

# Energy Saving Potentials of a Novel Electro-Hydraulic Method to Reduce Oscillations in Fluid Power Machines: The Case of a Hydraulic Crane

Guido Francesco Ritelli and Andrea Vacca  
Maha Research Center, Purdue University

## ABSTRACT

A common issue of many hydraulic machines is given by oscillatory behavior. This is a stringent problem of several load holding machines, such as the case of the hydraulic crane taken as reference in this study. Several methods are traditionally used to dampen these oscillations to reduce hazards and improve productivity. However, these methods are often highly energy dissipative. As pertain to the case of hydraulic cranes, a good dynamic behavior is usually obtained through high energy losses introduced by the counterbalance valves. These valves, which use is often mandatory for load handling applications, are known to introduce oscillatory behavior in the machine actuators. However, these valves can also mitigate the machine dynamics when proper but energy dissipative settings are utilized. This study experimentally evaluates the energy amount saved when more energy efficient valve settings are used. The dynamic response is shown to be greatly affected by these changes.

The authors' research team has recently developed an innovative electro-hydraulic method that offers an alternative and model-free way for reducing actuator oscillations. This study experimentally verifies the advantages, in terms of potential energy saving, related to the application of such technique to the case taken as reference.

The experiments presented show how for a realistic operation of the crane with a more energy convenient settings of the counterbalance valve it is possible to drastically reduce the overall energy consumption up to 34%, without compromising the dynamic behavior of the machine, if the novel control technique is utilized.

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## I. INTRODUCTION

A critical aspect of several mobile hydraulic machines is represented by the oscillatory behavior of the actuators. Vibrations are one of the main source of safety and productivity issues on fluid power (FP) machines such as mobile cranes, telehandlers, aerial platforms, winches, wheel loaders and many other. When machine vibrations are transmitted to the operator (for example through the machine cabin), they can cause health hazard and comfort level reduction [1,2,3]. But in general, for all hydraulic machines, excessive vibrations create hazards and worsen controllability and productivity of the machine, [4].

Vibration control - meaning all the solutions to reduce machine oscillations - is a challenging topic in the FP field. This is mainly due to the strong non-linearities of the hydraulic

system which often preclude an accurate prediction of the system response and the application of simple control methods available for linear systems. Nevertheless, over the past decades a relevant effort was made by the FP research community to formulate solutions to achieve good dynamic performances of FP machines. The result of this research led to pure hydraulic solutions based on the introduction of additional resistances (like fixed or variable orifices in the main flow lines or in the pilot lines) or capacitance (like accumulators) to the hydraulic circuit and electro-hydraulic methods in which the damping is performed actively using components electrically actuated on the basis of a proper control strategies. Significant examples are given by the studies by Krus and Gunnarsson, [5], or by Yao et al, [6], for the case of hydraulic cranes, or by the work by Williamson et al [7] and Eggers et al [8] for the case of machines based on

displacement control technology. For many applications, electro-hydraulic solutions introduce additional complexity into the system and decrease the reliability level of the machine. For these reasons, in applications such as hydraulic cranes manufacturers prefer pure hydraulic solutions, which guarantee low cost and simplicity. This is the case of machines which utilizes counterbalance valves (CBVs). These valves are widely used in case of suspended loads, and they permit to accomplish the load holding function and to control the actuators in both cases of resistive or assistive loads, as it will be described in the following section (Section II).

It is well known that CBVs highly affect the dynamic behavior of the system. Several studies, such as [9,10,11,12], described the impact of different CBV settings on the oscillating behavior of entire hydraulic systems. CBV manufacturers also provide guidelines about how to choose the proper setting of the CBVs according to the oscillatory nature of the application, [13, 14]. While an unsuitable setting of the CBVs can increase the machine vibrations, a proper setting of the CBVs allows the mitigation of oscillatory tendency of the machine. For this reason, CBVs can also serve as oscillation damping element of the system.

In applications like hydraulic cranes, an acceptable dynamic behavior of the system is obtained through settings of CBVs which introduce high energy losses during the operation of the machine.

The increasing demand for more energy efficient hydraulic machines calls for more attention about the analysis of the sources of power losses in FP systems. For instance, the problem of investigating the actual energy consumption associated to different settings of CBVs was never addressed with a systematic and a quantitative method. The present research experimentally addresses this aspect for the reference hydraulic crane shown in Figure 1. The section IV will describe the details of the performed experiments, including a description of the chosen drive cycles representative of the typical operation.

