

P1-13

PUMP MODE PREDICTION FOR FOUR-QUADRANT VELOCITY CONTROL OF VALVELESS HYDRAULIC ACTUATORS

Christopher WILLIAMSON, Prof. Monika IVANTYSYNOVA

Maha Fluid Power Research Center
Department of Agricultural and Biological Engineering, Purdue University
225 S. University Street, West Lafayette IN 47907, USA
Email: mivantys@purdue.edu

ABSTRACT

Displacement-controlled actuation offers energy savings by eliminating the metering losses associated with hydraulic valves and allowing energy recovery. In a closed-circuit configuration, four-quadrant pump operation can be achieved. This paper considers displacement-controlled boom lift cylinders on a skid-steer loader. Undesirable pump mode oscillation is observed while rapidly lowering small loads. Avoiding this oscillation requires actuator pressure control, which cannot be directly achieved due to insufficient pump dynamic response. The authors propose a predictive observer to provide sufficient lead time for feedforward control of actuator pressure. Design and analysis are presented for a discrete time linear observer which predicts future system states by delaying the input signal. Successful state prediction is demonstrated through simulation and experiment.

KEY WORDS

Displacement control, Valveless, Pump control, Velocity control

NOMENCLATURE

A_p	cylinder piston area	[m ²]	c_v	coefficient of viscous friction	[kg/s]
A_r	cylinder rod area	[m ²]	k_{Li}	coefficient of internal leakage	[m ³ /Pa·s]
C_H	hydraulic capacitance	[m ³ /N]	m_{eq}	actuator inertial load	[kg]
F_f	cylinder friction force	[N]	n	pump rotational speed	[rev/min]
F_L	cylinder static load force	[N]	p	fluid pressure	[Pa]
H	cylinder stroke	[m]	v	cylinder velocity	[m/s]
J_D	mass moment of inertia about boom axis of rotation	[kg·m ²]	w	cylinder force due to boom weight	[N]
K_{oil}	fluid bulk modulus	[Pa]	α	cylinder piston area ratio	[-]
Q	flow rate	[m ³ /s]	Δp	actuator differential pressure	[Pa]
T	digital sampling time	[s]	ϕ	boom angle	[rad]
V	pump displacement volume	[cc/rev]	τ_p	time constant of closed-loop pump displacement control	[s]
V_{line}	actuator line volume	[m ³]			

INTRODUCTION

Energy-efficient hydraulic systems and controls for mobile machinery has become a major research topic in recent years. Since a large percentage of power loss is due to metering flow through directional control valves, alternative circuit designs that reduce or eliminate these losses offer significant energy savings. One proposed solution is displacement-controlled actuation, in which a variable displacement pump controls the motion of a single or double-rod cylinder. This “valveless” or “pump-controlled” concept offers several advantages over traditional valve control, including higher energy efficiency and linear dynamic characteristics.

DISPLACEMENT-CONTROLLED ACTUATION

DC Concept

The closed circuit design shown in Figure 1 was proposed by Ivantysynova in 1998 and has been developed by her team since that time [1]. A similar concept was proposed independently by researchers in British Columbia in the early 1990s [2-3]. An open-circuit configuration may also be used for displacement control [4]. Displacement-controlled actuation (DCA) is based on the same operating principle as a hydrostatic transmission. Actuator position and velocity are controlled by adjusting the flow rate through a variable displacement pump. The differential flow rate produced by the movement of a single-rod cylinder is compensated by a low-pressure line (supplied by a charge pump and accumulator) via pilot-operated check valves.

