6). They express the static behaviour of the valve.

In the second example a typical load sensing system is studied. The system consists of LS-pump, mobile valve with three spool sections and three hydraulic cylinders with loads of different magnitude. The system parameters are listed in table 2.

Table 2. System parameters

V_{pump}	120 cc/rev , 2000 rev/min
Cylinders	100/50-1500
Loads	1000 , 2000 , 3000 kg

During the simulation run the cylinders are actuated by intervals of one second. The simulated output flow from the LS-pump is shown in fig.(7). The pump flow saturation can be seen when the second cylinder is actuated.

Figure 7. Pump flow with respect to time

The available flow is equally shared between actuators and cylinder velocities equals.

Figure 8. Simulated actuator velocities (flow sharing)

For the sake of comparison in fig. (9) actuator velocities are shown in case of not anti-saturation compensated spool.

Figure 9. Simulated actuator velocities (no flow sharing)

The lightest loaded actuator (1000 kg) will go on while the second lightest (2000 kg) one gets the rest of the available pump supply. The heaviest load (3000 kg) can not be moved. Load drop check valve holds the load.

Because of simultaneous motions of mobile machine, pump flow saturation is a typical phenomenon during work cycle. The discussed model is unavoidable in order to simulate those machine systems accurately.

SUMMARY

The modelling of multifunctional mobile valve is discussed. The theoretical background is presented and test bench simulations are shown for the model verification as well.

Easy parameterization based on component manufacturers' catalogues hugely increases the usability

of the model. The presented model has been successfully implemented as a part of a large simulation model of certain hydraulically driven mobile machine. It has turned out to be essential for achieving exact simulation results of such machines.

REFERENCES

- [1] Ellman, A. & Piché, R. : "A Modified Orifice Flow Formula for Numerical Simulation of Fluid Power Systems ". FPST-Vol. 3. Fluid Power Systems and Technology: Collected Papers ASME 1996.
- [2] Ellman, A. & Piché, R. : "Numerical integration of Fluid Power Circuit Models Using Two-Stage Semi-Implicit Runge-Kutta Methods". Proceedings of the Institution of Mechanical Engineering Part C: Journal of Mechanical Engineering Science, Vol 208, 1994. pp 167-175.
- [3] Ellman, A.U.: "Methods for Simulating Steady State and Dynamic Behaviour of Two-Way Cartridge Valve Circuits". SAE paper 901584, SAE 1990 Transactions Journal of Commercial Vehicles, Section 2 99 (1991). Society of Automotive Engineers, Warrendale, PA USA 1991, p. 384-393.
- [4] Handroos, H.M. & Vilenius, M. : "Flexible Semi-Empirical Models for Hydraulic Flow Control Valves" . ASME Journal of Mechanical Design 113 (1991) 3, p. 232...238.
- [5] Handroos, H.M. & Vilenius, M. : " The Utilization of Experimental Data in Modelling Hydraulic Single Stage pressure Control Valves". Trans. ASME, Journal of Dynamic Systems, Measurement and Control 112 (1990) September, p. 482...488.
- [6] Voac Hydraulics : Catalog 9129 855-06 (US).