EVALUATION OF VELOCITY CONTROL CONCEPTS INVOLVING COUNTER BALANCE VAVLES IN MOBILE CRANES

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ABSTRACT

This contribution reports about some simulation and experimental studies carried out on flow control concepts used in mobile loader cranes equipped with counter balance valves. First it is shown that a conventional flow control system based on meter-in, is potentially unstable. Then three concepts based on meter-out is described. It is found that none of the concepts can fulfill all the requirements asked for by crane manufactors, and the performance will to some degree depend on the specific application.

KEYWORDS: Velocity control Balance valve Mobile crane

NOMENCLATURE

M_t : total mass of piston and load referred to piston
b_e : effective bulk modulus
P : pressure
Q : flow
V : volume
K_q : overcenter valve flow gain
K_{qp} : overcenter valve flow-pressure coefficient
Q_P : displacement flow of piston

Subscripts:

r: piston side *f*: piston rod side

Other used symbols are shown in Fig. 1

INTRODUCTION

Counter balance valves are used to provide smooth control, preferably load independent, when lowering loads, to give protection in the event of a hydraulic hose failure and, in most circuits, to provide overload protection for the actuator.
A check valve allows freee flow into the actuator, then holds and locks the load against movement. A pilot assisted relief valve section will give controlled movement when pilot pressure is applied. The relief valve is nomally set to open at a pressure at least 1.3 times the maximum load induced pressure but the pressure required to open the valve and allow movement depends on the pilot ratio of the valve.
For optimisation of load control and energy usage, a choice of pilot ratios is available [4].
The development within mobile fluid power systems is in general directed towards increased functionality, improved response and controllability. For increased controllability the directional valve is typically a spool type valve with pressure compensation, and in loader cranes the cylinder is often equipped with a counter balance valve (CBV). Both of these valves handle a wide variety of functions. However, the combination is a well known source of oscillatory behavior or even instability [5], [6]. Legislations and safety reasons though make them a necessary element in many loader cranes, and put pressure on the requirement for velocity control.
This paper describes experience with the design and operation of different velocity control concepts in systems involving counter balance valves.

BASIC SCHEME

Some general proporties of the flow control schemes under study can be found by considering the conventional system seen in Fig. 1.

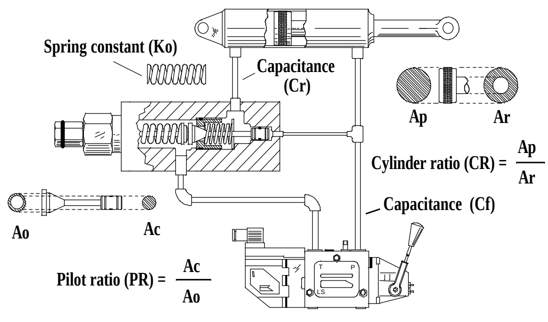

Fig. 1 Scheme of conventional system

Following the approach in [2], three transfer functions are introduced

$$H_f(s) = \frac{1}{C_f s}\,; H_r(s) = \frac{1}{C_r s}\,; G_M(s) = \frac{A_r^2}{M_t s} \qquad (1)$$

The basic equations, describing the dynamics of the system in Fig. 1, are for purpose of reducing the equations, conveniently represented in a block diagram, as shown in Fig. 2.

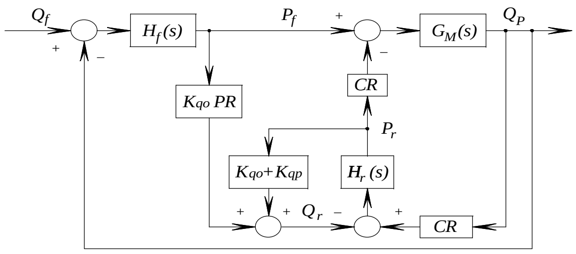

Fig. 2 Block diagram of conventional system

Closing the inner loops gives the open loop gain function $G_o(s)$.

$$G_o(s) = K \frac{1 + s/w_{2t}}{s(s^2/w_{3n}^2 + 2z\,/w_{3n}s + 1)} \qquad (2)$$