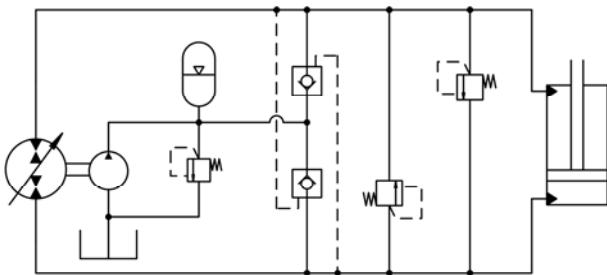


Figure 1 Displacement control circuit

DCA reduces power losses and allows energy recovery. Measurements of fuel consumption on displacement-controlled wheel loaders have shown savings of 15-25% [5]. Recent simulations of displacement-controlled excavators indicate reductions of up to 40% in total energy consumption [6].

Four-Quadrant Operation

The hydraulic pumps used for DCA must operate over center (reversible flow direction) in both pumping and motoring modes. Assuming that the direction of shaft

rotation remains constant, these pumps operate in four quadrants of a pressure-flow plane, as defined in Figure 2. This characteristic allows flexible flow control and power recovery. However, it also causes undesirable performance for certain operating conditions.

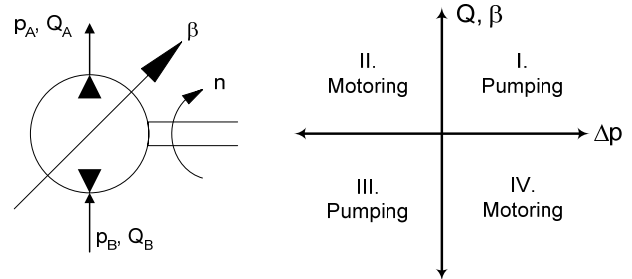


Figure 2 Definition of pump operating plane

$$\text{I. } v = \frac{Q_A}{A_p} \quad \text{II. } v = \frac{Q_B}{\alpha A_p} \quad \text{III. } v = \frac{Q_B}{\alpha A_p} \quad \text{IV. } v = \frac{Q_A}{A_p} \quad (1)$$

$$\alpha = \frac{A_p - A_r}{A_p} \quad (2)$$

Referring to Figure 1, the check valves connect the low-pressure side of the circuit to the charge line. If the net force on the cylinder changes direction, the pressure will also change to accommodate the load and the check valves switch accordingly. The problem that this presents is that the actuator velocity is a discontinuous function of the operating quadrant (Eq. 1). The area ratio α is typically between 0.5 and 0.75 (see Figure 4). If the pump changes operating modes while the cylinder is moving, the cylinder velocity will suddenly increase or decrease by a factor of $1/\alpha$.

Pump Mode Oscillation

Changing modes may be encountered in practice while lowering a light load at a high velocity. When the speed increases to the point where the friction force on the cylinder is higher than the load, the check valves switch, the pump shifts from motoring mode (quadrant IV) to pumping mode (quadrant III), and the cylinder velocity increases. Due to flow resistance and pressure resonance in the cylinder and lines, p_A may then rise above p_B and the pump shifts back to quadrant IV. This sequence then repeats itself, creating a limit cycle between pumping and motoring modes until the pump displacement is reduced enough to slow the actuator. An example of this phenomenon is shown in Figure 3 while rapidly lowering the boom of a displacement-controlled skid-steer loader with no load in the bucket.

Clearly, large oscillations in actuator velocity are undesirable and may be dangerous. From a hydraulic system design perspective, pump mode oscillation may be avoided or ameliorated by reducing cylinder velocity

and/or seal friction, increasing static cylinder loads, and increasing the α ratio. Where these solutions are not practical, feedback control may prevent oscillation. Several ideas for eliminating this effect by pressure and velocity control have been considered. As will be discussed in more detail later, the main difficulty in implementing an effective control law is excessive phase lag. The system dynamics are such that the time delay between control input and output measurement is too long for feedback control of the mode switching oscillations. What is needed is a way to anticipate an impending change in the pump's operating quadrant with sufficient time to change the pump displacement and avoid the transition. The goal of this paper is to develop a predictive pump mode observer that provides sufficient phase lead to allow four-quadrant control of actuator velocity.