After quantifying the large energy dissipation introduced by the typical settings of CBVs, this research will show the energy saving potentials of an alternative method for obtaining oscillation damping features. For this purpose, two different methods are available to improve energy consumption of CBV. The first one consists in developing new CBV designs for the specific application, and it represents the most common solution adopted by CBV manufacturers. Different designs for the CBV piloting system have been proposed; Sun Hydraulics recently introduced a new design, [15], called LoadMatch™, and it represents one of the most representative solutions in this regard. The second method, which this research is based on, consists in separating the primary function of the CBV (the load handling function) from the oscillating damping function. With that, the CBV can be designed promoting energy efficiency of the system. The oscillating damping feature is then demanded to a separate part of the system. This work

particularly considers an innovative control method applicable to all electro-hydraulic controlled systems. The method was first presented in [16] for the same reference machines considered in this work, and it showed high potentials for oscillation reduction without requiring changes in the hydraulic part of the system.

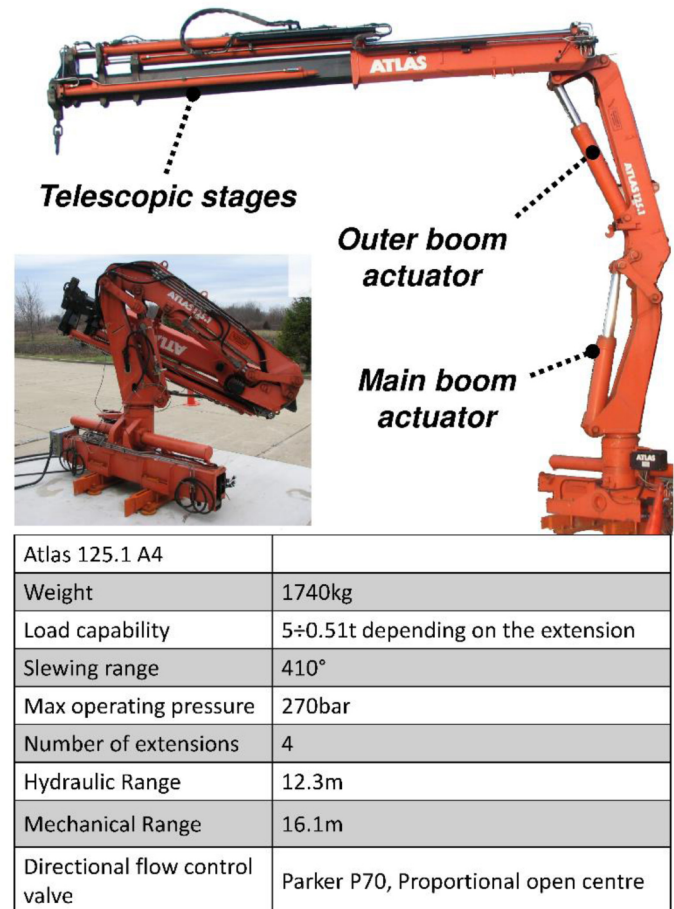


**Figure 1. Atlas 125.1 A4 crane assembled and instrumented at Maha Fluid Power Research Center and its main features**

Section IV summarizes the features of the control method used in this work, showing also how it can permit a good dynamic behavior of the reference crane, comparable to the its initial “manufacturer settings”, in which the CBVs are the solely elements used to guarantee stability.

To reach higher controllability a novel control strategy is applied. The control strategy has been recently developed at Maha Fluid Power Research Center and the success of its application is shown in [16].

After the definition of a working cycle representative of the regular operation of the machine, experimental results are shown in order to analyze the different energy consumption introduced by different CBV settings. The cycle is performed involving the machine in the moving of a barrel of about 600 [kg].

The energy consumption results are followed by a brief review of the control strategy and the results of its application.

The final discussion stresses the fact that adopting a proper control strategy to damp vibrations can drastically reduce the energy consumption of the machine.

## II. COUNTERBALANCE VALVES OPERATION

For the purpose of the present research, it is necessary to provide a concise description of CBVs operation and of the influence of the setting parameters. The simplified schematic of Fig. 2.a can be used to understand the operation of CBV for controlling both assistive and resistive loads. In particular, the figure shows a generic case where the cylinder extension has both resistive (opposite load with respect to velocity) and assistive phases (same direction for load and velocity).