Where

$$K = \frac{b_e K_q A_o (1 + PR\,CR) + K_o K_{qp} b_e}{K_o V_f\; CR^2}$$

$$w_{2t} = \frac{b_e K_q A_o (1 + PR\,CR) + K_o K_{qp} b_e}{K_o V_r}$$

$$w_{3n} = \frac{A_P}{\sqrt{\frac{V_r}{b_e} M_t}}\,; \quad z = \frac{1}{2} \frac{K_q A_o + K_o K_{qp}}{K_o A_P} \sqrt{\frac{b_e M_t}{V_r}}$$

Where $K_{qo} = K_q A_o / K_o$, $C_f = V_f / b_e$, $C_r = V_r / b_e$.

From the above mathematical description**,** stability and other performance characteristics can be computed.
The characterististic equation for the system described in Fig. 2 can be written as

$$a_3 s^3 + a_2 s^2 + a_1 s + a_o = 0 \qquad (3)$$

Where

$$a_3 = C_r C_f M_t\,; \qquad a_o = A_r^2 \{K_{qo}(1 + PRCR) + K_{qp}\}$$
$$a_1 = A_r^2 (C_f CR^2 + C_r)\,; \qquad a_2 = C_f M_t (K_{qo} + K_{qp})$$

Using the Routh-Hurwitz criterion, it is necessary and sufficient that the coefficient be positive and $a_2 a_1 > a_o a_3$. See also [1] and [2].

Therefore, for a stable system, we require that

$$\frac{V_f}{V_r} > \frac{PR}{CR} \cdot \left(\frac{K_{qo}}{K_{qo} + K_{qp}} \right) \qquad (4)$$

In a crane we often have high inertia loads combined with static loads. This means that the pressure sensitivity of the overcenter valve will be high and the term in the brackets in Eq. 4. will tend to 1. A lower pilot ratio will increase stability, but from this simple stability result it is obvious that the stability margin of the overall system will be very small as the hydraulic cylinder is a full stroke component. The enclosed volumes V_f and V_r on the meter-in and the meter-out side, respectively – influences the gain and the damping in the system. Hence, a big V_f and a small V_r leads to increased stability (Fig. 3), and vice versa (Fig. 4).

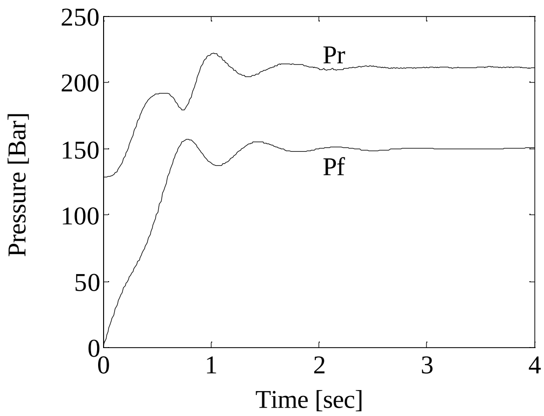

Fig. 3 Pressure response with step in Q_f

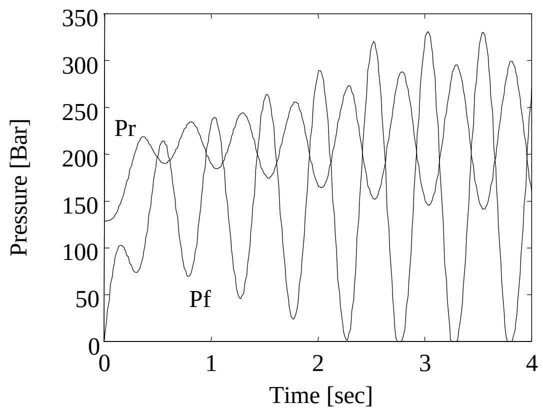

Fig. 4 Pressure response with step in Q_f

A commen way to try to achieve a more smooth and stable valve opening and load lowering is by adding different pilot features, i.e. different combinations of damping throttles, bypass lines and non-return valves in the pilot line to the counter balance valve.

