# Improving the efficiency of valve-controlled systems by using multi-chamber actuators

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## Abstract

This paper outlines how multi-chamber actuators can improve the efficiency of valve-controlled systems. Resistive control is a major source of energy losses in valve-controlled systems that share the same pump to drive multiple loads. In the proposed concept, by selecting different chambers, the load on the multi-chamber actuator can be transformed into different pressure and flow rate levels, allowing the adaptation of its load to the loads on other actuators. This can lead to a reduction of resistive control energy losses that occur between pump and actuators when driven simultaneously. Such systems are seen as an intermediate solution between resistive conventional hydraulics and throttle-less digital hydraulics. As a case study to highlight the possible efficiency improvement, a concept of a load sensing system with a conventional and a multi-chamber actuator is analysed. To determine its efficiency, the equations that describe its static behaviour are presented. Evaluating them for a set of load forces and speeds demonstrates how the load transformation occurs and how it can improve efficiency.

**Keywords**: Digital fluid power, multi-chamber actuators, throttling losses

## 1 Introduction

The study presented in this paper is motivated by the fact that the introduction of throttle-less digital hydraulics into a valve-controlled hydraulic system architecture might result in increased system efficiency without requiring a significant redesign. Although the full potential of throttle-less digital hydraulics to increase efficiency might not be achieved, it could still result in a considerable reduction of the system energy losses.

So far, most of the concepts for linear actuation in hydraulics have been designed with either a conventional resistive approach in mind or with a digital hydraulics approach in mind. One of the most common approaches of digital hydraulics for linear actuation was presented by Linjama *et al.* in [1], where the concept of a secondary controlled multi-chamber actuator is proposed. By using a parallel configuration of on/off valves, different connections between pressure sources and actuator chambers are obtained, resulting in discrete force levels. When the authors compare this concept to a conventional load sensing system, a 60% reduction in power losses is achieved. The authors also mention that because the typical load sensing system has several actuators, the supply pressure is optimized for one actuator only, resulting in significant throttling losses on the valves controlling other actuators. This is not the case for the secondary controlled actuator, since the loads are decoupled.

In [2] the energy efficiency of a three-chamber actuator controlled by on/off valves is evaluated. At each control edge there is a set of on/off valves with different flow areas. At each control mode (combination of actuator areas to pressures sources) the pressure of one of the chambers is controlled resistively through the selection of different valves, resulting in a different total flow area. The control objective is to find the mode that can drive the load but also reduce the pressure drop on the controlling valves, therefore reducing the losses due to the resistive control.

A similar concept as the one present in [1] is presented in [3] for the control of an excavator arm. The main difference is the additional pressure source, which results in a higher number of available force levels to be chosen. The same concept is investigated in Belan *et al.* [4], but applied to the control of flight control surfaces of aircrafts. In [4] the authors presented a detailed investigation of how the areas of the actuator chambers can be determined together with efficiency analysis. For the same load cases, the digital hydraulic actuator would result in an 80% reduction in energy losses in comparison to traditional servo-proportional control.

It is observed that the switching nature of these systems imposes a major challenge on the development of controllers and is also a significant source of losses due to the compressibility of the fluid. As mentioned in [1] compressibility losses are caused by the fact that the energy stored in hydraulic capacitance is lost when the chambers switch between pressure levels. Other researchers proposed alternative architectures that contain means of throttling control to improve mainly the control performance in detriment of a reduction in system efficiency.

In [5] the authors mention about the limited finite number of discrete force levels that can be generated by pure digital hydraulic configurations with parallel valves. To improve the force resolution, aiming to achieve an accurate control of the actuator, the authors propose that a proportional valve should be connected to one of the chambers. Results show that controllability can be improved while still maintaining high energy efficiency.

In [6] the authors present a model predictive control strategy for a four-chamber actuator connected to a common pressure rail through proportional valves, a Variable Displacement Linear Actuator (VDLA). It is similar to the concept presented in [1] except for the use of proportional valves which allow throttling control between the different force levels, resulting in smoother control while still maintaining high efficiency. In a simulation study presented in [7] the effectiveness of the VDLA is also shown at a system level for a hydromechanical hybrid motion system for a wheel loader. In [8] results show a 34-50% fuel efficiency reduction for an excavator with a hydraulic system architecture based on the VDLA. Although the high efficiency results motivate a change to the architecture of hydraulic systems in mobile machines, it is noted that a significant redesign is required.

Aiming to be an intermediate solution between valve-controlled architectures and architectures based on digital control, the goal of this paper is to show that the use of multi-chamber actuators in a valve-controlled system can result in an increase in system efficiency. The main purpose of the multi-chamber actuator is to adapt its required pressure and flow rate to reduce throttling losses that arise when multiple loads are driven simultaneously. During the literature review, the authors did not find any similar approach to the combination of valve-controlled and digital hydraulics. In this sense, another contribution of this paper to the research area is a different way of using the principles of digital hydraulics.

This paper is organized as follows: Section 2 presents a description of the proposed combination of multi-chamber actuator and valve-controlled systems; Section 3 provides a discussion on how a controller could be designed for this system; In Section 4 the concept selected for the case study is described in more detail. Section 5 presents the equations used for the efficiency analysis; Section 6 presents the analysis and efficiency results; Section 7 presents a discussion about the main points extracted from the results; and Section 8 contains the conclusions.

## 2 Concept Description

Figure 1a is a diagram of a secondary controlled actuation system, where the actuators are controlled by the switching of the on/off valves to establish different connections between pressure sources and actuator chambers. The combination of pressures and chamber areas results in different force levels, which can be used to perform the control action against the load. Since the valves operate either closed or fully open, the throttling losses are significantly reduced. It can be assumed that the pressures of the pressure sources remain constant and, therefore, such a system does not suffer from load interference, which means that the control of one actuator does not interfere with the control of the other actuators.

Figure 1b is a diagram of a valve-controlled load sensing actuation system with a single pump driving two loads. It is known that a major efficiency drawback in such systems arises from the throttling losses that occur between pump output and the actuators. Since the pump pressure is governed by the highest load pressure, the pressure losses on the valves controlling the other actuator are significant. In order to reduce the losses due to throttling, this paper proposes to change a conventional actuator for a multi-chamber actuator.

