Proceedings of the 7th JFPS International
Symposium on Fluid Power, TOYAMA 2008
September 15-18, 2008

P1-09

VARIABLE LOAD SENSING AND ANTI-STALL ELECTRONIC CONTROL WITH SLIDING MODE AND ADAPTIVE PID

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ABSTRACT

A variable displacement axial piston pump with Pressure compensator and Load Sensing control is coupled to a diesel engine. The pump is in an open circuit topology for load control in a typical excavator, skid-steer or front loader circuit. The active torque generated by the engine is subject to overloads, due to uncontrolled priority and sum of torque requests from hydrostatic transmission, implements and upper-structure control. In order to avoid engine overload a third 2 ways proportional valve is added to the control circuit. The valve controls the delivery pressure on the load sensing valve port, and the pressure is managed by an electronic system. This circuit architecture allow the designer to use the valve as a variable load sensing system, modifying the load sensing differential pressure depending on working conditions. The control must be in real time and a feed forward control based on the valve characteristic map is useful and helpful but a feedback control is needed to correct the steady state error; a PID and a Sliding mode are superimposed, in order to react to high dynamic transient. The controls runs properly on the bench and the advantages are: more engine efficiency and continuously working capacity, adapting load sensing pressure a more efficient use of power is possible.

KEY WORDS

Load Sensing, Engine Overload, Pressure Control, Proportional Valve, Sliding Mode

NOMENCLATURE

CAN: Controller Area Network
D : Pump Displacement
I : Valve Control Current (mA)
LVDT : Linear Voltage Differential Transformer
Q : Pump Flow
P_C : Control Pressure
P_D : Delivery Pressure
P_L : Load Pressure
P_S : Pump Swashplate Piston Control Pressure
P_T : Load Sensing Calibration Pressure
PWM : Pulse Width Modulation Control Signal
β : Pump Swashplate angle (deg)
ω : engine rotational speed (rpm)

INTRODUCTION

The increasing cost of petrol and the increasingly restrictive European and American regulations for pollution control and reduction make electro-hydraulic solutions for open circuit application in load control in earthmoving machines, attractive and cost effective .s. In the market of compact and mid-size excavators, skid-steer loaders, front and backhoe loaders, the "state of the art" solution for the implements open circuit, is a variable displacement axial piston pump with tandem pressure control and load sensing control hydraulic valves. Manufacturers require new solutions in order to comply with regulations, to reduce machine maintenance costs, to increase reliability and safety. At the same time electronic systems are increasing their role in the hydraulic systems design, and a more strict relation between flexibility, adaptability and electronic systems modifies the market profile in the same way this happened fifteen years ago for passenger cars market.

THE PLANT

The plant designed and realized in a test bench at IMAMOTER Institute is equipped by a mechanically controlled diesel engine and a driveline that is coupled with a variable displacement axial piston pump with pressure and load sensing control.

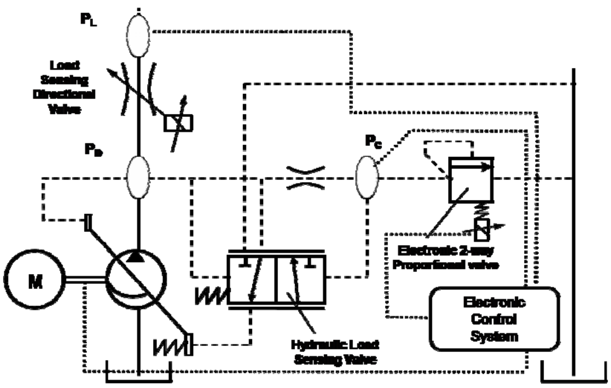

Figure 1 Circuit schematic (Pressure Compensator not shown)

The pump is designed for open circuit, reproducing the actual arrangement for an earthmoving machine. The load pressure is controlled by a servovalve placed in the position marked by P_L in Figure 1. The directional valves flow request is determined by an electronically controlled proportional distributor.
The Axial Piston Pump is a LVP type Casappa 65 cc. variable displacement axial piston pump equipped by a pressure compensator and a load sensing controller. PL is connected to the load sensing port of the LS compensator, while the delivery pressure is connected both to the P port of the pressure compensator and at the P port of the load sensing valve.

