ABSTRACT

This paper presents a conceptual study aiming to improve the compactness of electro-hydraulic compact drives (ECD’s). In most current ECD architectures, gas accumulators are used as volume compensators for the flow imbalance emerging whenever asymmetric single rod cylinders are used. To stay within a required reservoir pressure range typically from two to four bar, a large gas volume is required, compromising system compactness. Combining conventional ECD architectures with a bootstrap reservoir offers a greater degree of freedom in system design, which enables downsizing or avoidance of the gas volume. Another potential benefit by including a bootstrap reservoir is the possibility of elevating the backpressure of the ECD thus enhancing drive stiffness, expanding the application range and market acceptance. Based on an open analysis of the solution space occurring when introducing a bootstrap reservoir, three system architectures are selected for a conceptual study. The results show that the downsizing potential is strongly dependent on the maximum friction force and the area ratio of the bootstrap reservoir pistons, while a linear analysis reveals that for some system architectures the bootstrap reservoir may severely influence the system dynamics. Simulation results confirm the functionality of the proposed system architectures, and show that a potential for downsizing/avoiding the gas volume, as well as increasing the ECD stiffness is present.

Keywords: Electro-hydraulic, compact cylinder drive, self-contained cylinder, bootstrap reservoir

1. INTRODUCTION

Electro-hydraulic compact drives (ECD’s) are an emerging technology in a range of industrially available linear drive solutions. ECD’s can generally be characterised as pump-controlled cylinder drives based on variable-speed electric motors and fixed-displacement pumps combined in a compact unit including a fully enclosed oil circuit. A feasible ECD circuit must be able to compensate the flow imbalance emerging whenever asymmetric single rod cylinders are used. Numerous solutions to achieve this have been proposed and an extensive overview is given in [1]. Generally the compensation methods may be divided in two groups: valve-compensation and pump-compensation [2]. Examples of each approach are found in Figure 1, wherein (a) an inverse shuttle valve connects the low pressure chamber with the gas accumulator to compensate the flow asymmetry, while in (b) a pump-compensated architecture is shown. Here the pump displacements are matched, such that the two-quadrant pump is providing the cylinder rod flow. The two examples shown are well-known in research literature and detailed investigations and experimental validations may be found in [2]–[5] for valve-compensated systems, and in [6]–[10] for pump-compensated systems.

Figure 1: Examples of (a) valve-compensated and (b) pump-compensated ECD.
In current ECD architectures, gas accumulators are used to store the rod volume as well as compression and thermal expansion volumes. The accumulator defines the baseline pressure of the system, which has conflicting optimization targets: On the one hand, it is supposed to be high in order to avoid pump cavitation and to increase the bulk modulus of the oil/air mixture and hence yield stiffer cylinder drives. On the other hand, pump housings and especially their shaft sealing offer longer lifetime at lower pressures and may even malfunction if a low-pressure leakage line is unavailable. For this reason, accumulator pressures usually do not exceed four bar. In order to stay within this narrow pressure range under all operating conditions, the gas volume needs to be significantly larger than the rod volume of the cylinder, rendering the accumulator bulky in comparison to the cylinder. Especially for long stroke cylinders, this compromises the desired compactness of the entire system. One approach for reducing the accumulator volume and increase system stiffness is given by combining the conventional ECD architectures in Figure 1 with a so-called bootstrap reservoir.

1.1. Bootstrap Reservoirs

Bootstrap reservoirs can essentially be regarded as two interconnected differential cylinders, which may be arranged in different configurations as seen in Figure 2. Common for the three configurations is that one or more chambers are vented to the atmosphere, while the effective area between in the bootstrap chamber ($A_R$) and the reservoir chamber ($A_C$) differs significantly. In commercially available bootstrap reservoirs the reservoir piston area may be as much as ~85 times larger than the bootstrap area [11].

![Bootstrap Reservoirs](a)-(c) Bootstrap reservoirs shown as two interconnected differential cylinders.