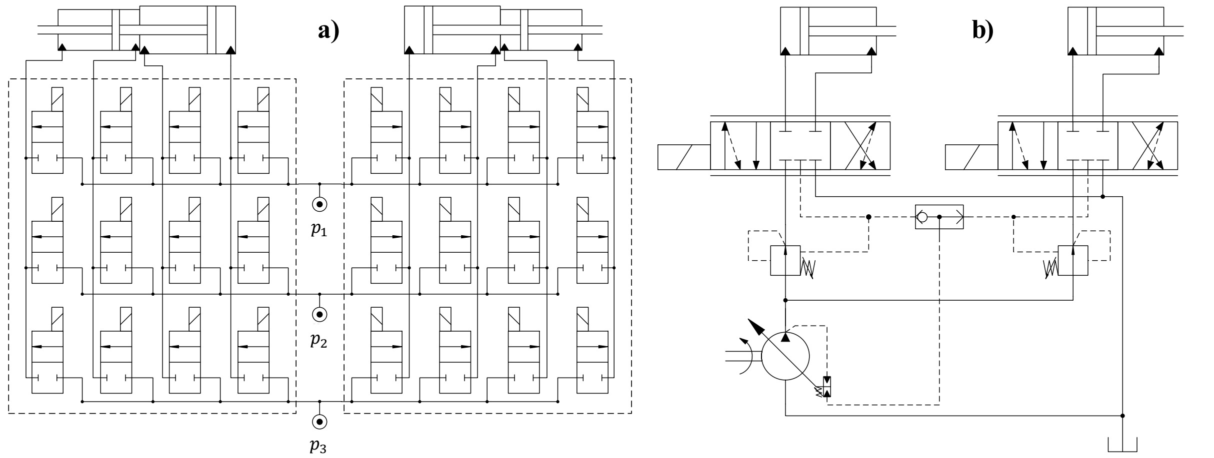


Figure 1 - a) Secondary controlled actuation system; b) Valve-controlled load sensing system.

Figure 2a is an example of the application of a multi-chamber actuator to a valve-controlled open-centre architecture. Figure 2b is an example of the application of a multi-chamber actuator to a valve-controlled load sensing architecture.

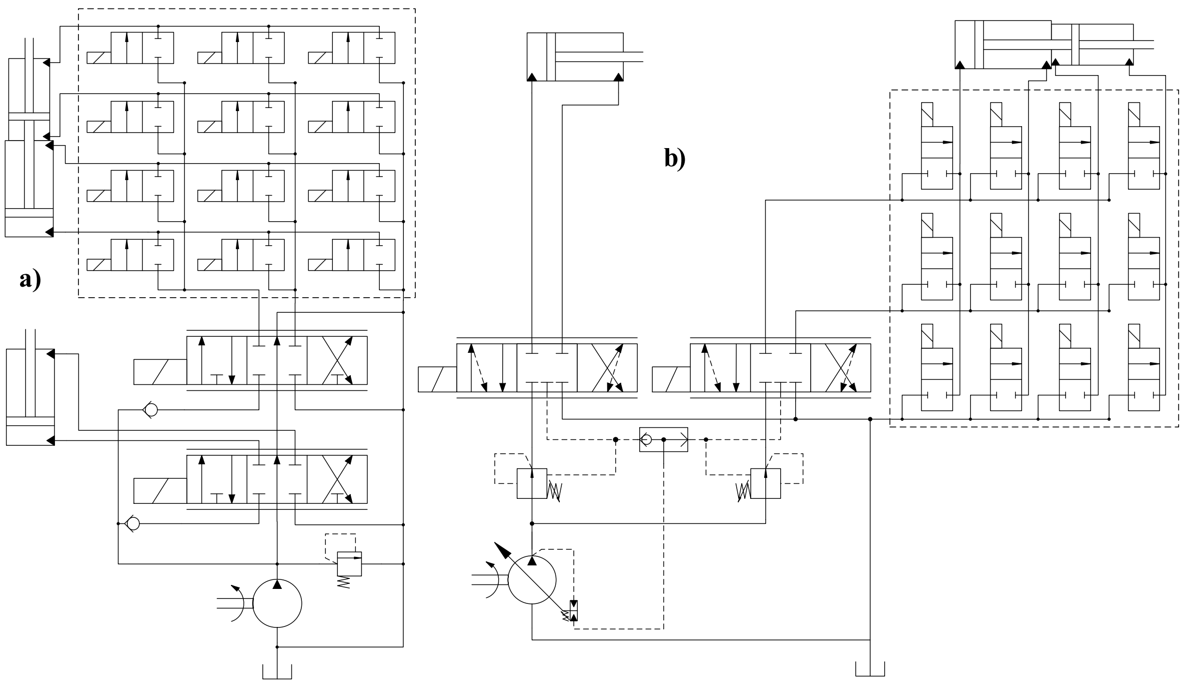


*Figure 2 - a) Open-centre actuation system; b) Load sensing closed-centre actuation system.*

The load on the multi-chamber actuator can be transformed into different pressure and flow rate levels through the combination of different chambers area. By selecting a combination that results in a match of load pressures between the two actuators, the throttling losses on the control valves can be reduced.

One interesting aspect of such concepts is that they do not require significant changes to the existing valve-controlled architectures. The third line that is connected between the on/off valves and the reservoir is seen as necessary to prevent the flow for expanding chambers that are not connected to the pressure supply from going through the proportional valves.

## 3 System Control

With the large number of possibilities to connect the proportional valve to the actuator comes the challenge of which combination to select for the different loads on the actuators and how to perform the switching between them. This is not the main topic of discussion of this paper, but some comments are made on how to select a combination to be implemented.

In a usual application of a multi-chamber actuator in a secondary controlled system, the main objective is for the actuator to achieve the desired control goal, and that would be position, speed or force. For the current concept, the objective is not just to achieve the control goal, but also to select a combination that minimizes system energy losses. In this sense, the selection of a combination is also driven by the current force and speed states on other loads. The proportional valves are still responsible for controlling position, speed or force.