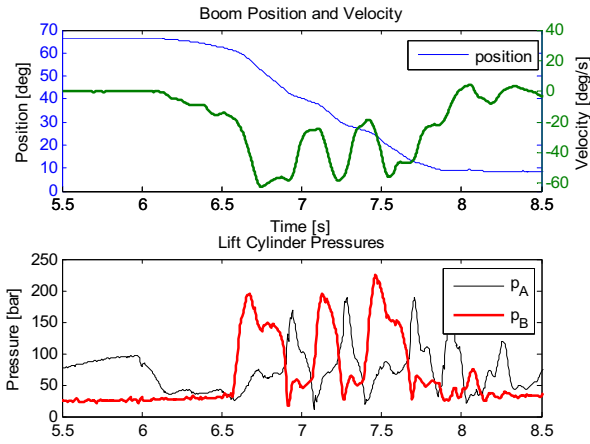


Figure 3 Measured pump mode oscillation

DYNAMIC MODEL

Pump Model

The DC skid-steer loader uses two variable displacement, axial piston swash plate type pump/motor units, one each for the boom lift and bucket tilt functions. Pump displacement is controlled by a hydraulic valve operating at a supply pressure of 15-30 bar. Pump dynamic response for small displacements depends on the bandwidth of the valve [7]. When a larger displacement is demanded, the swash plate velocity is limited by the maximum flow rate through the valve. There may also be signal transmission delay associated with the D/A amplifier between the microcontroller and the valve. DC actuation requires closed-loop control of the pump displacement. Simple proportional feedback is typical. Position/velocity control of the DC cylinders are arranged in a cascaded structure, with the pump displacement feedback as an inner loop [7]. The closed-loop pump dynamic response can reasonably be

approximated by a linear first-order system with a time constant of τ_p .

Actuator Model

The loader's lift cylinders are single-rod linear actuators, as shown in Figure 4.

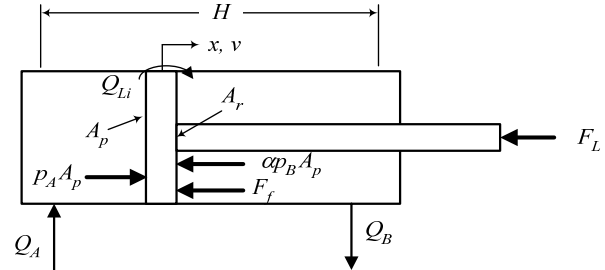


Figure 4 Single rod linear actuator

The actuator's equation of motion, assuming viscous friction only, is given by Eq. 3. Since the boom motion is actually rotational, the combined inertia of the boom and bucket must be expressed in terms of the actuator's linear motion as an equivalent mass in Eq. 4. Pressure build-up in the cylinder chambers is a function of flow rate and piston velocity, as in Eq. 6 and 7. Internal leakage across the piston seals is assumed to be only pressure dependent and external leakage is neglected.

$$m_{eq} \dot{v} = A_p (p_A - \alpha p_B) - c_v v - F_L \quad (3)$$

$$m_{eq} = J_D \left(\frac{\partial \phi}{\partial x_{cyl}} \right)^2 \quad (4)$$

$$\dot{p}_A = \frac{1}{C_{H_A}} \left[Q_A - A_p v - k_{Li} (p_A - p_B) \right] \quad (5)$$

$$\dot{p}_B = \frac{1}{C_{H_B}} \left[-Q_B + \alpha A_p v + k_{Li} (p_A - p_B) \right] \quad (6)$$

$$C_{H_A} = \frac{1}{K_{oil}} \left(V_{line} + A_p \frac{H}{2} \right) \quad (7)$$

$$C_{H_B} = \frac{1}{K_{oil}} \left(V_{line} + \alpha A_p \frac{H}{2} \right) \quad (8)$$

$$Q_A = nV \quad Q_B = \alpha nV \quad (9)$$

Constant hydraulic capacitances (Eq. 7-8) is assumed, which is defined at the center of the piston stroke. The flow rate to the cylinder is the product of pump speed n and displacement volume V , neglecting volumetric losses. Further, the rates of fluid flow entering and leaving the cylinder are assumed to be proportional.