By the pressure $p_C$ the reservoir pressure $p_R$ is elevated above atmospheric pressure to avoid pump cavitation and to ensure that the reservoir may function in arbitrary orientations. Bootstrap reservoirs are vital in aircraft hydraulic systems ([11], [12]), where either the hydraulic pump pressure or compressed air, pressurises the bootstrap chamber. In this paper, the configuration shown in Figure 2 (a) is utilized for analysis. For relative pressures, in static conditions neglecting friction and gravitational loads it is evident that $p_R A_R = p_C A_C$, leading to $p_R = \alpha_B p_C$ for $\alpha_B = A_C / A_R$, meaning that the bootstrap reservoir may be regarded as a volume-to-pressure transmission [13].

1.2. Paper Outline

In Sec. 2 the solution space occurring when combining a bootstrap reservoir with the two different ECD’s from Figure 1 is derived. From the feasible subspace three different systems have been selected, to each investigate the following three potentials originating from the combination of bootstrap reservoirs and ECD’s:

1. **Downsizing (Sec. 3)**
   - The volume-to-pressure transmission property of the bootstrap reservoir may be utilised to downsize the gas volume while increasing the gas pressure.

2. **Gasless (Sec. 4)**
   - The gas accumulator can be avoided by replacing it directly with the bootstrap reservoir. If no gas volume is present, the baseline pressure of the system must be defined by the bootstrap chamber, which has to be charged by means of additional circuitry in order to define a reservoir pressure above atmosphere.

3. **Stiffness (Sec. 5)**
   - If the high pressure bootstrap chamber is connected with the low pressure side of the cylinder, it may be possible to elevate the back-pressure of the cylinder and as such improve the stiffness of the ECD.

2. COMBINING COMPACT DRIVES AND BOOTSTRAP RESERVOIRS

Based on Figure 3, it is investigated how a bootstrap reservoir may be included in the design of the two ECD architectures from Figure 1.

![Combining Compact Drives and Bootstrap Reservoirs](Figure 3: To investigate the solution space, the possible configurations when combining ports A, B and C with ports 1 and 2 are considered.)
The valve-compensated ECD emerges when combining the single pump architecture 1P with the system (cylinder, inverse shuttle valve, and accumulator), whereas the pump-compensated ECD emerges if considering pump architecture 2P. Note, that the inverse shuttle valve is usually not included when considering pump architecture 2P, but is inserted here to extend the possible ways of introducing the bootstrap reservoir, thus gaining benefits.

The feasible solution space is initially constrained because port A must be connected to the low pressure port 1. System ports B and C may generally be connected to bootstrap port 1, 2 or stay unconnected. However, for the valve-compensated architecture (1P) system port C cannot stay unconnected. Altogether, this leaves six system architectures for the valve-compensated architecture (1P) and nine for the pump-compensated architecture (2P), which are listed in Table 1. In this table, the improvement potentials are indicated, as well as system architectures that are obviously infeasible or that require additional circuitry to obtain the indicated potential. Please note, that only obviously infeasible architectures are indicated, meaning that other architectures may turn out infeasible or needing additional circuitry when analysed thoroughly.

The circuit configurations with ID 2, 4, 6 are obviously infeasible, as the pump leakage oil is not able to re-enter the closed oil circuit. For ID 4, this is illustrated in Figure 4.

Pump leakage causes oil from the high pressure side \((p_H)\) leaking towards the low pressure side \((p_L)\), thus causing the bootstrap reservoir piston to drift until the end-stop is reached, rendering the device non-functional. To compensate piston drift, leakage oil must be able to re-enter the closed system from the low pressure side of the bootstrap reservoir. Since there is no part of the system with lower pressure available, this is clearly not possible by means of valves.

For the considered architecture this example shows that it is not possible to elevate the back-pressure of the cylinder, without including an active component to displace fluid from the low pressure to the high pressure side of the bootstrap reservoir.

To investigate each of the potentials arising when introducing a bootstrap reservoir in ECD design, three of the circuit configurations from Table 1 are selected. To investigate the downsizing potential, gasless potential and increased stiffness potential, system ID 3, 5 (valve-compensated) and 8 (pump-compensated) are chosen as a starting point, respectively. The selected architectures are shown in Figure 5.

