

A Novel Pressure-Feedback Based Adaptive Control Method to Damp Instabilities in Hydraulic Machines

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ABSTRACT

Excessive vibration and poor controllability occur in many mobile fluid power applications, with negative consequences as concerns operators' health and comfort as well as machine safety and productivity. This paper addresses the problem of reducing oscillations in fluid power machines presenting a novel control technique of general applicability.

Strong nonlinearities of hydraulic systems and the unpredictable operating conditions of the specific application (e.g. uneven ground, varying loads, etc.) are the main challenges to the development of satisfactory general vibration damping methods. The state of the art methods are typically designed as a function of the specific application, and in many cases they introduce energy dissipation and/or system slowdown. This paper contributes to this research by introducing an energy efficient active damping method based on feedback signals from pressure sensors mounted on the flow control valve block. These feedback signals quantify the machine oscillation and are used to modify the electro-hydraulic flow control valve command provided by the operator. The controller is based on an innovative application of the non-model-based extremum seeking optimization algorithm for the tuning of the control parameters of a gain scheduling controller. The gain scheduler adapts the control parameters to the monitored pressure and operator input in order to provide the best control performance over the entire range of operating conditions.

Although the proposed methodology has a general applicability, the paper shows the vibration damping obtained on a midsize hydraulic crane. The utilized experimental set up, equipped with transducers (including wireless accelerometers, pressure and flow sensors), was used to prove the effectiveness of the new control strategy for several loads. Significant improvements in terms of both oscillations overshoot and settling time are shown for typical operation of the considered machine.

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1. INTRODUCTION

Mobile hydraulic machines are typically affected by vibrations of the machine structure which have to be controlled in order improve operator's comfort, machine life and controllability. Moreover, international standard regulate the permissible vibration level for the operator's health protection [1].

Vibrations arise from several features of the system dynamics, such as: oil compressibility and hose compliance, discontinuous functioning of some components (hydraulic domain); mechanical structure elasticity, end-stops, static friction (mechanical domain); particular boundary conditions dependent on specific applications (e.g. handling of hanging load, drive on uneven ground, etc.). Those hydraulic machines systems features highlight the main challenges of their vibration damping, namely: nonlinear nature of the

hydraulic and mechanical systems as well as the unpredictable features of mentioned boundary conditions.

Although research on damping methods for hydraulic systems began decades ago, the problem still persists, especially in mobile hydraulic applications. The techniques currently used present significant room for improvement. In fact, today's vibration damping methods are designed for specific applications and sufficient levels of comfort and controllability is achieved just for a limited range of operating conditions.

Existing pure hydraulic (PH) methods are based on resistive effect, capacitive effect or the combination of the two. Resistive effect is directly responsible for energy dissipation, and in some cases it may produce system slowdown. Capacitive effect is directly responsible for system slowdown, and it may indirectly produce in some cases energy. Additionally, PH methods are designed for a

specific operating condition of the machine, while for other conditions their control effectiveness is inevitably reduced because of the inherent nonlinearity of fluid power systems.

Among the many examples of applications of PH methods based on resistive effect, the case of machines using load holding valves ([2],[3],[4]) is one of the most significant. Vibrations in those applications are often damped by setting the load holding valve's opening pressure way higher than what necessary to hold the load or to balance aiding load conditions, thus introducing excessive energy dissipations. Examples are represented by the standard circuit of hydraulic cranes [3], or other construction machines like aerial boom lifts [4]. Another example of PH method is represented by the use of the insertion of dynamic orifices in the pilot lines connecting the pilot stage and the main stage of piloted flow control valves ([2][5]-[6][7]). The addition of such orifices reduces oscillatory behavior by slowing down the dynamics of the hydraulic system.

A common example of damping based on capacitive effect is the introduction of accumulators in the main transmission line connecting power supply unit to actuator ([8],[9],[10]). The additional capacitance of the accumulator increases the time required to build-up the pressure slowing down the dynamics of the machine. This effect is not always desired, and therefore in some cases additional semi-active control [11] is obtained by the introduction of on/off valves to exclude the accumulator when the operator requires a faster dynamics of the machine: however, along with those valves, energy dissipations are introduced.