State Space System Model

The state equations derived in the previous subsections can now be assembled into a single dynamic model in linear state space form. The state vector consists of the pump displacement V and cylinder position, velocity and pressure. The outputs are the measurable states, which are pump displacement (swash plate angle), cylinder position and pressure. The control input u is the desired pump displacement V_d .

$$\dot{\mathbf{x}} = \mathbf{F}\mathbf{x} + \mathbf{G}u + \mathbf{d} \quad (10)$$

$$\mathbf{y} = \mathbf{C}\mathbf{x}$$

$$\mathbf{x} = (V \quad v \quad p_A \quad p_B)^T \quad (11)$$

$$\mathbf{y} = (V \quad p_A \quad p_B)^T \quad (12)$$

$$\mathbf{F} = \begin{pmatrix} -\frac{1}{\tau_p} & 0 & 0 & 0 \\ 0 & \frac{-c_v}{m_{eq}} & \frac{A_p}{m_{eq}} & \frac{-\alpha A_p}{m_{eq}} \\ n & -\frac{A_p}{C_{HA}} & \frac{-K_{Li}}{C_{HA}} & \frac{K_{Li}}{C_{HA}} \\ -\frac{\alpha n}{C_{HB}} & \frac{\alpha A_p}{C_{HB}} & \frac{K_{Li}}{C_{HB}} & \frac{-K_{Li}}{C_{HB}} \end{pmatrix} \quad (13)$$

$$\mathbf{G} = \begin{pmatrix} \frac{1}{\tau_p} & 0 & 0 & 0 \end{pmatrix}^T \quad (14)$$

$$\mathbf{d} = (0 \quad -F_L \quad 0 \quad 0)^T \quad (15)$$

$$G_p(s) = \frac{p_A(s) - p_B(s)}{V_d(s)} \quad (16)$$

The output disturbance \mathbf{d} is the gravitational force on the cylinder due to the weight of the boom and bucket, which varies with bucket load and boom angle. It is essential that the model include this disturbance, since the pump's operating mode depends on static pressure, which is directly related to the weight of the boom and bucket.

System Analysis

In order to avoid pump mode oscillation, the cylinder pressure must be controlled. Consequently, the first topic for analysis is the pressure dynamics and the relationship between control input (pump displacement volume) and measured pressure outputs. Define a transfer function from u to y_2 - y_3 , as in Eq. 16. Roots of its characteristic equation for nominal parameter values are listed in Table 1.

For feedback control of system pressure, the pump bandwidth should be several times faster than the

pressure dynamics. It is clear from Table 1 that this is not the case. Direct pressure feedback is not an option, but feedforward pressure control may be possible. Feedforward control requires that the control input be applied with sufficient phase lead to produce the desired output. Since the cylinder pressure is not periodic, the controller must anticipate future outputs. This state prediction problem will be addressed in section 3.

Table 1: Nominal pole locations of $G_p(s)$

Mode	Frequency (Hz)	Damping (-)
Integrator	0	1
Pump dynamics	5.3	1
Pressure resonance	6.5	0.05

Many of the parameters in the system model are unknown, uncertain and/or slowly varying. Estimated maximum and minimum parameter values were obtained from machine specifications and model identification measurements. Figure 5 illustrates the pressure frequency response with respect to pump flow rate considering model uncertainty.