<table>
<thead>
<tr>
<th>ID</th>
<th>X to #</th>
<th>Potential</th>
<th>Comment</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>B C</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>1 1</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>1 2</td>
<td>-</td>
<td>Infeasible</td>
</tr>
<tr>
<td>3</td>
<td>2 1</td>
<td>Downsizing</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>2 2</td>
<td>-</td>
<td>Infeasible</td>
</tr>
<tr>
<td>5</td>
<td>- 1</td>
<td>Gasless</td>
<td>Circuitry required</td>
</tr>
<tr>
<td>6</td>
<td>- 2</td>
<td>Gasless</td>
<td>Infeasible</td>
</tr>
<tr>
<td>7</td>
<td>1 1</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>1 2</td>
<td>Stiffness</td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>1 -</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>10</td>
<td>2 1</td>
<td>Downsizing</td>
<td></td>
</tr>
<tr>
<td>11</td>
<td>2 2</td>
<td>Stiffness</td>
<td></td>
</tr>
<tr>
<td>12</td>
<td>2 -</td>
<td>Downsizing</td>
<td></td>
</tr>
<tr>
<td>13</td>
<td>- 1</td>
<td>Gasless</td>
<td>Circuitry required</td>
</tr>
<tr>
<td>14</td>
<td>- 2</td>
<td>Gasless &amp; stiffness</td>
<td>Further investigation needed</td>
</tr>
<tr>
<td>15</td>
<td>- -</td>
<td>Gasless</td>
<td>Circuitry required</td>
</tr>
</tbody>
</table>

Figure 4: Infeasible system architecture (ID 4).
3. DOWNSIZING OF GAS VOLUME

The downsizing potential of the gas volume is investigated using the system architecture shown in Figure 5 (a). In this configuration, the bootstrap reservoir contains a low-pressure oil chamber and a high-pressure gas chamber, and may as such be regarded as a piston accumulator with uneven piston areas. This enables replacing the low pressure/high volume accumulator in Figure 1 with a gas working under high pressure/low volume. The gas is considered ideal and the compression/expansion process as polytropic obeying:

\[ p_{aG0} V_{G0}^k = p_{aG1} V_{G1}^k = p_{aG2} V_{G2}^k \]

(1)

\[ p_{aG0} < p_{aG1} < p_{aG2} \] is the pre-charge, minimum and maximum gas working pressure respectively. The subscripted “a” indicates that absolute pressures are utilised.

The minimum gas volume \((V_{G2})\) may be found as:

\[ V_{G2} = V_{G1} \left( \frac{p_{aG1}}{p_{aG2}} \right)^{\frac{1}{k}} \]

(2)

By defining the compensation volume or active volume as \(\Delta V = (V_{G1} - V_{G2})\), the ratio of initial gas volume \(V_{G1}\) to \(\Delta V\) for the conventional configuration in Figure 1 (a) is from Eq. (2) found as:

\[ \frac{R_{Acc}}{\Delta V} = \left( 1 - \left( \frac{p_{aG1}}{p_{aG2}} \right)^{\frac{1}{k}} \right)^{-1} \]

(3)

Please note that to utilise \(\Delta V\) in Eq. (3), no oil is present in the accumulator at \(p_{aG1}\), meaning that the initial gas volume \((V_{G1})\) is here regarded as the total accumulator size. To achieve proper utilisation and lifetime of the accumulator the pre-charge pressure \(p_{aG0}\) is customary chosen to 90% of \(p_{aG1}\) yielding a larger accumulator \((V_{G0} > V_{G1})\) than given by Eq. (3). To achieve a fair comparison between the volume of the accumulator and the gas-loaded bootstrap reservoir only the active oil volume \(\Delta V\) is included in this comparison.

For the configuration in Figure 5 (a), the reservoir piston friction \((F_{Pz})\) needs to be included when estimating the gas volume. The force equilibrium given in relative pressures for the reservoir piston, neglecting gravitational load is:

\[ \ddot{x}_{MB} = A_R p_{PR} - A_C p_{GB} - F_{Pz} \]

(4)

Assuming that the mass \(M_B\) is small, the acceleration force may be neglected and by expressing \(F_{Pz}\) as an equivalent pressure working on \(A_R \) \((F_{Pz} = p_{PR} A_R)\), yields:

\[ A_R (p_{PR} - \alpha_B p_{GB} - p_f) = 0 \]

(5)

The force equilibrium is transformed to absolute pressures, by also including the forces working on the system from the surroundings. The reservoir pressure is found as:

\[ p_{aR} = \alpha_B p_{aGB} + p_{atm}(1 - \alpha_B) + p_f \]

(6)