Electrohydraulic (EH) technology is used to damp vibrations by sensing a feedback signal related to the oscillation extent and evaluating the proper command signal for the electrohydraulic control valves [12]. EH methods, combined with PH methods, have potential to reduce the influence of PH on the system and consequently of their typical drawbacks. Additionally, EH solution can be designed in a way suitable to adapt the control parameters to the entire range of operating conditions, in order to have maximum control effectiveness over the entire functioning range of the machine. Although many works were published regarding EH method for fluid power systems, very few cases of industrial application can be observed in earth-moving equipment or cranes, since many of the proposed methods are not feasible in practice. In particular, feedback sensors of proposed EH methods often suffer high installation cost or reliability issues. Moreover, they often use control methods whose parameter tuning is not guided by well-defined rules, and the theoretical results are in many cases not supported by adequate experimental validation.

For example, several methods are based on tracking control (TC) of linear displacement sensors or accelerometers mounted on the oscillating parts. This method was successfully implemented in several lower power applications, including: active suspensions ([13], [14]); earthquake simulators ([15], [16]); vehicles braking systems

[17], hydraulic robots [18], active damping seats ([19],[20]); other applications (e.g. [21]).

The use of pressure feedback (PF) control, based on pressure sensors giving information related to the pressure in the actuator chambers seems to be a more viable solution to reduce the drawbacks of TC-EH solutions. This because pressure sensors can be located in a protected area of the machine, such as in proximity of the flow control valves manifold, which is convenient in terms of reliability and cost of installation. The pressure signals are used to indirectly sense the oscillation of the machine and to enable appropriate electrohydraulic valves commands to damp the system vibrations. Considering the typical range of undesired oscillations (below 10 Hz), an implementation of PF techniques using traditional, low bandwidth flow control valves already on many machines is realistic.

This technology was applied to traditional valve-controlled systems by Krus [22] and Yao [23], and to displacement-controlled systems by Williamson [24]. The main reason why PF has still not found significant industrial application is due to the much greater complexity of the control algorithm formulation compared to TC since the oscillation is measured only indirectly through pressure feedback. Past effort in PF methods attempted to determine a transfer function between the feedback pressure and the machine oscillation. To make possible the determination of the objective function with models easily usable by the controller, a high level of system simplification is necessary. In the case of Krus and Yao the lack of provided experimental results makes this simplification questionable in practice.

In the present work, an innovative, non-model based formulation of PF control is presented. In particular the relation between feedback pressures and machine vibration is evaluated by means of a formulation of the extremum seeking (ES) theory [25]. An optimization algorithm is implemented to determine through an automatic procedure the parameters of a nonlinear gain scheduler (GS) controller [26]. The adaptability of the GS controller is used to extend the control effectiveness to the entire range of operating conditions of the machine.

2. CASE STUDY

The proposed control method is applied to the case study of a midsized hydraulic crane (Atlas®, model 125.1 A5, [28]) instrumented at Maha Fluid Power Research Center of Purdue University (Maha), figure 1.

The crane is equipped with pressure sensors, flow meters and accelerometers. The experimental set up has also the capability of monitoring the electrical signals provided to the electrohydraulic valves. The ES optimization algorithm to tune the controller parameters was performed on a multi-domain simulation model of the crane presented in [27], although the concept behind the proposed optimization method could have been entirely based on actual experiments

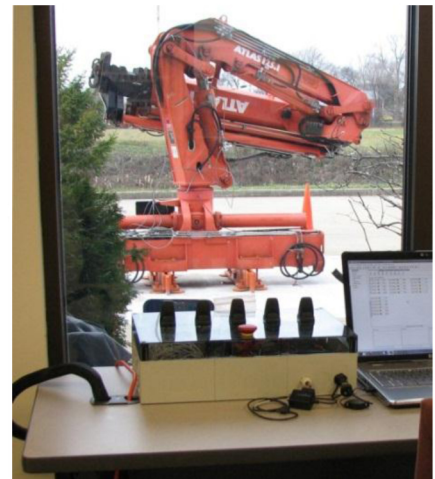
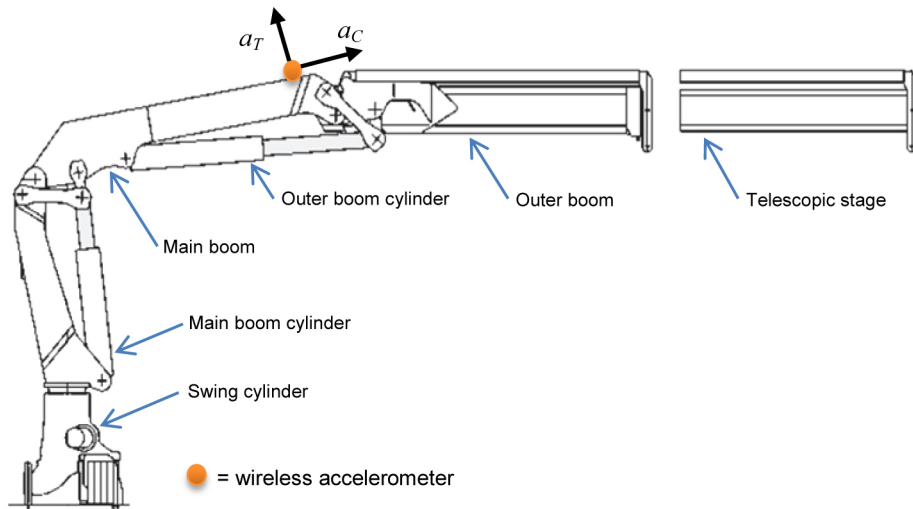