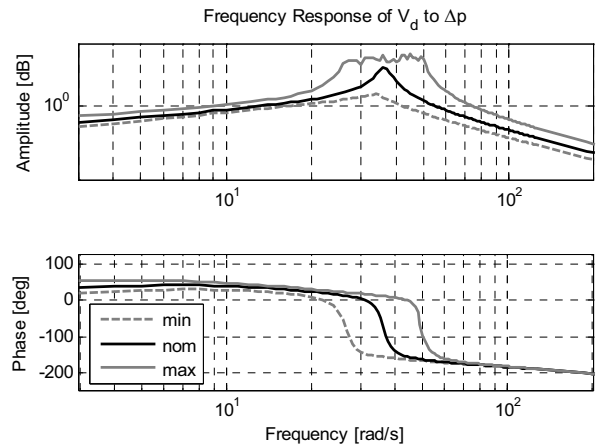


Figure 5 Frequency response of $G_p(s)$

From Figure 5, the maximum crossover frequency of the magnitude ratio is about 73 rad/s (11.6 Hz) and the minimum value is 41 rad/s (6.5 Hz), at which the phase lag is 173° and 161° , respectively. This corresponds to a time delay of 0.041 to 0.069 seconds between input and output.

PREDICTIVE OBSERVER

Observer Design

This section describes the design of an observer (estimator) that predicts future system states with sufficient lead time to allow feedforward pressure control. Analysis of \mathbf{F} , \mathbf{G} and \mathbf{C} from Eq. 10 indicates that all system modes are stable, controllable and observable. The observer will ultimately be implemented with a mobile digital microprocessor. Conducting design and analysis directly in the discrete time domain makes sense in order to avoid the faster sampling times required for a discretized continuous time design. To proceed with the observer design, the plant state-space model derived previously is converted to discrete time with a zero-order hold approximation. The basic design of a predictive observer is a direct result of the recursive form of the vector state equation, Eq. 17. Given the input vector \mathbf{u} for all time, any future state can be calculated from Eq. 18. For notational convenience, the load force disturbance \mathbf{d} is treated as an additional input.

$$\mathbf{x}(k+1) = \mathbf{A}\mathbf{x}(k) + \mathbf{B}\mathbf{u}(k) \quad (17)$$

$$\mathbf{x}(k+n) = \mathbf{A}^n\mathbf{x}(k) + \sum_{i=0}^{n-1} \mathbf{A}^{n-i-1}\mathbf{B}\mathbf{u}(k+i) \quad (18)$$

$$\hat{\mathbf{x}}(k+1) = (\mathbf{A} - \mathbf{L}\mathbf{C})\mathbf{x}(k) + \mathbf{L}\mathbf{y}(k) + \mathbf{B}\mathbf{u}(k) \quad (19)$$

$$\bar{\mathbf{x}}(k+n) = \mathbf{A}^{n-1}\hat{\mathbf{x}}(k+1) + \sum_{i=1}^{n-1} \mathbf{A}^{n-i-1}\mathbf{B}\mathbf{u}(k+i) \quad (20)$$

$$\tilde{F}_L = \mathbf{C}_d(\mathbf{y} - \mathbf{C}\hat{\mathbf{x}}) \quad (21)$$

$$\mathbf{C}_d = \begin{pmatrix} 0 & A_p & -\alpha A_p \end{pmatrix} \quad (22)$$

$$\hat{F}_L(k+1) = K_p\tilde{F}_L(k) + K_I \left(T \sum_{j=0}^k \tilde{F}_L(j) - \frac{T}{2}\tilde{F}_L(k) \right) \quad (23)$$

Of course, future inputs are unknown. It may be possible to predict states by assuming constant future inputs or by extrapolating from past inputs if u is smooth and varies slowly compared to the prediction time. Unfortunately, neither assumption is valid in the current application. The pump mode transition which the observer should detect occurs when the pump displacement is increasing and may occur rapidly, on the order of 50 ms. Assuming constant inputs fails to predict the event by underestimating the rate of pressure change, and extrapolating from previous inputs is too sensitive to noise. Since it is difficult to predict the future, another option is to delay the present. That is, by delaying the application of the control input to the plant, the future reference trajectory is known for a short time in advance. This may not be feasible for some applications, but may work for a skid-steer loader if the time delay is short enough. It is unlikely that a human operator will notice a lag time of less than 0.1 seconds.

