# MODE SWITCHING AND ENERGY RECUPERATION IN OPEN-CIRCUIT PUMP CONTROL

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## ABSTRACT

Today's mobile machines most often contain hydraulic valve controlled drives in an open loop circuit. For the purpose of saving energy, the constant pressure pumps have in the past frequently been replaced by load-sensing pumps and load-sensing valves. However, considering applications where the load is helped by the gravitational force, even these hydraulic systems often suffer from poor efficiency. In this article, a novel pump-controlled hydraulic system is studied where energy recuperation from lowering motions is possible. The pumps are fully displaceable in both directions, working as motors when lowering loads. The amount of recuperated energy is highly dependent on the chosen control strategy, the hydromechanical properties as well as the target application. Furthermore, the article describes how valve design becomes an important feature in an attempt to reach high efficiency and machine operability.

KEYWORDS: pump control, open circuit, energy efficiency, energy recuperation

## 1 INTRODUCTION

In mobile applications load-sensing solutions have significantly reduced energy consumption. However, in applications with unequal drive pressure levels the load sensing systems still result in energy losses, referred to as metering losses. In addition to these losses, most hydraulic systems today do not include the possibility to recuperate the potential energy stored in elevated loads, when these are lowered. Previously, several authors have shown that so called pump control or displacement control is a strong competitor in the energy efficiency debate because of its comparatively few loss elements and versatility in control [1–4]. However, in few of these articles is the utilization of energy recuperation is examined to any great extent, if even mentioned.

The main purpose of this study is to describe how well the open-circuit solution, previously presented by the author [5], can recuperate energy in lowering motions, depending on the chosen control strategy, and how the recuperated energy can be utilized by the application in hand. Furthermore, a theoretical linear analysis and non-linear simulations demonstrate the challenges in an energy efficient load lowering mode, referred to as the differential mode.

In this paper the author has studied a hydraulic system configuration where each actuator/-supply system comprises a variable displacement pump/motor working in an open-circuit together with four separate valves, see Figure 1 . The four valves render a concept versatile in control, as the cylinder chambers can be connected to pump and/or tank as well as be closed at any time. The concept effectively eliminates the metering losses; it has the potential of energy recuperation and enables a four quadrant load actuation.

In either operation quadrant the load speed is controlled directly by the relative pump displacement. The pump controller is capable of switching between displacement control and pressure control in case of excessive pressures. The valve configuration allows loads to be lowered in several different ways. Similar to conventional load-sensing systems, flow-control via the meter out orifice to tank (A-T) is possible. This manner of lowering a load is what we try to avoid in this study, as all potential load energy will be converted to heat in the orifice, thus accounted for as an energy loss. In contrast to this, the author of this study looks upon the advantages of letting lowering flow go through the pump (A-P), controlling the load speed by adjusting the relative pump/motor displacement. This can be achieved either by letting all flow from the piston chamber go to the pump, referred to as "normal lowering"; alternatively, the flow can be divided into one part going to the pump and the other to the piston rod chamber (P-B). This approach is referred to as "differential lowering" and can be seen as a pressure-flow transformation resulting in a decreased amount of pump flow needed to achieve the same piston speed. Since a pump/motor has a limited flow capability at a certain engine speed, the differential mode will make it possible to lower loads faster without increasing the pump size. On the other hand, the pressure level will increase at the same rate, making the load capability lower. An apparent advantage of the differential mode is that the cavitation issue in "normal lowering" is intrinsically solved as flow is taken directly from the piston chamber.