It is not a trivial task to map all possible system states to the best combination. It could be found through the development and implementation of an intelligent and optimization-based controller. This would be treated as a multi-objective optimization, where a trade-off between control accuracy of the multi-chamber actuator and system efficiency should be made. As part of the study carried out by the research group, a controller based on reinforcement learning is being studied and will be presented in future publications. In this paper the focus is on the advantage of having combinations to select from, and so establishing which combination is the best for a certain system state is a topic for further research, as mentioned.

It is also important to add that each combination represents a different actuator, so the dynamic characteristic of the system is also changed. This will impact on the controller of the proportional valve, since it is controlling different actuators. Such changes should be considered when designing the controller.

## 4 Selected Concept for the Case Study

The architecture proposed in fig. 2b was selected to perform an analysis of its efficiency and describe in more detail how it could be increased. A schematic diagram of the selected architecture with the main variables used in the analysis is presented in fig. 3.

The configuration of multi-chamber actuator and on/off valves is the same as for the concept presented in [3]. However, the difference lies in the power supply which, for the current concept, is a load sensing architecture rather than constant pressure sources. Therefore, for this current study no sizing of components is carried out. This does not affect the analysis presented, since the principle of how the efficiency can be improved can also be demonstrated.

The connection between the multi-chamber actuator and reservoir is considered to be necessary to avoid a high imbalance of flow rates in the ports of the proportional valve and also for when a flow from the reservoir is needed to fill the expanding chambers that are not connected to the proportional valve. Although this is not considered in this paper, it would be necessary to have a pressurized reservoir.

The original system as presented in [3] was designed to be operated as a secondary controlled system, which would result in 81 force steps. For the current application not all combinations that would be available for a secondary-controlled actuator are feasible. Most of the possible combinations are excluded due to at least one of the reasons listed in the sequence:

- **No flow through the proportional valve**

Certain combinations would result in no flow at one of the ports of the proportional valve and, therefore, they have been removed. Table 1 shows an example of such a situation, where no chamber is connected to the B port of the proportional valve. Depending on the position of the proportional valve, either the supply or the return port would experience a no-flow situation.

*Table 1 - Example of infeasible connection due to no flow at one of the proportional valve ports.*

| Ports | A | B | T |
|---|---|---|---|
| *Connected chambers* | AC | - | BD |

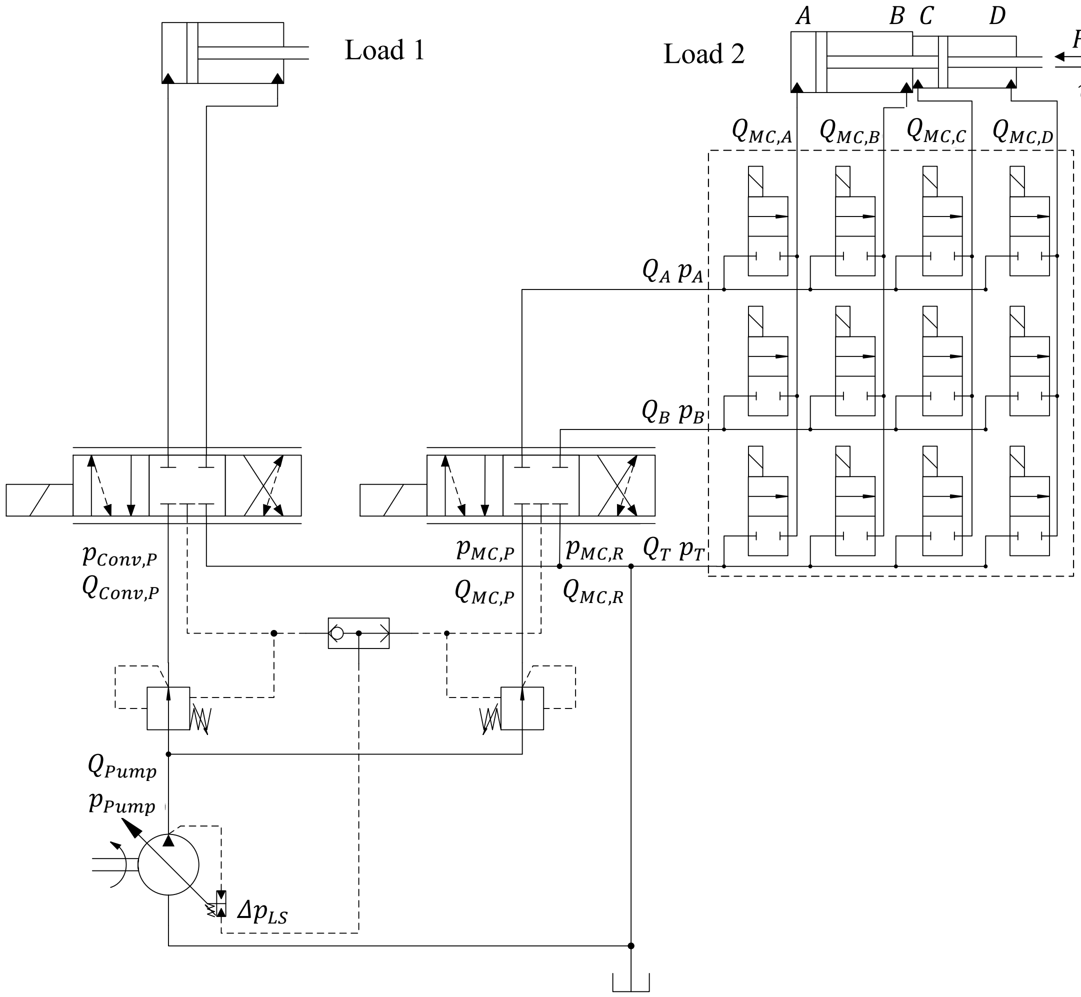


*Figure 3 - Selected architecture for case study.*

- **Flow with same direction in both ports of the proportional valve**

Certain combinations would result in flow in the same direction in both ports of the proportional valve, and so they have also been removed. In simple terms it means, for example, that it would result in suction of flow from reservoir through the proportional valve. One example of such a situation is presented in tab. 2.