The traditional design of a Luenberger observer takes the form of Eq. 19, where the estimated states are denoted with a carat. The predictive observer (Eq. 20) uses output feedback for the current state, but estimates future states ($\bar{\mathbf{x}}$) up to n steps ahead based solely on the input u .

For accurate state estimation, the observer must also consider the effect of the static load on the cylinder. The payload mass in the loader's bucket is unknown and variable, so an adaptive parameter estimation method is proposed. Estimation error may be calculated from the measured and estimated pressures, as in Eq. 21-22. The current disturbance estimate is adjusted with PI feedback on the estimate error (Eq. 23). A bilinear approximation serves as the integral term [8]. For estimation of future states in Eq. 23, constant load force disturbance is assumed.

Observer Analysis

Observer stability depends on the output feedback gains. The output injection matrix \mathbf{L} is chosen so as to place the observer poles around 40 Hz. This frequency is fast enough to provide rapid convergence while avoiding amplification of noise at higher frequencies. Feedback gains for the load force disturbance estimate are tuned empirically according to traditional PID rules. As this is a direct digital design, observer accuracy at the sample instances is independent of sampling time. Error due to intersample behavior may be avoided by sampling at least 10 times faster than the pressure frequencies [9], which is not a difficult requirement for the present application.

Observer robustness with respect to model uncertainty may be analyzed in the state space time domain. In Eq. 24, Δ_A represents the maximum singular error value of the \mathbf{A} matrix in the expected uncertainty range, where \mathbf{A}_0 represents the nominal model. A similar error value can be defined for \mathbf{B} , leading to a bound on the maximum state prediction error for n time steps in Eq. 25. Conversely, Eq. 25 could be used to find the maximum prediction time interval for a given error tolerance.

$$\Delta_A = \|\mathbf{A} - \mathbf{A}_0\|_2 \quad (24)$$

$$\tilde{\mathbf{x}}(k+n) = (\Delta_A \mathbf{A})^n \tilde{\mathbf{x}}(k) + \sum_{i=0}^{n-1} \left[(\Delta_A \mathbf{A})^{n-i-1} (\Delta_B \mathbf{B}\mathbf{u}(k+i)) \right] \quad (25)$$

VALIDATION

The predictive observer was demonstrated first in simulation with a nonlinear model of the skid-steer loader in Matlab/Simulink. This detailed model includes multi-body mechanics, pump displacement

dynamics, and loss models of various hydraulic components [10]. An observer sampling time of 5 ms and prediction time of 30 ms was used for both simulation and experiment. The duty cycle simulated with the nonlinear model consists of raising the boom and then lowering it quickly. As the boom drops, the lift pump oscillates between pumping and motoring modes. Simulation of the predictive observer with this model is shown in Figure 6. The observer successfully predicts that Δp is about to cross zero with >50 ms of anticipation.

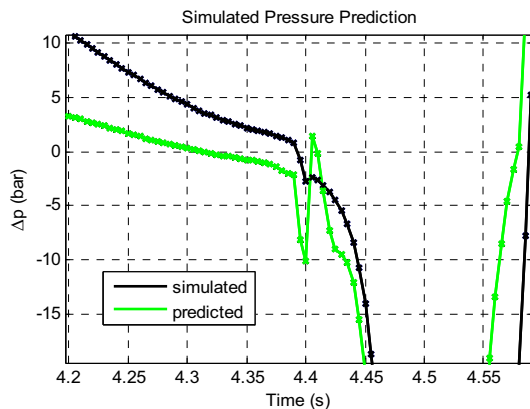


Figure 6 Results with simulated data

The same process was repeated experimentally. Observer pressure predictions with measured data are plotted in Figure 7. Again, the observer successfully predicts the zero crossing with about 60 ms of lead time.