Figure 1. Drawing of the reference crane Atlas 125.1 A5 (Atlas Maschinen GmbH, 1996) and picture of the experimental setup at Maha (crane and control station).

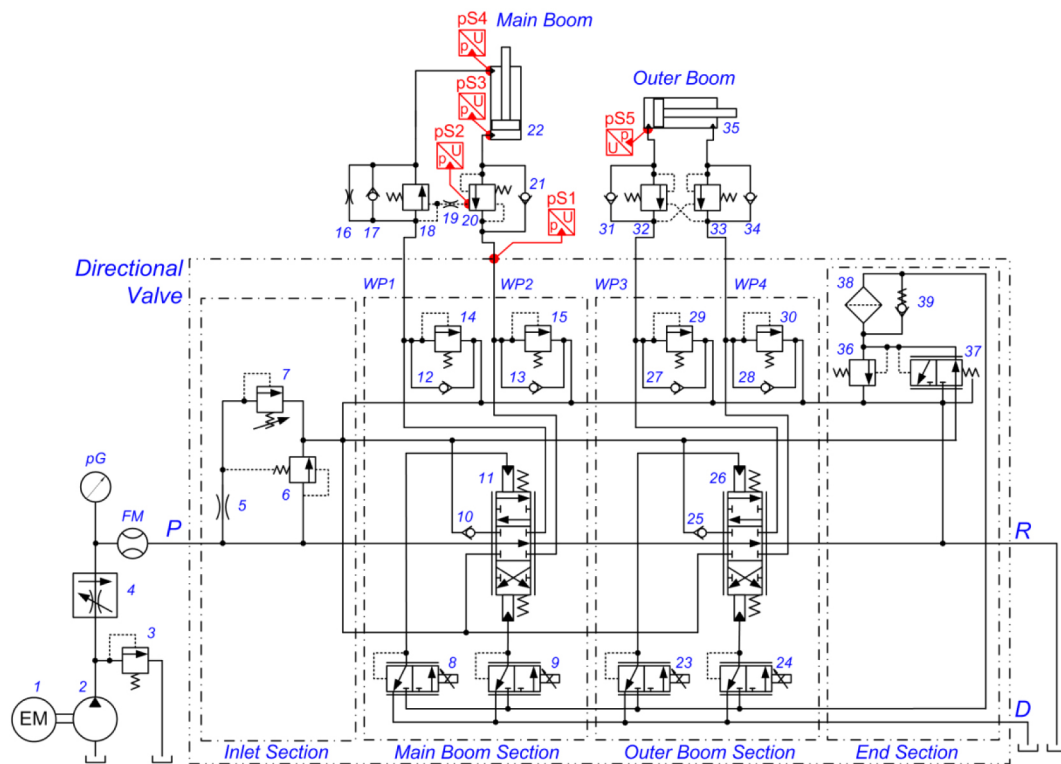


Figure 2. Portion of the reference crane hydraulic schematic (ISO 1219 standard of representation).

performed on the crane. The controller was implemented in a standard industrial unit operating at a maximum frequency of 333Hz (Parker IQAN MC-2). Finally, the control method was tested on real crane and compared with the standard control.