*Table 2 - Example of infeasible connection due to same flow direction in the ports of the proportional valve.*

| Ports | A | B | T |
|---|---|---|---|
| *Connected chambers* | A | C | BD |

Chambers A and C are always expanding or contracting at the same time. For the given example, if the pump was supplying flow rate to chamber A then chamber C would require flow from the reservoir through port B. It is the areas of the actuator that determine which combinations are excluded for this reason, which motivates an additional study on how the selection of areas would impact the number of feasible combinations.

- **Redundant combinations when defining the actuator movement direction**

The manifold of on/off valves allows each of its input ports (A/B/T) to be connected to any of the chambers, while the proportional valve can switch direction. This results in two possibilities of operating the proportional valve and digital manifold to achieve the same actuator motion.

For example, to make the actuator retract or extend, one possibility would be to maintain the proportional valve in one side, e.g. P→A and B→T and let the on/off valves change combination to achieve the different movement direction. This situation is presented in tab. 3.

*Table 3 - Example of redundant combination when operating the proportional valve in one direction.*

| Ports | A | B | T | Proportional valve | Actuator movement |
|---|---|---|---|---|---|
| *Connected chambers 1* | A | B | CD | P→A and B→T (+ position) | Extend |
| *Connected chambers 2* | B | A | CD | P→A and B→T (+ position) | Retract |

An advantage of this approach is that a 4/2 instead of a 4/3 proportional valve could be used. However, another possibility is to work with a smaller number of combinations and allow the proportional valve to operate to both sides. It would also result in retraction or extension of the actuator, but for the same combination of the on/off valves. This second case would involve a number of combinations that are redundant and, therefore, can also be excluded. One example of such a situation is shown in tab. 4.

*Table 4 - Example of redundant combination when operating the proportional valve in both directions.*

| Port | A | B | T | Proportional valve | Actuator movement |
|---|---|---|---|---|---|
| *Connected chambers 1* | A | B | CD | P→A and B→T (+ position) | Extend |
| | A | B | CD | P→B and A→T (- Position) | Retract |
| *Connected chambers 2* | B | A | CD | P→B and A→T (- Position) | Extend |
| | B | A | CD | P→A and B→T (+ position) | Retract |

For this example, combinations 1 and 2 are redundant, since it is the proportional valve that changes the actuator's direction of movement. This approach seems to be a more natural one to adopt because fewer combinations are needed, which facilitates the process of selection of combinations when designing a controller. Another positive point is that the proportional valve does not need to be changed or its control structure significantly modified.

In terms of selecting which of the redundant combinations to use, the decision was taken to select the ones that, for a given load in one of the four quadrants of force and speed direction of the actuator, make the proportional valve always operate to the same side for all combinations. This avoids a need for the proportional valve to change the operating side if the load changes in magnitude but not the operating quadrant. In the end, sixteen combinations are thought to be feasible for this system configuration. They are presented in tab. 5.

*Table 5 – Selected combinations of chambers.*

| Combination | A | B | T | Combination | A | B | T |
|---|---|---|---|---|---|---|---|
| *1* | A | B | CD | *9* | ABC | D | - |
| *2* | A | BD | C | *10* | ACD | B | - |
| *3* | A | D | BC | *11* | BC | D | A |
| *4* | AB | D | C | *12* | C | B | AD |
| *5* | AC | B | D | *13* | C | BD | A |
| *6* | AC | BD | - | *14* | C | D | AB |
| *7* | AC | D | B | *15* | CD | B | A |
| *8* | AD | B | C | *16* | - | - | - |

As will be shown in the results section, even though all these combinations are available to be chosen, not all of them are feasible for all load conditions. For certain load speed and force, some would result in too high pump pressure and/or might result in a requested flow rate above the maximum capacity of the pump. In this way, such considerations would be required when making the selection of a combination by a controller.

## 5 System Model

For the selected system, the pressure losses on the on/off valves are considered negligible. The subscript *MC* refers to multi-chamber actuator and *Conv* to conventional actuator. The flow rate at the port of each chamber of the multi-chamber actuator is given by

$$\begin{bmatrix} Q_{MC,A} \\ Q_{MC,B} \\ Q_{MC,C} \\ Q_{MC,D} \end{bmatrix} = v \begin{bmatrix} A_{MC,A} \\ -A_{MC,B} \\ A_{MC,C} \\ -A_{MC,D} \end{bmatrix}, \tag{1}$$

where $v$ is the actuator speed and the negative sign tells that it is an contracting chamber for a positive speed. The flow rates at the input ports of the digital valve manifold are

$$\begin{bmatrix} Q_A \\ Q_B \\ Q_T \end{bmatrix} = \begin{bmatrix} D_{AA} & D_{AB} & D_{AC} & D_{AD} \\ D_{BA} & D_{BB} & D_{BC} & D_{BD} \\ D_{TA} & D_{TB} & D_{TC} & D_{TD} \end{bmatrix} \begin{bmatrix} Q_{MC,A} \\ Q_{MC,B} \\ Q_{MC,C} \\ Q_{MC,D} \end{bmatrix}, \tag{2}$$

where $D_{ij}$ are binary variables that determine whether the valve that connects input port $i$ to the $j$ chamber of the multi-chamber actuator is closed or open. The sum of the chambers area connected to A, B and T are given by

$$\begin{bmatrix} A_A \\ A_B \\ A_T \end{bmatrix} = \left\| \begin{bmatrix} D_{AA} & D_{AB} & D_{AC} & D_{AD} \\ D_{BA} & D_{BB} & D_{BC} & D_{BD} \\ D_{TA} & D_{TB} & D_{TC} & D_{TD} \end{bmatrix} \begin{bmatrix} A_{MC,A} \\ -A_{MC,B} \\ A_{MC,C} \\ -A_{MC,D} \end{bmatrix} \right\| \tag{3}$$

Whether port A or B of the digital manifold is connected to supply or return of the proportional valve depends on its position. In this way, the flow rate from the supply port ($Q_{MC,P}$) and to the return port ($Q_{MC,R}$) of the proportional valve is given by

$$Q_{MC,P} = \begin{cases} Q_B \; if \; v < 0 \\ Q_A \; if \; v > 0 \end{cases}, \tag{4}$$

$$Q_{MC,R} = \begin{cases} Q_A \; if \; v < 0 \\ Q_B \; if \; v > 0 \end{cases}. \tag{5}$$