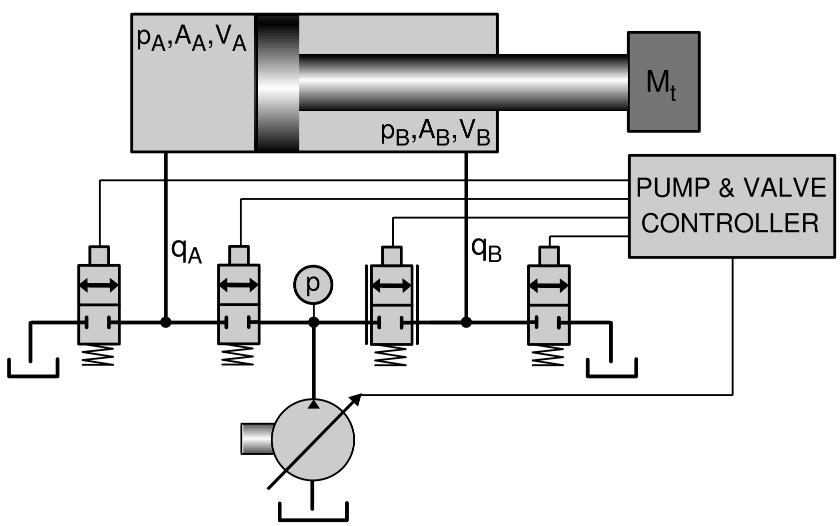
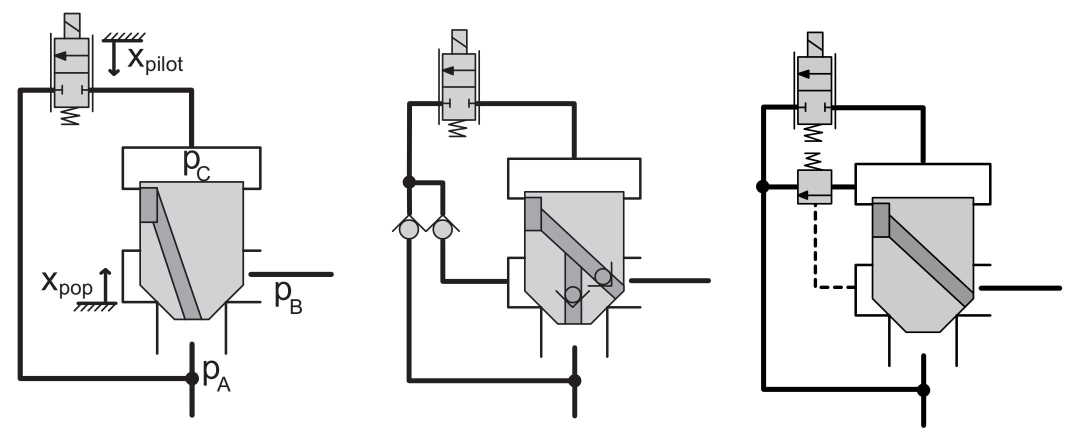



Figure 1: Pump controlled open-circuit solution

### 2.1 Valve Configuration

For good functionality of the open circuit solution the valves that are used for mode switching in the change of operation quadrant, must meet certain requirements. They must have a relatively high bandwidth, provide a "soft closing" in addition to producing low pressure losses at high flows. The valve configuration must also incorporate anti-cavitational capabilities for the load side. To meet these demands, four seat valves of the valvistor type have been chosen for each working cylinder. The valvistor is shown in Figure 2(a). In the main poppet there is an inner orifice that consists of a small rectangular slot with a total area that equals the pilot valve maximum orifice area. When the pilot valve opens, $x_{pilot}$, the pressure, $p_c$, will decrease and the poppet force equilibrium yields a poppet displacement upwards, $x_{pop}$, until the slot orifice area equals the pilot orifice area. Hence, the valvistor is a follower servo. More details on the functionality of the valvistor are presented by Eriksson, B. [6]. In this study the standard valvistor, originally presented by Andersson, B. [7], has been modified to allow flow control in both directions, see Figure 2(b). This modification is necessary to achieve the desired recuperative lowering motions, when flow from the load cylinder is taken back via the pump/motor.




(a) A-type valvistor (b) Bi-directional valvistor (c) Pressure limited valvistor

Figure 2: Different valvistor configurations

There are several reasons why the valvistor has been chosen for this concept. First, the dynamics of a valvistor is generally described as a first order system, which in this case brings out the advantageous property of soft valve closing which is good as the valve is used for fast mode-switching. Furthermore, the valvistor is suitable as a load holding valve because of its inherently great stiffness against pressure disturbances as well as low leakage properties in closed condition. Moreover, the valvistor pilot circuit can in a simple manner be complemented with a pressure relief function, see Figure 2(c). The valvistor yields cost efficient and compact solutions since the pilots are relatively small due to the high flow gain from pilot stage to main stage. The open-circuit solution with valvistors inserted to it is illustrated in Figure 3.

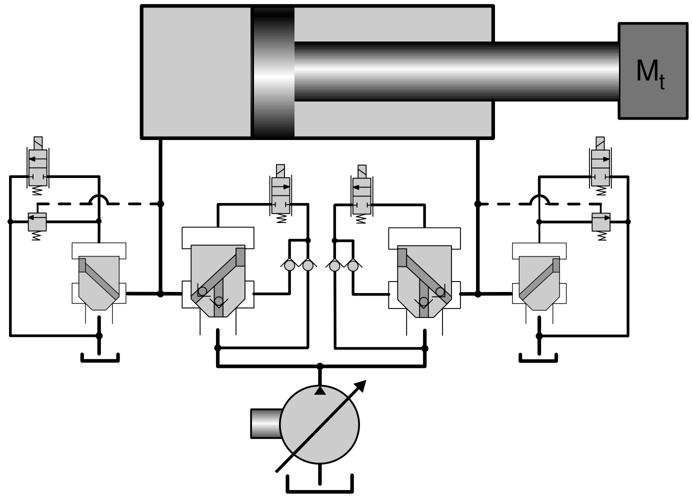



Figure 3: Open-circuit solution with four valvistors implemented

## 2.2 Energy Management

The open-circuit has no hydraulic accumulators or other chargeable devices, which means that all potential energy must be consumed while it is recuperated. The load energy is transferred via the pump/motor shaft, through a power take-out (PTO) to all other energy consuming functions as well as back to the primary power source. Consequently, other energy consumers must be present, otherwise the power loss, which usually takes place in a meter out orifice, has merely been replaced with a power loss in engine friction.