THE SOLUTION TARGET

The purpose of the project is to design an electro-hydraulic system that can represent a retrofittable solution for existing systems, capable to add the functionality of anti-stall of the i.c. engine and, if a torque estimate is available, capable to perform a torque control or torque limitation.
The aim is compliance to European regulations, and the target application is torque and speed limitation in mobile application. The implementation of the system is investigated with the purpose to inherit the stability of the hydraulic Load sensing controls, because a complete electronic solution isn't cost effective for mobile applications, yet, due to the higher cost of both a three way proportional valve with high pressure limit and a position feedback (typically an LVDT sensor) and an high-end electronic control system.

THE ELECTRO-IDRAULIC CONTROL SYSTEM

The simplest solution found does not imply the design of new integrated components. The system is equipped by a third electronically controlled two-way pressure regulation proportional valve . The valve characteristic is very attractive because of a good linearity of the regulated pressure as a function of the control current provided to the valve coil (Figure 2).

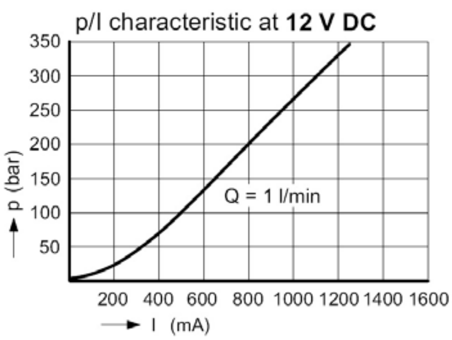

Figure 2 Proportional Valve Characteristic (Current vs. regulated pressure)

The valve P port is connected to the delivery pressure PD of the circuit with a calibrated orifice, the A port is connected to the load sensing port of the pump flow compensator, and the T port is connected to the reservoir.
Two pressure sensors are placed into the system: one at the load port downstream the directional flow request valve in order to acquire the load sensing pressure P_L, and another at the load sensing port of the load sensing valve, in order to sense the controlled pressure P_C (Figure 3).
The first goal of the electronic control system is to copy the load sensing pressure at the A port of the two way

proportional valve in order to reproduce the load sensing control conditions of the hydraulic circuit.
The load sensing control is properly performed by the system only if:

$$P_C = P_L \tag{1}$$

As a first need the controller and the valve must follow the load pressure with a limited delay, in order to avoid flow loss or pressure transient worse than the original hydraulic solution.
In order to regulate the anti-stall control a third , hall-effect engine speed sensor in the driveline was placed in the system. The only other sensor in the system is the current provided to the valve control feedback in the embedded electronic controller.

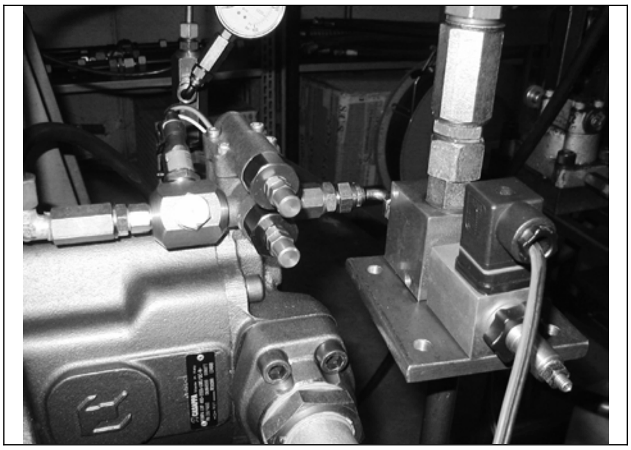

Figure 3 Pump LS and RP valves with the two-ways proportional electro-valve and P_C sensor