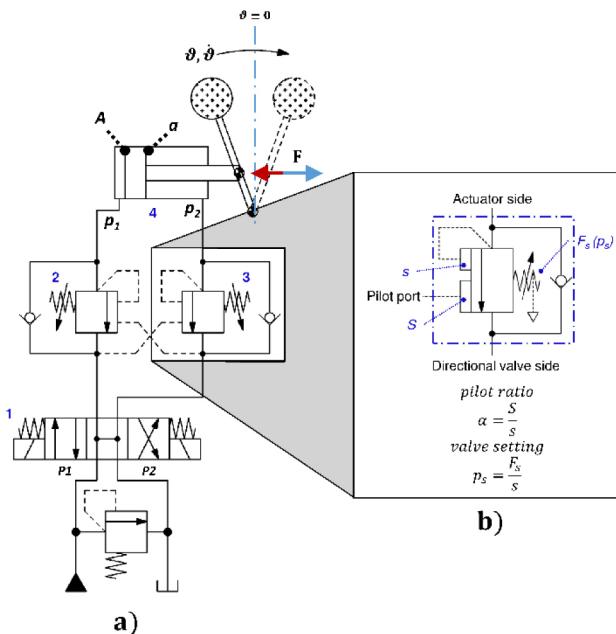


Figure 2. Typical CBV application and CBV's symbol and main features

As shown in Figure 2.b, a CBV is a three-port metering element that combines three typical valve functions in a single component. The main stage, pressure piloted, determines the proper valve opening respect to the flow returning from the load; in order to achieve the controllability of the load in all operating conditions, the main stage also operates as pressure relief valve; finally, a check valve permits the flow to bypass the main stage, when the flow is provided to the load.

The steady state characteristic of a CBV is mainly determined by two parameters:

1. Pilot ratio ( $\alpha$ ), which is the ratio between the areas which the pilot pressure ( $S$ ) and the actuator pressure ( $s$ ) act on;

2. Setting pressure ( $p_s$ ), which is the ratio between the force exerted by the setting spring ( $F_s$ ) and the actuator pilot area ( $s$ ).

With reference to the configuration shown in Fig. 2, the CBV accomplishes also the load holding function, when the actuator is not operated.

The detailed operation of the CBV can be understood by considering the case of the directional flow control valve (1, in Figure 2) in position P1 (cylinder extension). In this case, the check valve of CBV (2) allows the flow to freely go in the left side chamber of the actuator (4). The discharge flow of the right side chamber of the actuator is regulated by the CBV (3). During the cylinder extension the load changes from resistive (positive force) to assistive (negative force). Figure 3 is used to represent the pressure established in both actuator chambers as a function of the external force value.

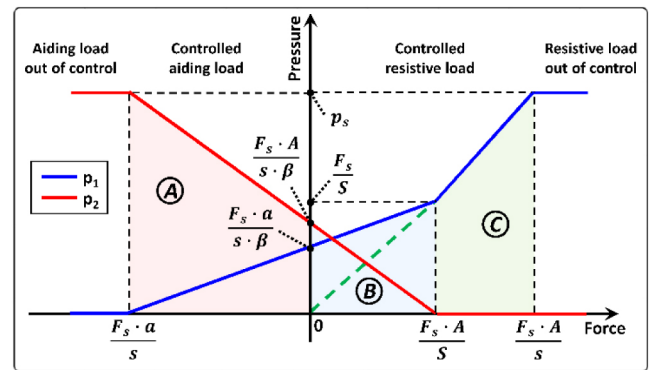


Figure 3. CBV characterizing graph. [17]

Obviously,  $p_1 > p_2$  for the resistive force case (zone C), while  $p_2 > p_1$  for the assistive case. At steady state conditions, the force balances at the cylinder and at CBV (3) provide the representation of Figure 3. Table 1 provides the force balance equations valid for the application described. The two sets of equations are obtained performing the force balance for the actuator and the CBV.

Table 1. Force balances for the example shown in Figure 2

| Assistive load                                                                                                                          |       | Resistive load                                                                                                                         |       |
|-----------------------------------------------------------------------------------------------------------------------------------------|-------|----------------------------------------------------------------------------------------------------------------------------------------|-------|
| $\begin{cases} p_1 = \frac{-F \cdot s + a \cdot F_s}{s \cdot \beta} \\ p_2 = \frac{A \cdot F_s - F \cdot S}{s \cdot \beta} \end{cases}$ | Eq. 1 | $\begin{cases} p_1 = \frac{F \cdot s + a \cdot F_s}{s \cdot \beta} \\ p_2 = \frac{A \cdot F_s + F \cdot S}{s \cdot \beta} \end{cases}$ | Eq. 2 |
|                                                                                                                                         |       |                                                                                                                                        |       |
| Where:                                                                                                                                  |       |                                                                                                                                        |       |
| $\alpha = \frac{S}{s}; \beta = A + a \cdot \alpha$                                                                                      |       |                                                                                                                                        |       |

Assuming a constant inlet flow, for the resistive load case a certain force at the load is required to fully open CBV (3). For low resistive forces, the throttling function of CBV (2)

introduces a further pressurization of the supply system. For assistive loads, the action of CBV (3) permits to satisfy the force balance at the cylinder, thus permitting the load control also in these conditions. The power demand of the system is always given by the pressure level at the supply,  $p_1$ . It is possible to deduce that only for the green region in Fig. 3 the CBV (3) is not introducing losses. In all the other cases (blue and red regions of Figure 3), CBV introduces over pressurization (thus power losses) into the system.