In the following fictive example the open-circuit solution is implemented in a wheel loader. Concerning the hydraulics, the lowering power generated by the load could be used to help all other hydraulic pumps attached to the PTO to power their functions respectively, see Figure 4. For example, bucket lowering and the steering function, or tilting, are often used simultaneously, [8]. Another interesting possibility is to transfer recuperated power to the vehicle drivetrain via the combustion engine. Furthermore, the speed of the cooling fan motor could be increased when recuperated power is available. By doing so the fan could work at lower speed during the rest of the working cycle, consuming less energy.

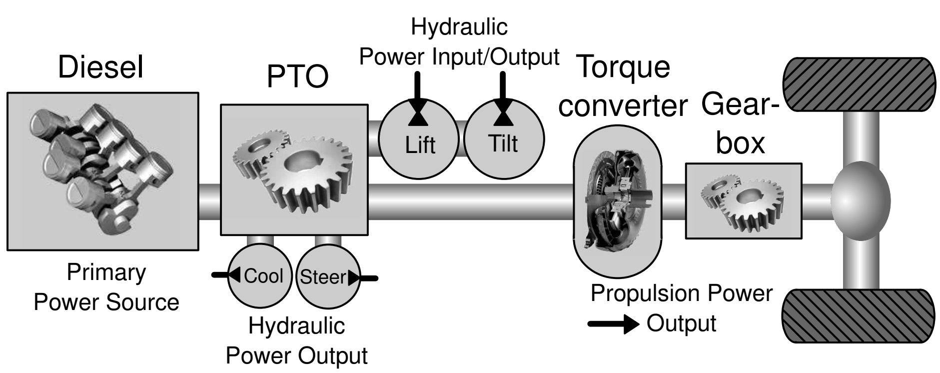
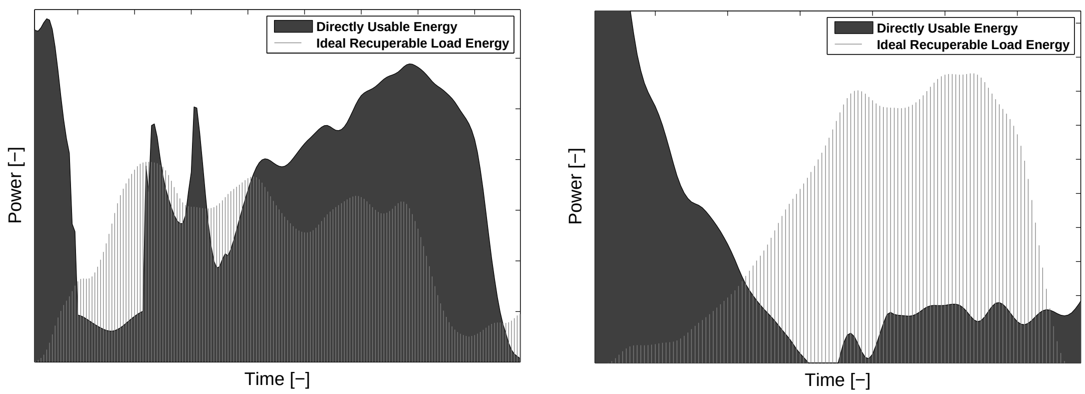



Figure 4: Energy distribution of recuperated energy in a wheel loader application

One must note that the most energy efficient operator driving characteristics for a conventional hydraulic system are not the same as for the system presented here. In the open-circuit solution, the operator can affect the energy efficiency, to a much greater extent, for example by letting the load lowering drive the vehicle backwards when reversing out from a truck. Figure 5 illustrates the potential load energy versus the required energy while lowering a load in a typical loading cycle of a wheel loader with load-sensing hydraulics. The black area in Figures 5(a) and 5(b) represents the sum of the energy required by the steering, propulsion and cooling for the lifting and tilting functions respectively. The grey area is the ideally recuperable energy for each function. While lowering the bucket, cooling is a relatively small energy consumer compared to the energy required for vehicle propulsion backwards as well as for steering. When the bucket is tilted out, there is not much other activity to consume the potential load energy; this operation consequently requires either a change in control strategy or increased engine rpm. However, in using the potentially recuperable energy in the present solution the total hydraulic energy consumption for this loading cycle will be reduced by 5-10%. In a future solution the surplus power can be transferred to a hydraulic or electric buffer, which would save approximately another 5%.