As per the system structure, PL and PC are measured by the electronic controller and PD can only be inferred. This control structure allows to control LS both in a traditional and in an adaptive way, simply introducing an offset by the electronic controller. PL is not exactly reproduced but rather an error is added that can be varied depending on the engine working point and/or machine working point, taking into account all the torque request by other hydraulic systems connected to the engine.
The new LS control equation can be written as:

$$P_C = P_L + E_T(\omega, P_L, T_\Sigma) \tag{2}$$

Where T_Σ is the sum of the Torque requests to the engine and E_T is the Electronic Tuning result as a function of the three variables in brackets.
The load sensing differential pressure at the load sensing hydraulic compensator

$$\Delta P = P_D - P_C = P_D - P_L - E_T(\omega, P_L, T_\Sigma) \tag{3}$$

allows to modify the ΔP as the implemented controls request. If, for example, an incipient stall condition is recognized, a ΔP greater than the actual (P_D-P_L) can be generated varying the 2-way valve working point, in order to reduce the pump flow and then the torque requested to the engine until the engine speed returns inside the allowable range. In the same way, if an external request is performed, for example by the CAN network, a similar action can take place.

SYSTEM CONSTRAINTS

As in the largest part of industrial and mobile applications, the cost reduction forces some design choices: Casappa called for the usage of a handy electronic control unit used for fan drive application, where real time constraints are less restrictive. A 10 MIPS (theoretical) 8 bit RISC microcontroller was used, optimizing the computational performance, reaching the minimum task repetition time of 3 ms, that we found as being the minimum for a sufficient control stability and performance.

CONTROLLER PERFORMANCE AND LOGGING TOOL CHARACTERISTICS

The control tuning difficulties and the high transient dynamics imposed the use of a functional tool logging not only the test bench physical data (pressure, engine speed and valve current) but also the internal control variables.

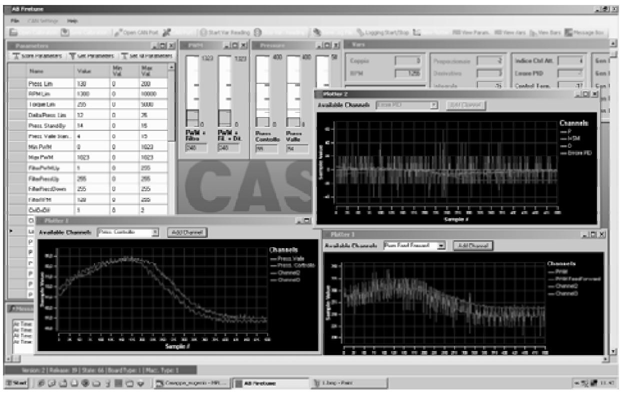

Figure 4 Calibration and logging System via CAN

Traditional communication tools based on a standard serial RS232 communication line were inadequate both for speed and computational resources consumption. A new tool based on intensive usage of CAN network was designed in C# (Figure 4), in order to use a higher communication speed and throughput, and to take advantage of the multi-buffering embedded CAN controller characteristics. This choice allows to manage and send in complete autonomy large amount of

information without lack of performance of the control system.
Even if a low cost 8 bit RISC microcontroller was used, a complete image of the control variables and system data is refreshed for each control task, i.e. every 3 ms, thanks to the presence of 16 message buffer that can be used both in transmission or in reception of the messages by the CAN network.

THE CONTROL STRATEGIES

The system regulates the $\Delta P = P_D - P_L$ in the load sensing valve in order to modify the pump displacement, and consequently both flow rate delivered and torque requested by the pump.