The next step is to calculate the pressures on the proportional valve ports connected to supply ($p_{MC,P}$) and to return ($p_{MC,R}$). As for the flow rate, they are dependent on the proportional valve position and are calculated as

$$p_{MC,P} = \begin{cases} \frac{F_L}{A_A} \; if \; F_L > 0 \; and \; v > 0 \\ 0 \quad if \; F_L > 0 \; and \; v < 0 \\ \frac{F_L}{A_B} \; if \; F_L < 0 \; and \; v < 0 \\ 0 \quad if \; F_L < 0 \; and \; v > 0 \end{cases}, \qquad p_{MC,R} = \begin{cases} 0 \quad if \; F_L > 0 \; and \; v > 0 \\ \frac{F_L}{A_A} \; if \; F_L > 0 \; and \; v < 0 \\ 0 \quad if \; F_L < 0 \; and \; v < 0 \\ \frac{F_L}{A_B} \; if \; F_L < 0 \; and \; v > 0 \end{cases}, \tag{6}$$

where $F_L$ is the load force. The pressure on the reservoir port of the digital valve manifold is considered to be zero. To simplify the analysis, the load on the conventional actuator is considered to result in a pressure ($p_{Conv,P}$) and flow rate ($Q_{Conv.P}$) at the supply port of the proportional valve. With this information one can determine the pump pressure ($p_{Pump}$) and flow rate ($Q_{Pump}$) as

$$Q_{Pump} = Q_{MC,P} + Q_{Conv,P}, \tag{7}$$

$$p_{Pump} = max(p_{MC,P}, p_{Conv,P}) + \Delta p_{LS}, \tag{8}$$

where $\Delta p_{LS}$ is the additional pressure difference for the load sensing system. The hydraulic power required by each load is calculated as

$$P_{MC} = Q_{MC,P} p_{MC,P}, \tag{9}$$

$$P_{Conv} = Q_{Conv,P} p_{Conv,P}. \tag{10}$$

The hydraulic power supplied by the pump is calculated as

$$P_{Pump,hyd} = Q_{Pump} p_{Pump}. \quad (11)$$

The power to drive the pump is calculated as

$$P_{Pump} = \frac{P_{Pump,hyd}}{\eta_{vol}\eta_{mech}}, \quad (12)$$

where the mechanical and volumetric efficiencies are considered to be functions of pressure and flow rate only, since a constant pump speed is adopted for this analysis. Knowing the power to each load and the power supplied by the pump, the throttling losses in the control valves between pump and actuators ($P_{Ctrl,Loss}$) are calculated as

$$P_{Ctrl,Loss} = P_{Pump,hyd} - P_{MC} - P_{Conv}. \quad (13)$$

By applying eq. (9) to (11) into eq. (13) and by evaluating eq. (8) for the two possible cases, the analysis can be further extended to

$$if\ p_{Pump} = p_{MC,P} + \Delta p_{LS} \qquad P_{Ctrl,Loss} = (Q_{MC,P} + Q_{Conv,P})\Delta p_{LS} + Q_{Conv,P}(p_{MC,P} - p_{Conv,P}), \quad (14)$$

$$if\ p_{Pump} = p_{Conv,P} + \Delta p_{LS} \qquad P_{Ctrl,Loss} = (Q_{MC,P} + Q_{Conv,P})\Delta p_{LS} + Q_{MC,P}(p_{Conv,P} - p_{MC,P}). \quad (15)$$

From eq. (14) and (15) it is straightforward to evaluate how the selection of different chambers can affect the system efficiency. The first term of both equations indicates that the flow rate of the multi-chamber actuator should be small to obtain a reduction of those terms, which means selecting a combination with smaller areas connected to supply. The second term of both equations indicates that having similar pressure levels would also result in a reduction of those terms, which means selecting a combination that results in as close a pressure as possible to the pressure on the conventional actuator.

It must be noticed that each combination of areas affects the flow rate and pressure, so both terms of eq. (14) and (15) are affected, which means a reduction in the first term might result in an increase in the second term. In this sense, for every different force and speed on the actuators eq. (13) should be re-evaluated. However, the combination will also affect the efficiency of the pump, for which the power loss ($P_{Pump,Loss}$) is calculated as

$$P_{Pump,Loss} = P_{Pump} - P_{Pump,hyd}. \quad (16)$$

The total efficiency of the hydraulic system is then calculated as

$$\eta_{Syst} = 1 - \frac{P_{Pump,Loss} + P_{Ctrl,Loss}}{P_{Pump}}. \quad (17)$$

These equations allow the pressure and flow diagrams for the two loads to be plotted, where for the multi-chamber actuator there will be different flow rates and pressures for the different combinations of areas that are connected to the ports of the proportional valve, as defined in eq. (4) to (6).

## 6 Results

As show in eq. 1 and 2, flow rates at the proportional valve ports are calculated directly from the actuator speed and combination of chambers. Results for a positive and negative actuator speed are presented in fig. 4. The opening position of the proportional valve is also shown in fig. 4, where -1 means P→B and A→T, and +1 means P→A and B→T.

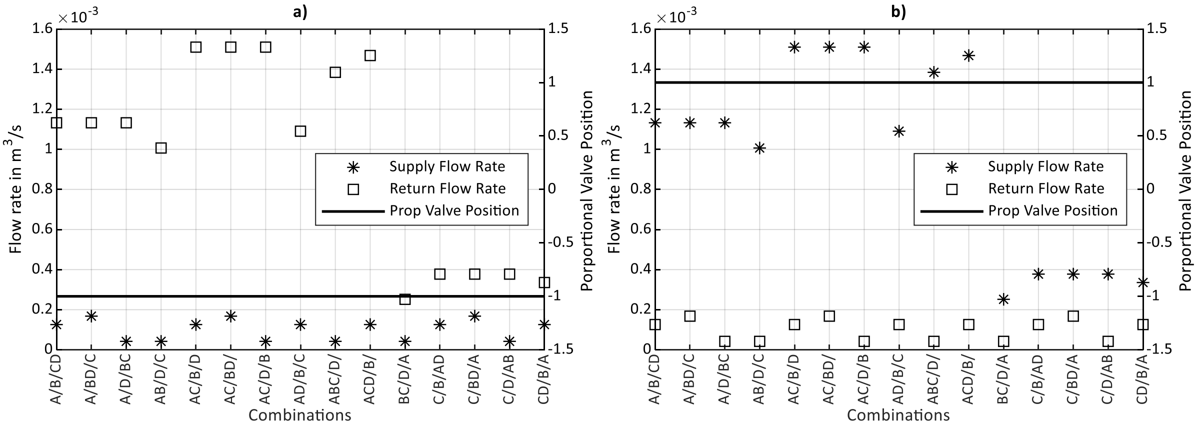


*Figure 4 - Flow rate at supply and return ports of the proportional valve; a) $v = -0.2\ m/s$; b) $v = 0.2\ m/s$.*