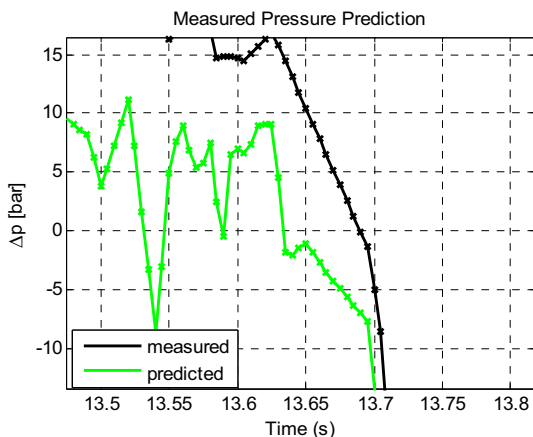


Figure 7 Results with measured data

A few limitations become apparent in the experiment. The state prediction is fairly sensitive to noise in the input command signal. Also, the load force disturbance estimate can negatively affect the pressure state prediction. Higher gains (and faster convergence) for the disturbance estimate tend to reduce the phase lead of the pressure prediction. A model-based method

for load disturbance estimation (such as considering boom kinematics) may reduce the necessity of high feedback gains. Efforts to reduce sensitivity to model uncertainty would also improve the observer's accuracy in practice.

CONCLUSION

Modeling and analysis of the DC hydraulic system indicate that 40 ms or more of lead time is required for feedforward control of actuator pressure and velocity. The intended skid-steer loader application allows a reference input delay of this duration. A discrete time observer is proposed to predict future changes in system pressure. Successful observer operation was shown with simulated and measured data. In future work, the proposed observer will be combined with a pressure and velocity control algorithm for pump-controlled actuation in four operating quadrants.

REFERENCES

1. Ivantysynova, M., 1998. "Energy saving hydraulic actuators for mobile machines", Proc. of 1st Bratislavian Fluid Power Symposium, pp. 47-58.
2. Lawrence, P.D. ; Salcudean, S.E. ; Sepehri, N., et al. 1995. Coordinated and Force-Feedback Control of Hydraulic Excavators. *4th Internat'l Symposium on Experimental Robotics, ISER'95*.
3. Hewett, Allan J. 1994. *Hydraulic Circuit Flow Control*. US Patent No. 5,329,767. July 19, 1994.
4. Heybroek, K., Larsson, J., and Palmberg, J.O., 2006. "Open Circuit Solution for Pump Controlled Actuators," *Proc. of 4th FPNI PhD Symp.*, pp. 27-40.
5. Rahmfeld, R. 2002. *Development and Control of Energy Saving Hydraulic Servo Drives for Mobile Systems*. PhD Thesis, VDI Fortschritt-Berichte. Reihe 12 Nr. 527. Düsseldorf: VDI.
6. Williamson, C., Zimmerman, J. and Ivantysynova, M., 2008. "Efficiency Study of an Excavator Hydraulic System Based on Displacement-Controlled Actuators", *Bath/ASME Symposium on Fluid Power and Motion Control*.
7. Grabbel, J. and Ivantysynova, M., 2005, "An Investigation of Swash Plate Control Concepts for Displacement Controlled Actuators," *Internat'l Journal of Fluid Power*, Vol. 6, No. 2, pp. 19-36.
8. Chiu, G. and Peng, H., 2007. *Digital Control Systems*. www.lulu.com/content/551583.
9. Goodwin, G., Graebe, S. and Salgado, M., 2001. *Control System Design*. Prentice Hall, pp. 381.
10. Williamson, C. and Ivantysynova, M., 2007, "The Effect of Pump Efficiency on Displacement-Controlled Actuator Systems," *Proc. of the Tenth Scandinavian International Conf. on Fluid Power (SICFP '07)*, Tampere, Finland, Vol. 2, pp. 301-326.