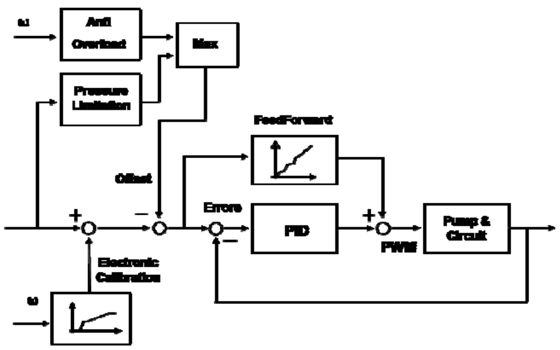

Figure 5 Control System Structure

In order to evaluate whether the system could substitute the traditional pump control structure with RP+LS control valves, a pressure control is added to the variable gain load sensing control and to the anti-stall control. The control system structure and priority is shown in Figure 5, where the electronic calibration is a function of the engine speed (ω). The feed forward function is a 2-way valve characteristic map, that helps to generate the right pressure P_C to be reproduced in real time once the P_L is acquired by the system supplying the exact current that correspond to the pressure target to be reached.
A control system structure based on a classic PID was designed, due to the basic pressure-follower characteristic conceived for the controller, even if tuning of parameters could be difficult, due to nonlinearities of the plant.

THE PID TUNING TECHNIQUE

In order to find the best parameter set for the PID controller, the closed loop Ziegler-Nichols method was applied, carrying the plant and the controller (Figure 6) at the limit cycle instability by a step input, logging the P gain of the PID controller and the oscillation frequency. The data found are significant for the best PID control parameter set definition.
Many test were performed and important differences were found in PID parameters definition as a function of flow rate and pressure; both for P gain (Figure 7) and for oscillation frequency (
Figure 8).

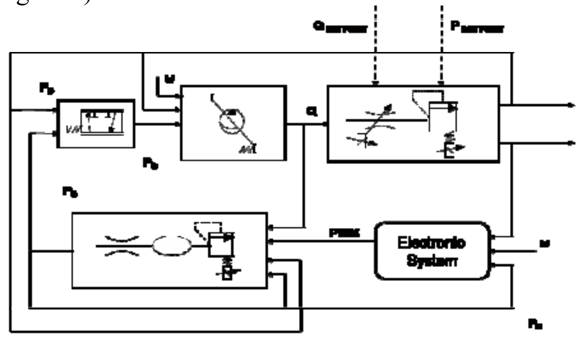

Figure 6 System Model

The parameter definition was thus the minimum set that grant the stability in all working points of the system.

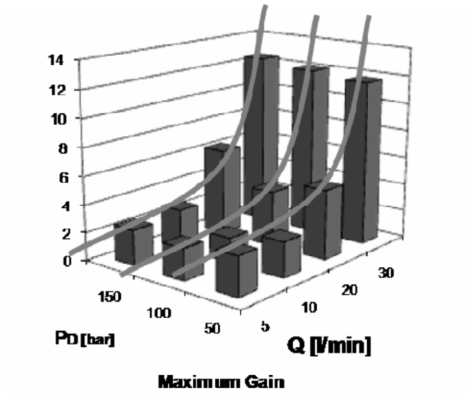

Figure 7 Maximum PID Gain

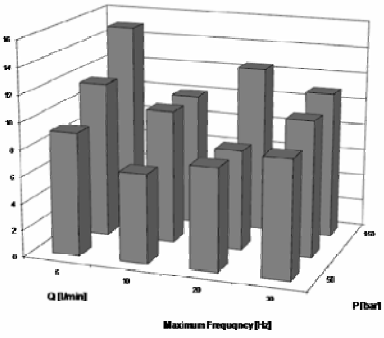

Figure 8 Maximum PID Frequency

A high sensitivity to displacement variation at low flow rates was found, according to the equation

$$\frac{\partial D}{\partial x} = \frac{D_{\max}}{\tan(\beta_{\max})} \cdot \frac{R^2{}_p - 2x^2}{(R^2{}_p - x^2)^{3/2}} \quad (4)$$

where R is the swashplate radius and x is the actuator displacement.
The criticality cannot be properly managed without a direct displacement measure being available in the system, therefore the PID parameters were settled based on the most critical working condition.
The forced parameter choice reduces the performance in some condition, and is not sufficient in order to grant the minimum performance requested by Casappa.
Consequently an analytical approach was attempted, and the stability margin was analyzed.