The minimum and maximum reservoir pressures in the presence of the maximum piston friction \(p_{Pm}\) are given by:

\[ p_{aR1} = \alpha_B p_{aGB1} + p_{atm}(1 - \alpha_B) - |p_{Pm}| \]

\[ p_{aR2} = \alpha_B p_{aGB2} + p_{atm}(1 - \alpha_B) + |p_{Pm}| \]

(7)

From Eq. (7) the required limits for the gas pressure in the bootstrap chamber is found as:

\[ p_{aGB1} = \alpha^{-1}_B (p_{aR1} - p_{atm}(1 - \alpha_B) + |p_{Pm}|) \]

\[ p_{aGB2} = \alpha^{-1}_B (p_{aR2} - p_{atm}(1 - \alpha_B) - |p_{Pm}|) \]

(8)

From Eq. (8) it may be observed that piston friction narrows the allowable range of gas pressures, requiring a larger gas volume. It is required that \(p_{aGB1} < p_{aGB2}\), from which it follows that \(|p_{Pm}| < 0.5(p_{aR2} - p_{aR1})\). The initial gas-volume of the bootstrap chamber, \(V_{GB1}\), may be found from Eq. (3), as:
The volume of the bootstrap reservoir is calculated as
\[
V_{GB1} = \Delta V_{\alpha_B} \left(1 - \left(\frac{P_{aGB1}}{P_{aGB2}}\right)\right)^{-1}
\]  
(9)

The volume of the bootstrap reservoir is calculated as
\[
V_{\text{Boot}} = \Delta V + V_{GB1}, \text{ such that the ratio of the bootstrap reservoir volume to } \Delta V \text{ can be established as:}
\]
\[
R_{\text{Boot}} = \frac{V_{\text{Boot}}}{\Delta V} = 1 + \alpha_B \left(1 - \left(\frac{P_{aGB1}}{P_{aGB2}}\right)\right)^{-1}
\]  
(10)

By defining \( R = \frac{R_{\text{Boot}}}{R_{\text{Acc}}} = \frac{V_{\text{Boot}}}{V_{G1}} \), the relative size of the gas-loaded bootstrap reservoir to the conventional low pressure gas accumulator may be estimated. Note that the potential for downsizing the gas volume only considers the active volumes, such that volume differences caused from different material wall thickness or other inactive volumes are not included. In Figure 6 (a), \( \Delta P \) is plotted as a function of the required reservoir pressures, while in (b) it is plotted as a function of the bootstrap area ratio and the maximum friction force. Adiabatic conditions have been assumed by defining \( \kappa = 1.4 \).

It is evident that the volume ratio decreases as area ratio and friction force decreases. For a bootstrap area ratio of 1/25, and low friction the bootstrap reservoir may be up to 65% smaller than a gas accumulator for the considered pressure limits. On the other hand, for a bootstrap reservoir with a large area ratio (\( \alpha_B > 0.25 \)) and large friction forces the gas-loaded bootstrap reservoir is actually larger than using a conventional gas accumulator. In the current investigation, the reservoir pressure is allowed to span from two bar to four bar (relative pressures).

Table 2 shows examples of bootstrap reservoir and gas accumulator sizes for different parameters.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>( \alpha_B )</th>
<th>1/10</th>
<th>1/10</th>
<th>1/20</th>
<th>1/20</th>
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<tbody>
<tr>
<td>( \alpha_B )</td>
<td>0.1</td>
<td>0.75 bar</td>
<td>0 bar</td>
<td>0.75 bar</td>
<td></td>
</tr>
<tr>
<td>( P_{aR1} )</td>
<td>20.9 bar</td>
<td>28.4 bar</td>
<td>40.7 bar</td>
<td>55.7 bar</td>
<td></td>
</tr>
<tr>
<td>( P_{aGB1} )</td>
<td>40.9 bar</td>
<td>33.4 bar</td>
<td>80.7 bar</td>
<td>65.7 bar</td>
<td></td>
</tr>
<tr>
<td>( V_{GB1} / \Delta V )</td>
<td>0.26</td>
<td>0.91</td>
<td>0.13</td>
<td>0.45</td>
<td></td>
</tr>
<tr>
<td>( R_{\text{Boot}} )</td>
<td>1.26</td>
<td>1.91</td>
<td>1.13</td>
<td>1.45</td>
<td></td>
</tr>
<tr>
<td>( R_{\text{Acc}} )</td>
<td>3.27</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>( R )</td>
<td>0.39</td>
<td>0.59</td>
<td>0.35</td>
<td>0.44</td>
<td></td>
</tr>
</tbody>
</table>