For most of the combinations there is a noticeable difference in flow rate between the two ports, which is caused by the asymmetry in the chamber areas. Further consideration of this difference is provided in the discussion section.

By considering a load on each actuator, it is possible to analyse in which conditions the selected concept could result in lower throttling losses. To show representative examples, two cases of actuator loads and speeds are presented that result in a regular load where the load is driven by the power supply, and an overrunning load where the actuator is driven by the load instead.

To establish a comparison with a conventional system, the same calculations are performed for a conventional actuator instead of a multi-chamber actuator driving Load 2 (fig. 3). The conventional actuator areas are $A_{Aconv} = A_A + A_C$ and $A_{Bconv} = A_B + A_D$ which is equivalent to combination 6 in tab. 5. In this configuration it would be capable of exerting the same force as the multi-chamber actuator. The results for this conventional actuator are always shown in the right-most position on the x-axis of the plots with the name $ConvL2$. The left-most position on the x-axis of the plots presents the conventional actuator driving Load 1, named $ConvL1$.

Figure 5 shows the resultant pressure and flow diagrams calculated with eq. (1) to (6) for each of the combinations. In both plots the load force on the multi-chamber actuator is 35 kN, speed is -0.2 m/s for the overrunning load and 0.2 m/s for the regular load, flow rate and pressure of the conventional actuator are 4.2x10$^{-4}$ m$^3$/s (25 L/min) and 15 MPa. The width of the bars for each combination is the flow rate required from the pump and the height is the load pressure. The dash-dotted line represents the pump pressure calculated with eq. (8). In these plots, the throttling losses can be visualized as the area between the pump pressure curve and the loads.

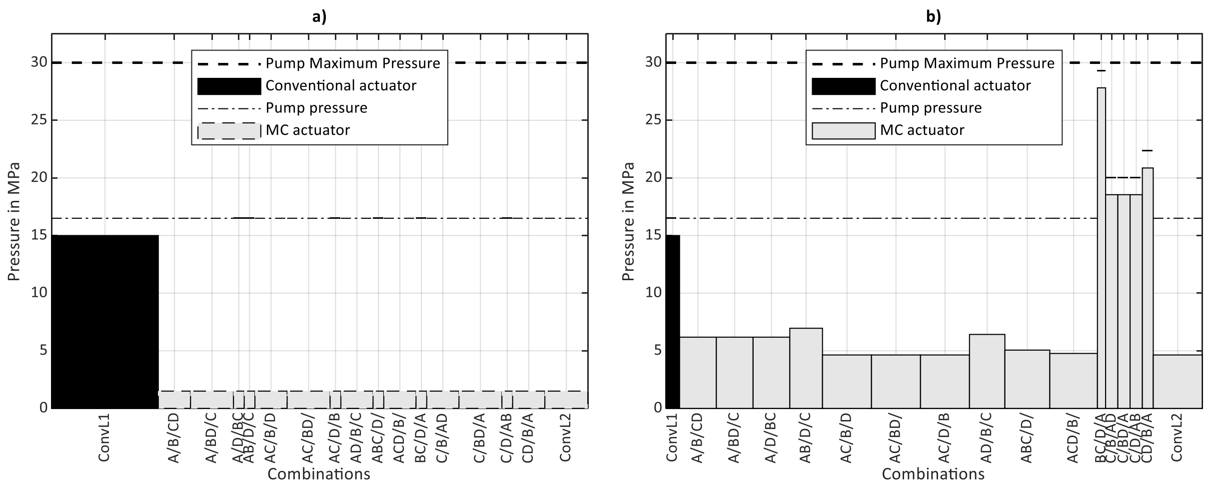


*Figure 5 - Proportional valve supply port flow rate vs pressure: a) Overrunning load; b) Regular load.*

Figure 5a shows the results for the overrunning load, where the pressure in the chambers connected to supply would be very small. In the calculations they were considered to be zero but to enable the visualization, in the plot they were assigned a small value. Independent of the pressure, the pump must supply the flow rate to fill up the chambers. In such situations the combination could be chosen to minimize the flow rate required from the pump to reduce the throttling losses. For example, combination *A/D/BC* has lower throttling losses than combination *A/B/CD*.

Figure 5b shows the results for a regular load, where it is seen that certain combinations have the potential to reduce the throttling losses. For this load situation, the combination that would result in a higher pressure and lower flow rate, like combination *CD/B/A,* would have an advantage over a low pressure and high flow combination like *A/B/CD*. This will become clear when the efficiency plots are shown. For loads higher than the one evaluated, it is likely that certain combinations would result in too high pressure, rendering them infeasible. This is because the chambers connected to supply result in a small combined area. Figure 6 presents the pressure and flow diagrams for the return port of the proportional valve.