EXTENDED SCHEMES

Clearly, the conventionel scheme is not suited to flow control. However, it is the lack of pressure control in a normal meter-in flow control system that causes the problem.

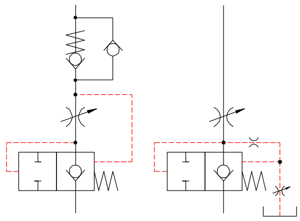

Fig. 5 Two ways of achieving pressure control

This suggest that one should look at meter-out concepts. A counter balance valve is a pilot operated valve, and ideally a conventional pressure compensated directional valve can not control pressure – though it is natural to look at meter-out systems. The basic idea is to control pressure in the meter-in side and flow in the meter-out side. In the control schemes presented in the next sections the pressure in the meter-in side is controlled by two methods, as shown in Fig. 5. The left drawing symbolizes a pressure compensated directional valve with a sequence valve placed downstream. Thereby is it possible to control the pressure in front of the sequence valve as function of the flow. In the figure to the right there is an adjustable pressure drop across the main orifice. Letting the adjustable spring force depend on the lever motion and keeping the main orifice sufficiently open, the valve will behave like a pressure control valve. In the three schemes presented next the concept figures is shown with the flow control-sequence valve combination.

Scheme A

In this scheme, Fig. 6., the meter-out control is achieved by letting the CBV act as a pressure reducing valve. This reduced pressure compensates the meter-out orifice passage created by the proportional valve
When using a sequence valve the opening characteristic must be matched to the characteristic of the CBV, in such a way that the outlet flow is higher than the inlet flow. The meter-in and meter-out side is then decoupled and the system operates stable. Using either pressure control or a sequence valve the pressure characteristic should be independent of the spool movement, and held constant. In case of hose break the lowering speed is increased. If the load changes direction, the concept with a sequence valve will change to meter-in flow control.

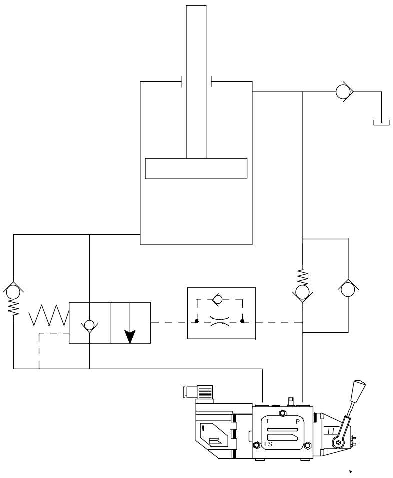

Fig. 6 Scheme A

Using pressure control some lever movement must be used to create a higher pressure, and the velocity will be load dependent.

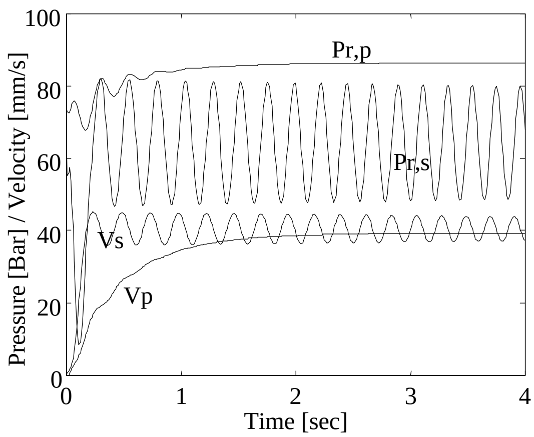

Fig. 7 Simulation results with step in Q_f

In Fig. 7. some simulation results are shown, where a stpe has been applied to the spool in the directional valve. Subscript ”p” stands for “pressure control” and subscript “s” stands for “sequence valve”. The upper curves is the pressure in the piston side of the cylinder and the lower curves is the piston velocity. From the figure it is clear that the system is rather undamped.

Scheme B

In this scheme a pressure reducing valve is placed before the outlet orifice of the directional valve, thus acting as a meter-out presure compensator (Fig. 8).
The pressure characteristic in the inlet side is made so that the CBV always is fully opened.
There is no need to match the area characteristics of the pressure controlling elements and the CBV.