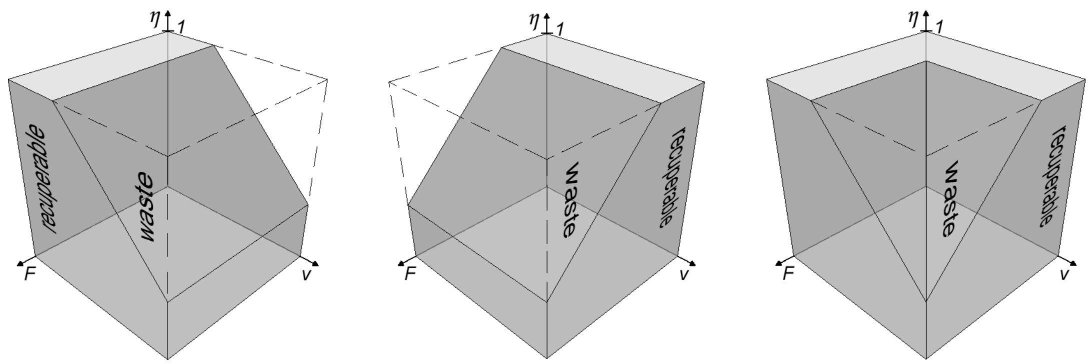

(a) Recuperation potential in lift function (b) Recuperation potential in tilt function

Figure 5: Recuperable energy in a typical loading cycle of a wheel loader

To choose the most energy optimal control strategy for the working hydraulics, the system controller must be capable of estimating the total available power online as well as the total required energy. One way to implement this is to supervise the diesel engine, and compute what power it generates. Also, the total power take out must be estimated online. In practice these actions often require extra transducers installed on certain components. In case of recuperative motions, the control system must also define where recuperated power can be consumed, i.e. by other working hydraulic functions or by the powertrain.

## 3 RECUPERATION EFFICIENCY

Figure 6 illustrates the working range for various lowering modes. The axis pointing upward is the recuperation efficiency, $\eta$. Here a cylinder area ratio, $\kappa$=2/3, is used together with a pump of typical size for a medium sized construction machine. For simplicity the desired maximum lowering speed is set to three times the maximum lifting speed. As seen in Figure 6(a) full energy recuperation can only be achieved up to a third of the desired lowering speed because of the limited pump capacity. In Figure 6(b) the cylinder area ratio makes it possible to lower at the desired speed, but because of the pressure increase, only a third of the desired loading capability can be obtained. The figures show that the overall potential in energy recuperation is the same in both cases, but depending on the most common point of operation one of the solutions is better than the other. In applications where lowering with high speeds and low loads is the most common, the differential mode is the most advantageous. When it is more common to lower heavy loads at low speeds, the normal mode is to be preferred. For example, if the application is a wheel loader loading gravel, the bucket is usually lowered empty at high speed; hence would the differential mode be more appropriate. However, where switching between these two modes is possible, higher recuperation efficiency could be obtained over a greater working range, illustrated in Figure 6(c).




(a) Normal lowering mode (b) Differential lowering mode (c) Differential/normal lowering mode

Figure 6: Ideal efficiency regarding maximum system pressure and pump flow

In order to realize the normal lowering over the whole working range one must control flow to tank over an orifice to reach speeds exceeding the maximum pump capacity, thus decreasing the recuperation efficiency. In the differential case, one must instead restrict "the degree of" differential mode at higher loads to avoid reaching the maximum pressure level. In practice, this can be achieved by using a valve, that senses the pressure level in the piston chamber, which for a given maximum pressure level, starts closing the connection between the cylinder chambers (Figure 7(a) in Section 3.1), converting all power related to the pressure exceeding the pre-defined maximum level to heat. If this valve closes completely, normal mode is achieved, and flow to the piston-rod chamber must instead be taken from tank (T-B). See Section 3.1-3.3 for further analysis in this subject. However, the most obvious difference between these two solutions is that the differential mode requires pressure control of an orifice while the normal mode requires flow control over an orifice. The most energy efficient strategy is determined according to the application and under what working conditions the machine usually operates. Given that point of operation, one can decide which solutions is the most suitable. In case a mode switching solution is selected there are of course important changes in system properties to consider. For example in case of going from normal mode to differential mode not only the pressure level will change but also the dynamic load properties, such as hydraulic hydraulic damping and eigenfrequency.