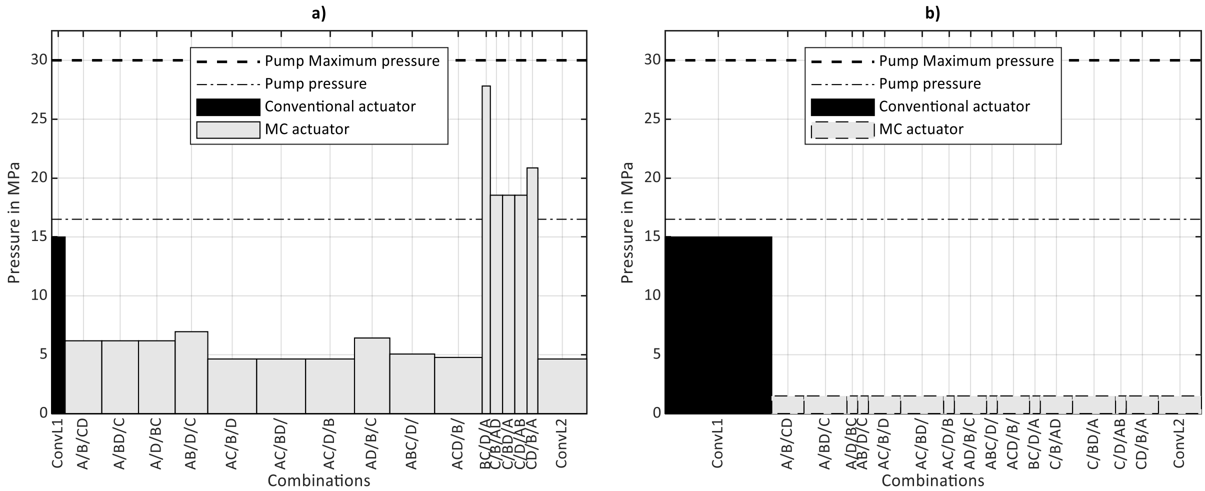


*Figure 6 - Proportional valve return port flow rate vs pressure: a) Overrunning load; b) Regular load.*

Figure 6a presents the return flow and pressure diagrams for the overrunning load, where the movement is controlled by the meter-out edge of the proportional valve. This diagram shows that, if one would like to recover this available energy and has the means for that in terms of installed components, it would be possible to also modulate the pressure and flow rate to the energy recuperation and storage system as well.

Figure 6b highlights the flow rate on the return port for a regular load. In the calculations the pressure is assumed to be zero since it would be connected to a reservoir, but in the plot a small value is assigned to it to enable the visualization.

Figure 7 presents the results for the calculation of pump and throttling power losses and the required hydraulic power according to eq. (9) to (16).

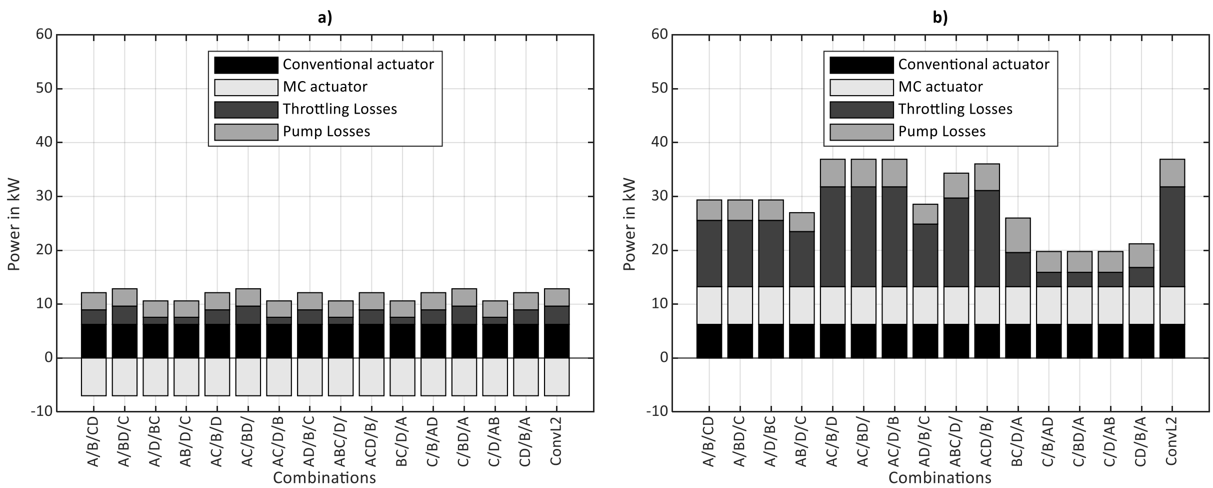


*Figure 7 - Power supply and power loss: a) Overrunning load; b) Regular load;*

Figure 7a shows the power supplied to and lost by the system for the case of an overrunning load. The load on the multi-chamber actuator is lost when throttling the flow back to the reservoir, since in the presented architecture it is not possible to recover this energy. Although it is an overrunning load, the throttling losses can be affected by which combination is selected. This is because the combination affects the flow rate supplied to the unpressurized chamber of the overrunning actuator, which still result in throttling losses over the control valves. By selecting a combination with smaller chambers connected to supply, the throttling losses are minimized. Although it is an overrunning load, the compensation valve must control the same pressure drop upon the proportional valve.

Figure 7b shows the power supplied to and lost by the system for the case of a regular load. It is clear that some combinations would result in less power supplied by the system to perform the movement. Although the pump losses are not constant for all combinations, the largest difference is caused by the reduction in throttling losses.

Figure 8 shows the system efficiency for both conventional and multi-chamber actuators calculated by eq. (17). It is notable that most of the combinations of chambers would result in a higher efficiency than if the same load was driven by a conventional actuator with equivalent areas. The increased efficiency is very clear for both cases of regular and overrunning load.