3.1. Selection of reservoir diameter

To reduce throttling losses the inverse shuttle valve connecting the low pressure chamber and the reservoir volume is typically not particularly restrictive. This means that the pressure dynamics of the connected chambers are almost similar, in turn causing volume changes of one chamber to directly influence the adjoining chamber dynamics. As such the two movable masses (cylinder and reservoir rods/pistons) may interact with each other, causing the cylinder motion dynamics to be changed when introducing the bootstrap reservoir. This may cause difficulties designing a position/velocity controller, meaning that these interactions should be reduced if possible.

To investigate how these interactions depend on the selected reservoir diameter, a simplified linear model, based on the simplified model structure showed in Figure 7 is considered. The inverse shuttle valve is assumed ideal (no pressure drop), the gas pressure constant, while pump shaft dynamics and leakage is neglected.
This is found to contain two complex pole pairs and one complex pair of zeros. The variation of the poles and zeros as the reservoir diameter is changed from 30 mm to 100 mm, is shown in Figure 8. The viscous damping coefficient is scaled linearly as a function of piston circumference, while the reservoir volume is fixed leading the reservoir stroke length to be ranging from 720 mm ($D_R = 30 \text{ mm}$) to 65 mm ($D_R = 100 \text{ mm}$).

The remaining parameters are fixed, including the mass of the bootstrap rod and pistons of 10 kg. In Figure 8, the linearization point is at the centre position and a positive cylinder speed of 100 mm/s. No qualitative differences are found by varying the linearization point.

At $D_R = 30 \text{ mm}$ Figure 8 shows that the cylinder motion dynamics are dominated by the complex pole pair located at $-0.75 \pm 250i$, because the dynamics from the complex zeros and the other complex pole pair are located close to each other at $-7.5 \pm 60i$, thus cancelling each other. At large reservoir diameters the complex pair of zeros accompanies the complex pole pair that started at $-0.75 \pm 250i$, while the complex pole pair starting $-7.5 \pm 60i$, moves towards the starting position of the other pole pair. Thus, for large reservoir diameters the motion dynamics are also dominated by a complex pole pair at $-0.75 \pm 250i$. However, for critical diameters the cylinder motion dynamics may not be dominated by second order dynamics only, as the complex pair of zeros is located far away from any of the complex pole pairs in the complex plane.

From a dynamical point of view, the reservoir diameter should therefore be chosen far away from the critical diameters. From a practical point of view, it may be infeasible to select a small diameter, as this yields a long stroke length, and requires low friction. Large reservoir diameters are therefore recommended. For the given system parameters a reservoir diameter of 100 mm is feasible, and thus selected for all three considered systems. Likewise an area ratio of $\alpha_B = 0.1$ is chosen.

3.2. Simulation Results

A simulation model of the system in Figure 5 (a) has been formulated and solved using MATLAB/Simulink. The full set of equations is found in Appendix A, while the modelling parameters are found in the Nomenclature. The load is modelled as a mass/spring/damper system as seen in Figure 9 (a), where the spring pre-tensioning is such that four quadrant operation is obtained. This load system is used for the remainder of the paper.

For the chosen cylinder dimensions (63/36-500mm), the needed $\Delta V$ is 0.51 L. The bootstrap reservoir is modelled with a maximum friction force of $A_B = 0.75 \text{ bar} = 590 \text{ N}$ and $\alpha_B = 0.1$. From Table 2 it is seen that the needed accumulator/reservoir volume is reduced from 1.66 L to 0.97 L. An example of bootstrap reservoir dimensions is given in Figure 9 (b). The cylinder piston is controlled to follow a sinusoidal position reference, using a proportional position feedback controller and static/passive velocity feedforward. The simulation results for four quadrant operation is given in Figure 10.
Figure 10: Four quadrant simulation results for the ECD incorporating a gas-loaded bootstrap reservoir (Figure 5 (a)).