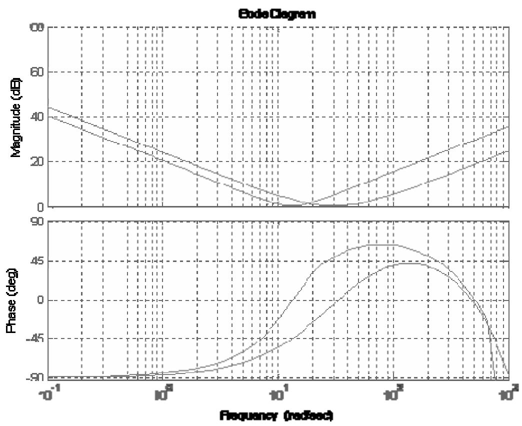

Figure 9 Stability Margin with Increased Derivative

The time domain expression of the controller action is expressed in (5):

$$u(t) = 1{,}05e(t) + 16{,}15\int_0^t e(t)dt + 0{,}017\frac{de(t)}{dt} \quad (5)$$

The experimental results point out a good precision but slow dynamics due to a limited phase margin. The integral gain was reduced while the derivative gain was increased in order to increase the phase stability margin. The Bode diagrams are shown in Figure 9.
The time domain expression of the modified controller action in expressed as:

$$u(t) = 1{,}05e(t) + 10{,}5\int_0^t e(t)dt + 0{,}06\frac{de(t)}{dt} \quad (6)$$

While the same expression in the frequency domain is shown in:

$$R(s) = \frac{0{,}06s^2 + 1{,}05s + 10{,}4}{s} e^{-0{,}0015s} \quad (7)$$

derived using 3 ms task control repetition rate.
In order to reduce the negative effects of an excessive integral term charge, an anti-windup feature was added in dynamic conditions.

THE ANTI-STALL CONTROL

The PID is the basis to perform all controls implemented in the system. In fact the control system structure allows to supply all the control actions as a Load Sensing calibration offset. The anti-stall control strategy is performed modifying the adaptive load sensing control when the engine speed is under a fixed limit value: a negative offset is added in (2), so a $P_C < P_L$ is reproduced and thus the pump flow is reduced, until the torque reduction is sufficient to reinstate the right engine working point conditions. In anti-stall condition P_C and P_L are different and the difference is function of the engine overload (Figure 10).

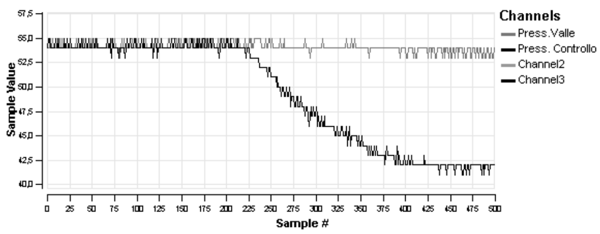

Figure 10 P_L and P_C diagram in anti-stall transient

THE ADAPTIVE LOAD SENSING CONTROL

The same principle was applied in the variable gain load sensing control calibration where a variable ΔP is obtained introducing a degree of freedom of P_C with respect to P_L, adding a positive offset in equation (2).
A variable flow rate target is then obtained in order to fit the torque requests to the engine working point. In Figure 11 an acceleration transient is reported, where the variance of P_C respect to P_L is placed in evidence.

THE SLIDING MODE

Even if the tuned PID demonstrates a good compromise between performance and stability, there are particular conditions where the excessive time delay (due to limited microcontroller performances and the PID controller linearity) evidence some control difficulty. In order to manage high dynamic flow request transient, starting from low or null oil flow, a sliding mode was added acting as a switching control with a dead band in the ΔP, i.e. the pressure error for the electronic LS control, and d(ΔP)/dt surface, its derivative.
The result obtained are shown in Figure 12, where a step transient in flow request is reported.