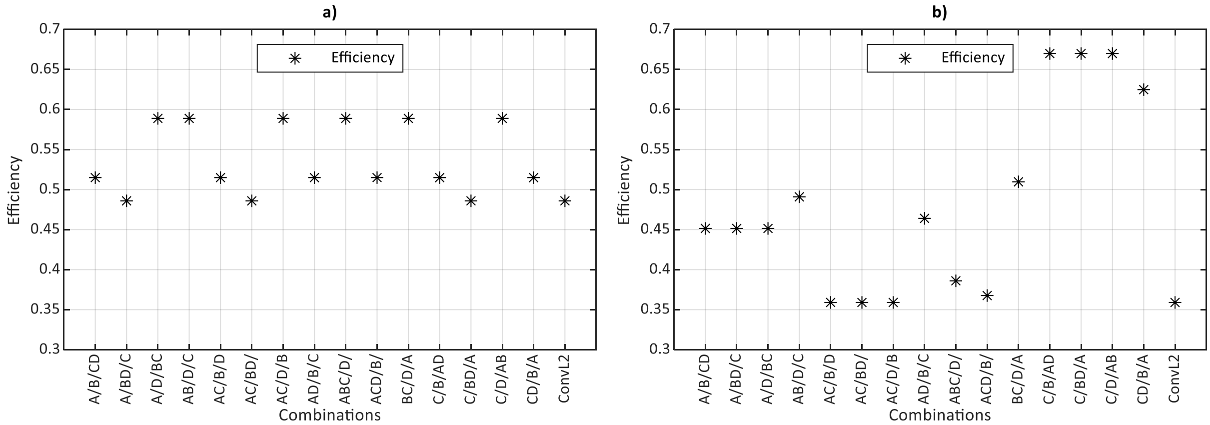


*Figure 8 - System efficiency: a) Overrunning load, v=-0.2m/s; b) Regular load, v=0.2m/s.*

## 7 Discussion

Figure 4 shows a considerable difference between the magnitudes of flow rates at the ports of the proportional valve. This flow rate disparity is an issue since the pressure drops in the control edges of the proportional valve would be significantly different. However, when selecting combinations from the redundant ones, the larger flow rates for the different chambers can be assigned to the same ports, where the opposite is also true. In this sense, the asymmetry of the proportional valve could be designed/selected with the aim of having as close a match as possible to the asymmetry of the chamber areas. A study on matching the area ratio of actuators and areas of control edges of proportional valves is presented in [9].

The ratio between the chamber areas has a significant effect on the available combinations to be chosen from – on the one hand because more combinations can be made feasible with respect to flow direction in the proportional valve, as discussed in Section 4. On the other hand, they could be chosen to not result in many infeasible pressures or flow rates, e.g. when subjected to loads with high speed and/or force. The choice of areas would also affect the overall efficiency of the system, where a wider choice of combinations would be available depending on the load. This is a topic for further research, possibly by defining an optimization problem for the system parameters.

An argument against such concept is that it requires more hydraulic lines to establish the connections between actuator, digital manifold, proportional valve and reservoir. Another argument against it is that the return line, port T, would need to be pressurised to avoid cavitation in the actuator.

As mentioned in [10], an important feature of the parallel connected systems is that no switching is needed in order to maintain any of the combinations. Switching the valves is needed only when a different output is desired. In this sense, the current system can operate according to the load expected for a particular operation, for example a grading operation of an excavator. For the whole duration of that operation, it could maintain the same actuator set-up since no major load variations would exist. If another operation has larger load variations then switching can be executed more frequently as well.

Although in this study a multi-chamber actuator with four chambers was studied, the same concept could be applied to a three-chamber actuator. This would result in a smaller number of available combinations, but likely also reduced component costs, as a consequence of having fewer on/off valves and simpler cylinder construction.

## 8 Conclusions

This paper presented a possible solution to reduce throttling losses of valve-controlled hydraulic systems with the use of multi-chamber actuators. A case study was presented for a load sensing system used to drive two loads with the same pump. It was shown that the different combinations of areas of the multi-chamber actuator can be used to adapt the resultant pressure and flow rate from its load to the pressure and flow rate resultant from the load on the other actuator. In this way, throttling losses that occur due to resistive control can be reduced significantly. It was also suggested that a smart selection of the available combinations can lead to a simpler controller, since the number of combinations is reduced and possibly fewer changes to the control of the proportional valve are required. Further research should be directed towards the evaluation of design parameters and design of the controller.

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## Nomenclature

| Designation | Denotation | Value | Unit |
|---|---|---|---|
| $v$ | Multi-chamber actuator speed | - | $m/s$ |
| $F_{load}$ | Multi-chamber actuator load | - | $N$ |
| $p_{Conv,P}$ | Conventional actuator prop. valve load pressure | - | $Pa$ |
| $Q_{Conv,P}$ | Conventional actuator prop. valve flow rate | - | $m^3/s$ |
| $p_{Pump}$ | Pump pressure | - | $Pa$ |
| $Q_{Pump}$ | Pump flow rate | - | $m^3/s$ |
| $\Delta p_{LS}$ | Load sensing pressure differential | 1.5e6 | $Pa$ |
| $Q_{MC,P}$ | Multi-chamber actuator prop. valve supply flow rate | - | $m^3/s$ |
| $p_{MC,P}$ | Multi-chamber actuator prop. valve pressure of the port connected to supply | - | $Pa$ |
| $Q_{MC,R}$ | Multi-chamber actuator prop. valve return flow rate | - | $m^3/s$ |
| $p_{MC,R}$ | Multi-chamber actuator prop. valve pressure of the port connected to return | - | $Pa$ |
| $Q_{MC,A}$ | Multi-chamber actuator prop. valve port A flow rate | - | $m^3/s$ |
| $p_{MC,A}$ | Multi-chamber actuator prop. valve port A pressure | - | $Pa$ |
| $Q_{MC,B}$ | Multi-chamber actuator prop. valve port B flow rate | - | $m^3/s$ |
| $p_{MC,B}$ | Multi-chamber actuator prop. valve port B pressure | - | $Pa$ |
| $Q_{MC,T}$ | Multi-chamber actuator port T flow rate | - | $m^3/s$ |
| $p_{MC,T}$ | Multi-chamber actuator port T pressure | 0 | $Pa$ |
| $Q_{MC,A}$ | Multi-chamber actuator port A flow rate | - | $m^3/s$ |
| $Q_{MC,B}$ | Multi-chamber actuator port B flow rate | - | $m^3/s$ |
| $Q_{MC,C}$ | Multi-chamber actuator port C flow rate | - | $m^3/s$ |
| $Q_{MC,D}$ | Multi-chamber actuator port D flow rate | - | $m^3/s$ |
| $[A_A\ A_B\ A_C\ A_D]$ | Multi-chamber actuator chambers areas | [27 3 9 1] $A_D$ | $m^2$ |
| $A_D$ | Multi-chamber actuator chamber D areas | 2.097e-4 | $m^2$ |

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