## 3.1 Static Calculations

To demonstrate the basic functionality of the system, operating in the differential mode, a static model was constructed on the basis of the system shown in Figure 7(a). Here the control valve, which is a normally open pressure limiter, is mounted directly in the main circuit to simplify the calculations. The flow through the valve is given by the orifice equation, Eq.1

$$q_B = K_s \cdot (x_{max} - x_v) \cdot \sqrt{p_A - p_B} \tag{1}$$

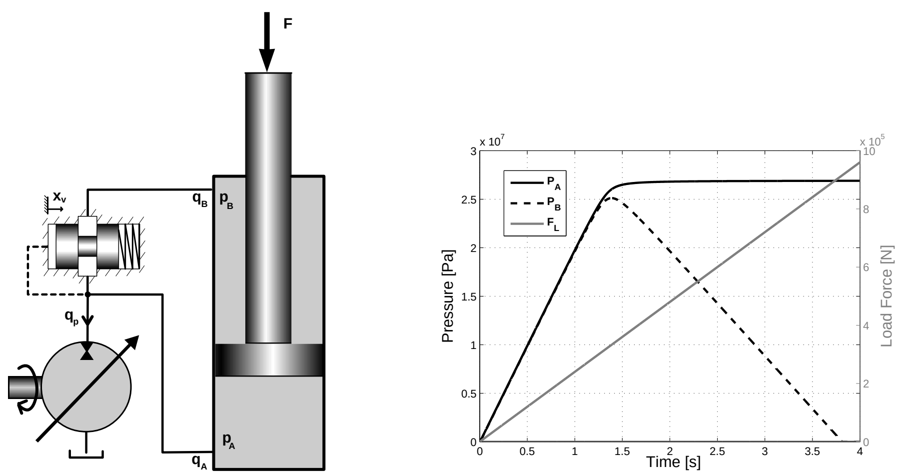
where $K_s$ is the orifice coefficient and $x_v$ is the valve closure. The static force equilibrium is given by Eq. 3

$$-p_A \cdot A_{red} + k_s \cdot l_s + k_s \cdot x_v = 0 \tag{2}$$

In this model the fluid compressibility and the valve dynamics have been ignored. Neither has cavitational effects been taken into account. This yields a piston speed, $v_p$, directly proportional to the pump flow, independent of the valve closure.

$$v_p = \frac{q_p}{A_A} \cdot \frac{1}{1-\kappa} \tag{3}$$

The input variables to the calculation are pump flow, $q_p$, and load force, $F_L$. In Figure 7(b) the pump flow is kept constant and the load force is ramped up from zero (right axis). When the pressure level (left axis) has reached the pre-defined cracking pressure (25 MPa in the figure), the pressure in the piston chamber, $p_A$, is kept at a fairly constant level and the other pressure, $p_B$, is closing up to zero as the load force increases further.




(a) Simplified system, base for static calculations

(b) Static pressure response to applied load pressure

Figure 7: Static behavior of system pressure limiter

## 3.2 Linearized Model

The static calculations in Section 3.1 are good to describe the conceptual idea of the pressure limiter, but to get a better understanding of the dynamic system behavior a linearized model is derived.

A realistic linear model can be concieved from the physical equations described in Section 3.1 along with equations for the load dynamics. After linearization and Laplace transformation of these equations, Eq. 4–9 are obtained.

$$\Delta Q_A = \Delta Q_B \tag{4}$$

$$\Delta Q_B = K_q \cdot \Delta X_v + K_c \cdot (\Delta P_A - \Delta P_B) \tag{5}$$

$$\Delta P_A = (-\Delta Q_A + s \cdot \Delta X_p \cdot A_A) \cdot \frac{\beta_e}{V_A \cdot s} \tag{6}$$

$$\Delta P_B = (\Delta Q_B - s \cdot \Delta X_p \cdot A_B) \cdot \frac{\beta_e}{V_B \cdot s} \tag{7}$$

$$\Delta X_p = \frac{\Delta P_B \cdot A_B - \Delta P_A \cdot A_A + \Delta F_L}{M_t \cdot s^2 + B_p \cdot s} \tag{8}$$

To further consider the valve dynamics, the pressure limiting valve is looked upon according to Figure 8. After linearization and Laplace transformation the valve closure $\Delta Xv$ is given by Eq.9.

$$\Delta Xv = -\Delta P_A \cdot \frac{C}{1 + T_r \cdot s} \tag{9}$$

, where the constant $C$ is related to the spring coefficient, $k_s$, and pressurized area, $A_{red}$, within the valve pressure-sense port.

$$C = \frac{A_{sen}}{k_s} \tag{10}$$

and $T_r$ is a time constant, determined by properties related to the spring as well as the sense-channel volume and orifice.

$$T_r = \frac{k_s V_{sen} + \beta_e A_{sen}^2}{k_s \beta_e K_{c,sen}} \tag{11}$$

Interesting for further analysis is the transfer function from external load disturbance, $\Delta F_L$, to the change of pressure in the piston chamber, $\Delta p_A$. The algebraic solution to this closed loop circuit, computed in a typical operating condition for a construction machine, yields a fourth order transfer function.