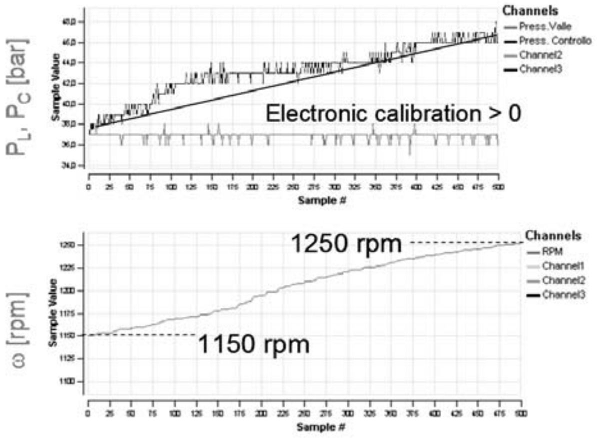

Figure 11 LS with electronic calibration characteristic

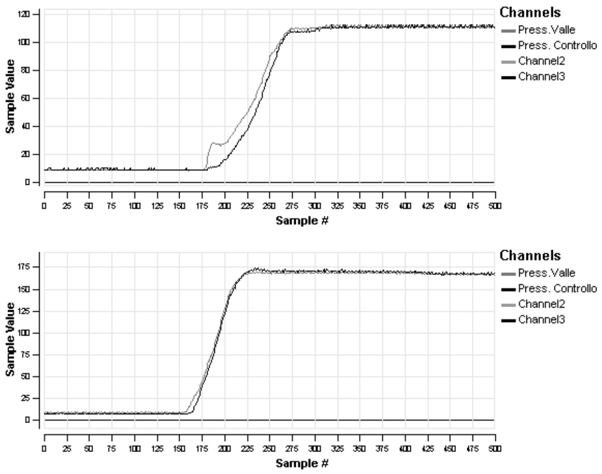

Figure 12 Flow request without and with Sliding Mode

PRESSURE CONTROL AND CONTROLS INTERACTION

An implicit maximum pressure control is obtained simply introducing a condition in pressure reconstruction expression:

if $(P_L + P_{Electronic\ LS\ calibration} < P_{max})$ *then* $P_C = P_L$

else $P_C = P_{max} - P_{Electronic\ LS\ calibration}$

when the P_D increases beyond a maximum limit, fixed by a set parameter, the increasing ΔP limits the pump displacement and consequently the pressure increases (Figure 13). Experimental results demonstrate that in identical condition the RP hydraulic control operates very rarely and just to reduce the pressure peak, due to the dynamics of the load sensing that is limited by the dynamics of two valves: the 2-way valve and the LS valve.

Figure 14 shows how the system can manage the transition between different control strategies priority: starting from a load sensing ΔP of 27 bar, at 120 bar (maximum pressure limit in this test) the difference between the two pressure is reversed and the flow rate is resettled to zero.

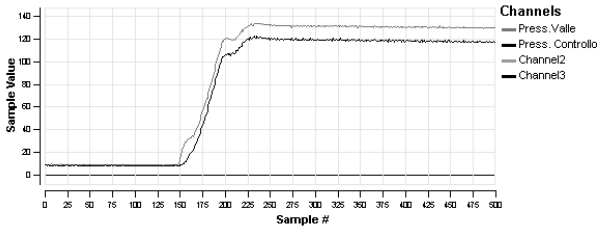

Figure 13 Variable gain LS implicit P_D Control

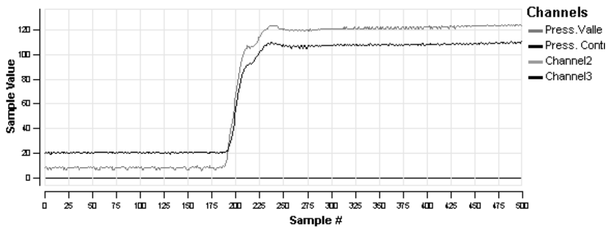

Figure 14 P_L vs. P_C compensator in a transient

CONCLUSIONS

The adaptive load sensing control presented in this paper appears as an interesting option to design flexible control systems. The system nonlinearities are managed exploiting the hydraulic controls layout and available experience. The performance is strongly constrained by the electronic system control computational rate and by the sensors placed in the plant. The lack of Swashplate position sensor is a practical limit in real flow evaluation and limits the maximum controller performance. Despite all these limits the application gave good results on test bench and will be proven on a real excavators, in order to increase experience and to design a commercial application.

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