As with scheme A, the concept with a sequence valve will change to meter-in flow control and with pressure control the scheme will be load dependent if the load goes over center.

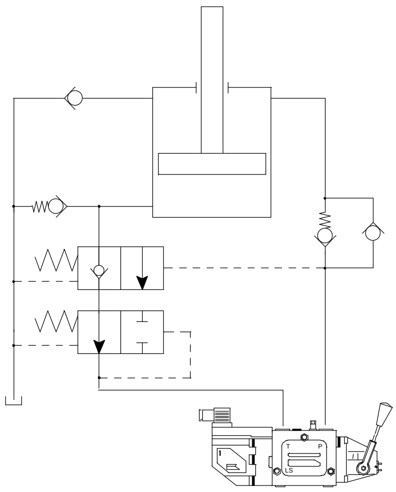

Fig. 8 Scheme B

If the compensator is placed near the CBV away from the proportional valve there might be some dependancy on oil temperature due to the restriction in the return line between the CBV and the proportional valve. Simulation results are shown in Fig. 9.

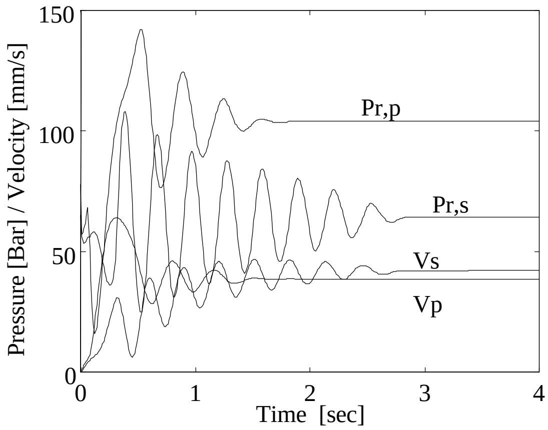

Fig. 9 Simulation results with step in Q_f

An advantage of the system is that it works with most counter balance valves.

Scheme C

The contradistinction between this scheme and the other two is that pressure compensation is done over the orifice of the CBV and not over the return orifice of the proportional valve (Fig. 10).

The velocity control is related to the postion of the spool in the CBV, and thereby to the pressure in the inlet side.

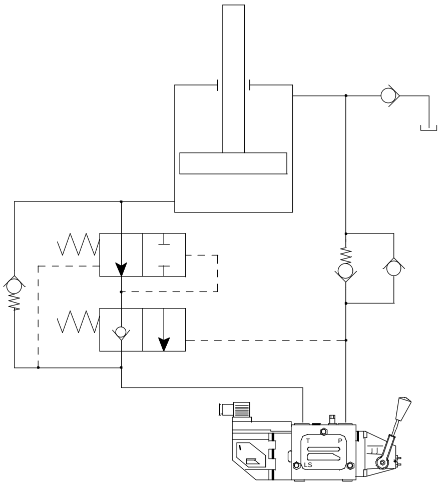

Fig. 10 Scheme C

Hence some adjustment is necessary. The speed is unchanged in case of hose break. Simulation result is shown in Fig. 11.

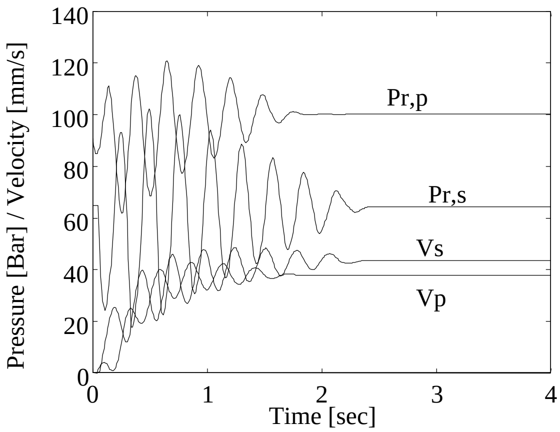

Fig. 11 Simulation results with step in Q_f

Evaluation

Although all three schemes can be made stable, scheme C only have the advantage of unchanged speed in case of hose break. Adjustment can be difficult as the system holds many parameters that may obstruct the adjustment. Also the outlet compensator must be sealed to the spring chamber in order to prevent leakage from cylinder to tank.