$$G_{sys} = \frac{\Delta P_A}{\Delta F_L} = \frac{(s+\omega_1)(s+\omega_2)}{(s+\omega_3)(s+\omega_4)(\frac{s^2}{\omega_5^2} + \frac{2\delta_h s}{\omega_5} + 1)} \tag{12}$$

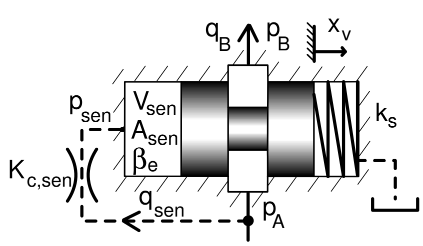



Figure 8: Assumed valve functionality

The C-value is determined by what pressure level, $p_{max}$, beyond the cracking pressure level, $p_{crack}$, is acceptable before the pressure limiter should be completely closed.

$$x_{max} = (p_{max} - p_{crack}) \cdot \frac{A_{red}}{k_s} \tag{13}$$

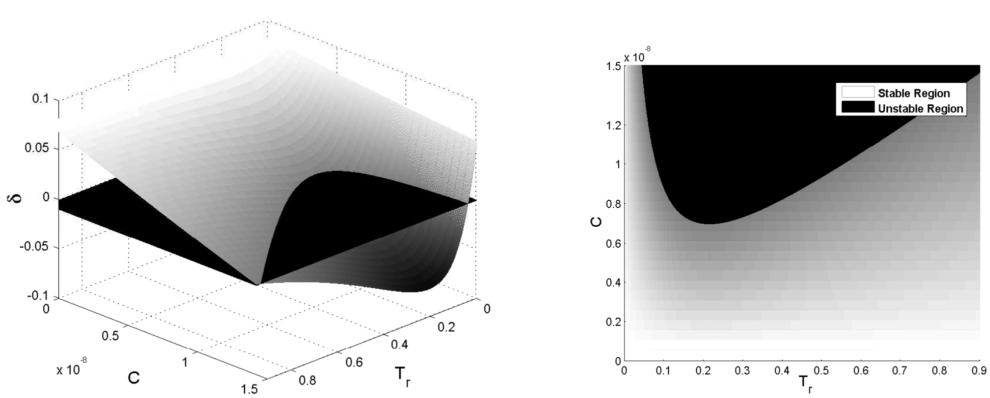
which along with Eq10 yields the C-value

$$C = \frac{x_{max}}{(p_{max} - p_{crack})} \tag{14}$$

Furthermore, there are physical restrictions on the valve properties, such as the choice of a realistic spring coefficient as well as size of the pressurized area in the sense port, see Eq. 10. Also the minimum diameter of the valve orifice is critical as cavitation on the piston rod chamber must not occur at full lowering speed. For example, in a 350 bar system the cracking pressure is set to, say 250 bar, then at 350 bar the valve should be completely closed. In this case $C = 1 \cdot 10^{-9}$ is a suitable value in order to get an appropriate valve size and closure.

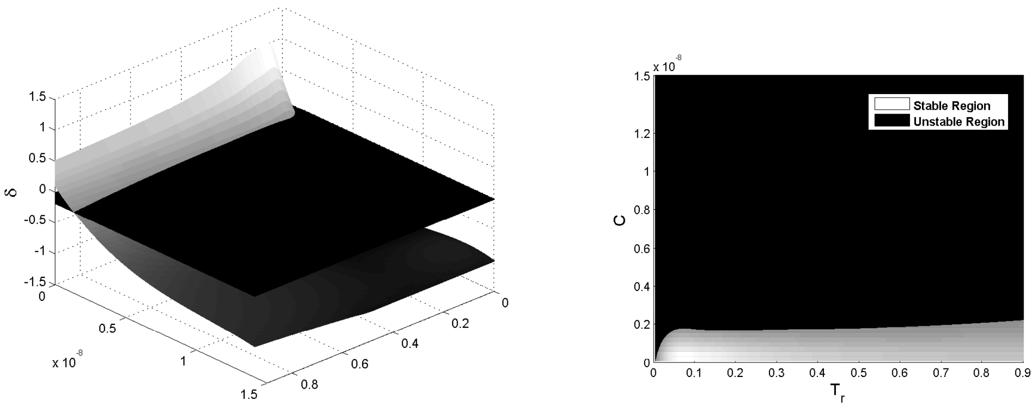
If the cracking pressure is set closer to the maximum this yields a higher C-value. Looking at the poles of the transfer function in Eq. 12, an increase in C-value eventually leads to system instability. How the other system parameters, such as cylinder area ratio, working volumes and inertia load affects the limit for instability is rather complex. However, for a given application, these properties are known, only leaving out the properties of the pressure limiting valve as design parameters. Except for the C-value, the adjustable parameters are; the valve time constant, $T_r$, described by Eq. 11 and the geometric characteristics of the valve, described within $K_q$ and $K_c$ in Eq 5. In practice, an increased value of $T_r$ corresponds to a slower valve response to pressure increase. This would intuitively mitigate the risk of instability as the dynamic pressure build-up, will not be as remarkable as the valve will react slowly, adopting its closure only to static changes in pressure. In Figure 9 the system damping is shown for a set of realistic $C$ and $T_r$ values, linearized close to the valve cracking pressure.