As shown in Figure 3, both  $p_1$  and  $p_2$  curves are defined by the valve settings and actuator parameters introduced in Figure 2, namely the pilot ratio ( $\alpha$ ), the preload of the spring ( $F_s$ , its equivalent pressure,  $p_s$  is called valve setting) and the actuator areas ( $A$  and  $a$ ). In particular, Figure 4 highlights the effect of the variation of  $F_s$  and  $\alpha$  on the characterization graph. The figure shows just the consequences on the pressure  $p_1$ , directly connected to the power required by the system when the valve 1 is in position P1. Increasing spring preload or decreasing the pilot ratio usually promotes higher energy consumptions.

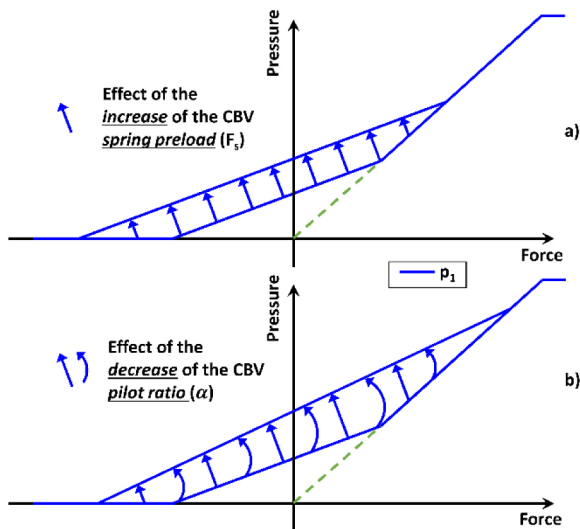


Figure 4. Effect of the CBV parameters on the characterizing graph

Similar considerations can be made for the control valve in position P2.

The steady-state based representations of CBVs operation provided in Figure 3 and Figure 4 can also be utilized to provide simplified interpretations about the effects of the valve when it is utilized to damp system oscillations. In particular, Figure 5 shows how increasing the pilot ratio, the area in which the CBV (3) is throttling increases. The area in which CBV is fully open (thus inactive in the system) decreases with  $\alpha$ . This is in line with the general requirements of operating with low pilot ratios to increase the damping features of the CBV given by a wider valve control range.

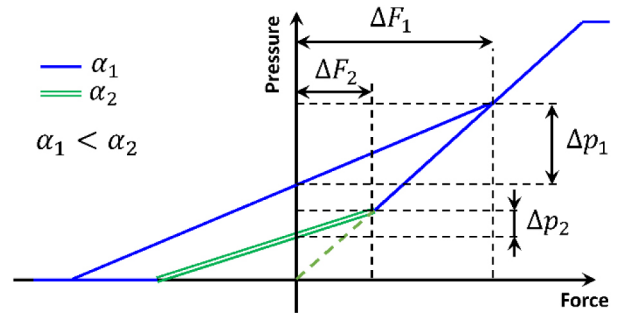


Figure 5. Different control ranges on the characterizing graph of the CBV depending on their different pilot ratio

A more detailed analysis of the dynamic behavior of systems based on CBVs can be found in [9, 10, 11, 17]. These mentioned works considers also aspects neglected in this research, such as the influence of the hysteretic behavior introduced by the internal seals of the valve.

### III. THE REFERENCE MACHINE

The present study takes as particular reference machine the truck loading crane shown in Figure 1. Based on a commercial product (Atlas 125.1 A4 [18]), the reference hydraulic crane was assembled and instrumented at Maha Fluid Power Research Center of Purdue University to support general research on valve controlled hydraulic systems. The entire hydraulic layout of the machine is almost unaltered with respect to the commercial version: the directional valve block is the original one, while the CBVs are replaced with new valves with adjustable spring settings, similar to those typically assembled on this kind of machine. The flow supply is represented by a flow control valve which simulates a fixed displacement pump (running at fixed speed) that typically supplies truck loading cranes.

Figure 7 shows the partial hydraulic schematic of the reference machine, limited to the two actuators involved in the experiments discussed in the following part of the paper (outer boom and main boom, Figure 1). The figure details all the sensors installed into the hydraulic circuit, whose details are provided in Table 2.

In addition to the sensors indicated in Figure 7, a tri-axis accelerometer was also used to quantify the vibrations of the outer boom actuator. The location of the accelerometer is shown in Figure 6. The accelerometer permits to quantify the tangential acceleration of the outer boom. To achieve high amplitude for the acceleration signal, the tip of the outer boom arm was chosen as location of the sensor. This permitted to obtain noise levels from the sensor of significantly lower amplitude respect to the acceleration signal of interest of this research.

The acquisition system is composed of industrial modules developed by Parker Hannifin, which include one IQAN-MC2 and two IQAN XC-10. The acquisition system allows for the implementation of real-time control and the