In scheme A and B with a sequence valve, the characteristic of the inlet restriction must match the characteristic of the CBV, so that the opening of the CBV corresponds to the inlet flow. Further the outlet compensator must be adjusted to a setting which breaks

the conventional control loop, though the flow out of the cylinder must slightly exceed the flow into the cylinder. In scheme A the pilot pressure must be held constant, and in scheme B the CBV should just be kept open.
Based on the above discussion scheme A and B were chosen for experimental investigation. Scheme A for its simplicity and relatively stable behaviour, and scheme B with pressure control for its robustness and stability properties.

EXPERIMENTAL RESULTS

The simulation and measurements were carried out with an HMF 680 mobile crane shown in Fig. 12. Only the 1st boom cylinder was used in the study.
All simulations and measurements were made with the crane extended 8 meter and with a payload of 500 kg, the input being a step in required velocity.

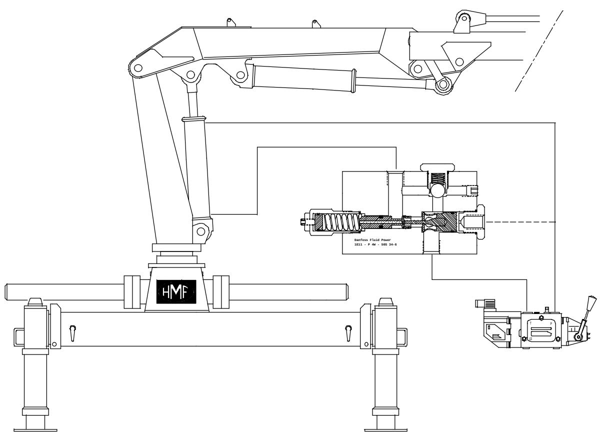

Fig. 12 HMF 680 mobile loader crane

By changing the spools in the proportional valve it is possible to have load-independent flow control or pressure control. In Fig. 13 is shown some experimental results with scheme A and B .

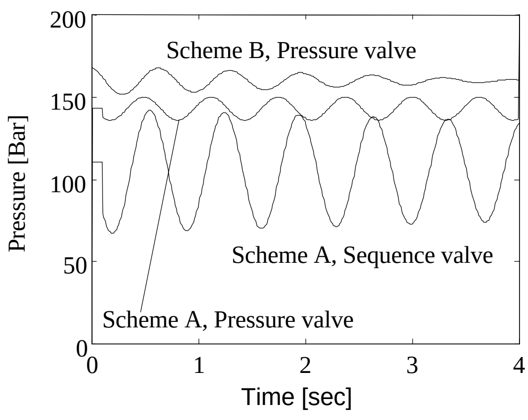

Fig. 13 Experimental results with step in Q_f

In the figure we see the same tendency as in the simulation study. Scheme A being prone to load oscillations and the stable behaviour of scheme B with pressure control.

CONCLUSIONS

In this study the main goal was to compare three different flow control schemes for mobile cranes equipped with counter balance valves. They are all based on meter-out control, utilizing pressure control in the meter-in side. The pressure control is achieved either by letting the meter-in side cavitate or by controling the pressure by the lever on the proportional valve. They all show a stable behaviour. Scheme A and B were chosen for an experimental study that corresponded well with the simulation results.
Despite their relative advantages and disadvantages it is very difficult to compare the schemes. This is because an important part of the function is the operator's "feeling" of the control of the machine.
Most of todays hydraulic systems have a standard single-spool configuration and provide limited functionality while the controlling land affects both sides of the actuator. The scheme B and C are more complex and require use of pressure compensators for controlling meter-in and meter-out seperately. No doubt that the a new generation of valves will appear that will utilize electronic control with individual valve software for flow and pressure requirements – but due to their relative sophistication, we believe they will cohabits with more traditional schemes as the one presented here.

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