The hydraulic system of the crane utilizes a proportional directional valve [7] working with a constant input flow rate set to 45 l/min containing open center directional EH valves to control each actuator. The crane is constituted by main boom, outer boom and five telescopic stages. The swing

cylinder performs the rotation of the crane; the main boom cylinder performs the relative motion between main and outer boom; the outer boom cylinder performs the relative motion between outer boom and the telescopic stage. The crane is rigidly connected to a stand. This arrangement represents a simplifications respect to what occurs in applications in which the hydraulic machines is supported by flexible elements (e.g. shock absorbers of a truck mounted crane).

Table 1. Description of the components in Figure 2

Reference to Fig. 2	Description	Reference to Fig. 2	Description
1..3	Constant flow rate power supply.	16..19	Pressurizing valve to damp the main boom lowering.
4	Pressure compensated flow control valve.	20+21, 31+32, 33+34	Overcenter valves.
5..7	Pressure limiter integrated in the directional valve inlet section.	22	Main boom cylinder.
8, 9, 23, 24	Pilot stage of the directional valve.	35	Outer boom cylinder.
10+11, 25+26	Main stage of the directional valve.	pS1..pS5	Pressure sensors details (Parker IQAN SP500).
12+14, 13+15, 27+29, 28+30	Anticavitation valve on the working port of the directional valve.	FM	Flow meter (Parker SCFT-0380-PD).
36..38	Pressure reducing system which determines the inlet pressure of the pilot stage.	pG	Pressure gauge.

Due to the complexity of the problem, the focus of this paper is the study of the main boom control only. However, all crane actuators are similar systems and the control methodology can be applied to any of them, as it will be presented in a future work. For sake of clarity only the part of the hydraulic circuit pertinent to the performed tests is presented in the hydraulic schematic in [figure 2](#). [Table 1](#) describes the components represented in [figure 2](#). Although the outer boom is held in position during the test cycles, its hydraulic circuit contributes in determining the overall dynamics, and for this reason it was included in the schematic of [figure 2](#). In facts due to the compressibility of the fluid in the outer boom cylinder and the elasticity of the structure, the outer boom may oscillate when the main boom is operated.

The oscillatory tendency of the “open loop controlled” crane is damped exclusively through the pure hydraulic method of the additional resistance introduced by the energy dissipative settings of the overcenter valves (see [Figure 1](#), elements 20+21, 31+32, 33+34) and pressurizing valve (16..19). For the present work the original settings of these valves was used, so that the effectiveness of the proposed control methods will be showed respect to a solution representative of the state of the art. However, in a future work more energy efficient settings, in terms of valve pilot ratios and spring preloads (that usually promotes instabilities) will be investigated, so that the functionality of the machine will be ensured by the oscillation damping introduced exclusively by the proposed control.

3. CONTROL METHOD

3.1. Control Method Principle

The basic principle of the PF control used in this work is illustrated for a generic linear actuator in [Figure 3a](#). The pressure sensor can be mounted directly on a valve block including a shuttle valve used to determine the maximum pressure between those in the actuator chambers. The feedback signal from the pressure transducers is sent to the

control unit which uses this information along with the operator command to determine, according to the implemented control logic, a signal which modifies the original operator's input in order to minimize machine vibrations.

[Figure 3b](#) highlights that the control logic has to take into account that this feedback signal is just an indirect measure of the oscillation extent, since the transfer functions H_1 , H_2 and H_M relate the pressure to the load position: H_1 relates the pressure at the workport 1 to the pressure at the cylinder chamber 1; H_2 relates the pressure at the workport 2 to the pressure at the cylinder chamber 2; H_M relates the piston displacement to the load displacement. The proposed algorithm is non-model based; therefore the analytical formulation of those transfer functions is not required. Testing the real machine (or running simulations like in this work) the ES optimization is performed as described in section 3.2 for a set of operating conditions representative of the entire functioning of the machine, the step response of the machine is evaluated as a function of the control parameters. For each operating condition, the set of control parameters minimizing the oscillation in the step response are considered the best control parameters for that operating condition, and they are then stored in the GS controller.

The feedback signal p_{WP} is acquired by the control unit which high-pass filters it in order to determine the oscillating component of the signal. This component is then split in two parts: the first part is multiplied by the gain P , while the second parts is differentiated and multiplied by the gain D . Those two terms are then subtracted from the original operator command C to obtain the signal to the proportional valve U . In other words, this operation basically consists in the modification of the operator control signal with a component which is out of phase respect to the vibration phase. Intuitively, according to the sign of the high pass filtered pressure it is possible to distinguish two cases: if positive, the signal U will be reduced in order to limit the valve opening and reduce the pressure itself; if negative, vice-versa.