The black plane illustrates where the system damping is zero, thus marginally stable. Seen in Figure 9(b) high C-values can be chosen either by using a very fast valve or quite a slow valve. Note that without the pressure limiting valve, the system damping is zero as




(a) Hydraulic damping in respect to $T_r$ and $C$ (b) Stable region in respect to $T_r$ and $C$

Figure 9: System damping and stability when linearizing at valve cracking pressure




(a) Hydraulic damping in respect to $T_r$ and $C$ (b) Stable region in respect to $T_r$ and $C$

Figure 10: System damping and stability region when linearizing close to zero

this is an ideal pump controlled system. This stable region will be greater in case further system damping is introduced, such as friction and leakage. However, as $G_{sys}$ in Eq.12 will change with a different point of linearization, the stability region will also change. Figure 10(b) illustrates the stability region when the valve opening is chosen closer to zero. The stable region is now substantially smaller and will become even smaller as the valve closes further. Here, it still helps to use a higher value in $T_r$ but eventually no realistic value is good enough to maintain stability.

### 3.4 Non-linear Model

To proceed with the analysis and to get a better understanding of the instability issue, described in Section 3.3, the system was modelled in Modelica. Complementing the static system of equations, in Section 3.1, with the missing dynamic equations for the load and the valve, a dynamic and non-linear model is conceived. Moreover, the main orifice and the sense-channel orifice are both modelled as turbulent restrictors, seen in Figure 11.

The load case parameters are the same as for the linearized model in previous section. The $C$-value is still described by Eq 10, but $T_r$ is now determined by the properties of the sense-channel turbulent restrictor and will hence vary with valve closure. However,

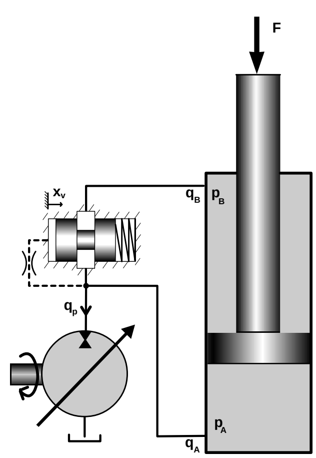
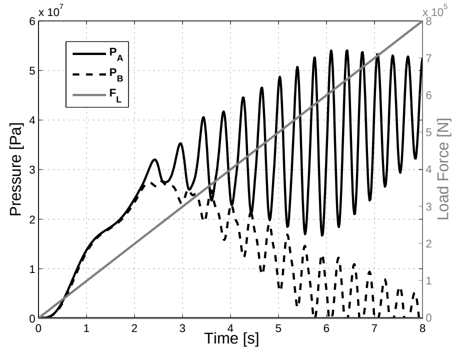
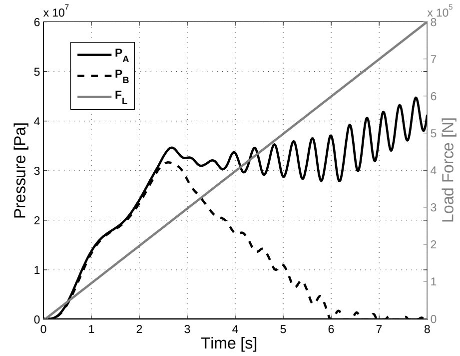



Figure 11: Base for non-linear model




(a) Dynamic pressure response to applied load pressure, low $T_r$-value




(b) Dynamic pressure response to applied load pressure, high $T_r$-value

Figure 12: Non-linear, dynamic pressure response of system pressure limiter

making the restrictor area smaller will of course still increase the $Tr$-value, given the same pressure level. Furthermore, a higher $T_r$ value will dynamically increase the cracking pressure which is statically given only by the $C$-value. In Figure 12(a) the instability issue is obvious. In this figure, a very fast valve has been used, low $T_r$-value. In Figure 12(b) a higher $T_r$-value is used, thus a more stable behavior is shown even though instability is a fact as the valve opening approaches zero.

## 4 FUTURE WORK

The open-circuit solution will be implemented in a full scale wheel loader, where it will be evaluated in respect to energy efficiency and operability. Different ways of recuperating energy from the lowering motions will be evaluated, especially the strengths and weaknesses of the differential mode. The hydraulic solution of the differential mode presented in this article, will be further investigated. This solution and its instability issues are familiar from previous investigations on the dynamic properties of the over-center valve, by Persson, T. [9]. His work will be an inspiration for further research. An alternative solution to the differential mode is to accommodate electro-hydraulic pressure control of an orifice, thus making the control strategy more flexible. Concerning implementation, the pressure limiting valve will be implemented in the valvistor valve configuration.

## 5 CONCLUSIONS

The chosen valve configuration for the open-circuit carries out a flexible solution that allows the working hydraulics to lift and lower loads in several different modes of operation. In a wheel loader application the energy recuperated from load lowering can in many cases be used immediately by for instance vehicle propulsion and/or other hydraulic functions. Furthermore, the advantages with normal lowering mode versus differential lowering mode have been investigated. Which mode is the most suitable depends on what the operator is trying to do. To achieve an energy efficient load lowering the choice of mode depends on the requested speed, the magnitude of the load as well as pump/motor efficiency at that given operating condition. Moreover the possibility to switch between normal mode and differential mode is an interesting aspect regarding increased efficiency. In this study a hydraulic solution to the differential mode is suggested and analyzed. The suggested pressure limiting valve demonstrates an unstable behavior when its valve closure approached zero. This behavior is explained by the dynamic pressure build-up, present in the up-stream volume due to the increased valve closure, amplifying the load pressure which further closes the valve.

## 6 LIST OF NOTATIONS

| Quantity | Description | Unity |
|---|---|---|
| $A_A$, $A_B$ | Effective area, piston chamber, piston rod chamber | $m^2$ |
| $A_{red}$ | Pressurized area in pressure limiter sense port | $m^2$ |
| $B_p$ | Viscous cylinder friction coefficient | $\frac{Ns}{m^2}$ |
| $C$ | Valve closure coefficient, $\frac{A_{red}}{k_s}$ | $\frac{m^3}{N}$ |
| $F_L$ | Load disturbance force | $N$ |
| $G_{sys}$ | Transfer function from $\Delta F_L$ to $\Delta P_A$ | $\frac{1}{m^2}$ |
| $\beta_e$ | Bulk modulus | $Pa$ |
| $\kappa$ | Cylinder area ratio | – |
| $\delta_h$ | Hydraulic damping at the hydraulic resonance frequency | – |
| $\Delta F_L$ | Linearized load disturbance force | $N$ |
| $\Delta P_A$, $\Delta P_B$ | Linearized pressure acting on $A_A$, $A_B$ | $Pa$ |
| $\Delta Q_A$, $\Delta Q_B$ | Linearized flow from/to the cylinder chambers | $\frac{m^3}{s}$ |
| $\Delta X_p$ | Linearized piston displacement (stroke) | $m$ |
| $\Delta X_v$ | Linearized valve displacement | $m$ |
| $\omega_i$ | Resonance frequency for the i:th pole of $G_{sys}$ | $\frac{rad}{s}$ |
| $K_c$ | Flow-pressure coefficient | $\frac{m^3}{sPa}$ |
| $k_s$ | Spring coefficient | $\frac{N}{m}$ |
| $K_s$ | Valve coefficient, $C_q w\sqrt{\frac{2}{\rho}}$ | $\frac{s}{m}\sqrt{N}$ |
| $K_q$ | Flow gain coefficient | $\frac{m^2}{s}$ |
| $l_s$ | Spring pre-contraction | $m$ |
| $M_t$ | Inertia mass load | $kg$ |
| $p_A$, $p_B$ | Pressure acting on $A_A$, $A_B$ | $Pa$ |
| $p_c$ | Pressure in volume between valvistor poppet and pilot | $Pa$ |
| $p_{crack}$ | Pressure when pressure limiting valve starts to close | $Pa$ |
| $p_{max}$ | Maximum allowable system pressure | $Pa$ |
| $q_A$, $q_B$ | Flow from/to the cylinder chambers | $\frac{m^3}{s}$ |
| $q_p$ | Flow to/from pump | $\frac{m^3}{s}$ |
| $s$ | Laplace operator | – |
| $T_r$ | Time constant of pressure limiting valve | $s$ |
| $V_A$, $V_B$ | Volume of piston chamber, piston rod chamber | $m^3$ |
| $v_p$ | Piston velocity | $\frac{m}{s}$ |
| $x_v$ | Valve displacement | $m$ |
| $x_{max}$ | Maximum valve displacement | $m$ |
| $x_{pilot}$ | Valvistor pilot valve displacement | $m$ |
| $x_{pop}$ | Valvistor main poppet displacement | $m$ |

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