The primary observation is that the reservoir pressure $p_R$ is kept within the design limits from two to four bar. The gas pressure $p_{GB}$ is only position dependent, while $p_R$ is also motion direction dependent, due to the directional shift of the friction force. Severe velocity oscillations are observed for both $\dot{x}$ and $\dot{z}$, when the direction of the external load changes. This is caused by the position change of the inverse shuttle valve, which is a general drawback of valve-compensated ECD [14], [15] and is not related to the use of the bootstrap reservoir. This is confirmed in Figure 11 where no abrupt velocity oscillations are observed. Here the pre-tensioning of the load spring is adjusted such that the load is unidirectional.

The robustness towards changed friction forces working on the bootstrap reservoir, is investigated in Figure 12. If the maximum friction force equals the design friction force, the reservoir pressure approaches the design limits. If the friction force is reduced, the reservoir pressure is kept well within limits, while it is exceeding the limits, if the friction force is larger than expected.

Figure 11: Simulation results for two quadrant operation.

Note that the changed friction is only observed in the reservoir pressure and not the gas pressure, as this is only position dependent.

Figure 12: Simulation results in two quadrant operation, for varying bootstrap friction forces.

4. GASLESS SYSTEM ARCHITECTURE

To investigate the potential for avoiding a gas volume, additional circuitry is added to the system configuration in Figure 5 (b). The proposed system is shown in Figure 13, where conventional pressure compensator valves PC2 and PC3, are used to charge and discharge the bootstrap chamber respectively.

Figure 13: Proposed gasless system. A charge valve (PC2) and a discharge valve (PC3) are included.

The functionality of the added circuitry is illustrated in Figure 14. When the bootstrap reservoir retracts (positive cylinder speed), the bootstrap chamber volume increases, requiring an inlet/charge flow to maintain the charge pressure. This flow must be acquired from the cylinder chamber with the largest pressure, as the low pressure chamber is connected to the reservoir side of the bootstrap reservoir, otherwise violating $p_c > p_R$. To control the charge flow, a normally open pressure compensator PC2 is proposed. When $p_c$ is smaller than the reservoir pressure plus some equivalent spring force, the chamber is charged, while the valve is closed if $p_c$ exceeds the equivalent spring force plus the reservoir pressure. A shuttle valve, is used to select the largest system pressure as the charge source, rendering this approach functional for bidirectional external load directions.
When the bootstrap reservoir extends (negative cylinder speeds), the bootstrap chamber volume decreases, requiring a flow to leave the bootstrap chamber. This is achieved by the normally closed discharge valve PC3, which directs the discharge flow to the reservoir chamber, when $p_C$ exceeds the reservoir pressure plus some equivalent spring force.

It is pivotal for the functionality of the proposed system that the limits of $p_C$ enable the valve springs to be selected such that at least one of the valves PC2 and PC3 are fully closed at all times. As indicated in Figure 14, this entails the pressure in the bootstrap chamber to be dependent on the motion direction rather than the piston position, in-turn causing $p_R$ to be motion direction dependent.

Because the bootstrap reservoir does not need to contain any gas at the fully extended position, the volume of the reservoir can be made smaller than for the system in Sec. 3. The total active reservoir volume for the gasless system is 0.62 L compared to 0.97 L for the gas-loaded bootstrap reservoir (Sec. 3).

4.1. Simulation Results

Similarly to Sec. 3.2, the proposed system is investigated via a simulation study. The set of equations can be found in Appendix A. For the valves PC2 and PC3, it is assumed that the valve dynamics are dominated by the pressure dynamics in the pilot lines. Therefore the spool opening dynamics are simplified and modelled as first order systems ($t_v = 10$ ms), with the input being the effective pressure difference experienced by the valve spool.

Figure 15 shows the simulation results for the gasless system for the four quadrant operation cycle. As with the system based on the gas-loaded bootstrap reservoir, severe velocity oscillations are observed as the direction of the external load force changes. When this happens, the gasless system is furthermore experiencing a more critical problem. When both chamber pressures are low (small external load), there is no charge source available to fill the bootstrap chamber at positive cylinder speeds. This causes $p_C$ to decline, in turn lowering $p_R$ below acceptable limits. If no friction forces were present $p_R$ would approach atmospheric pressure. Due to friction $p_R$ drops even further, in the given example to $-0.5$ bar (relative), which is non-tolerable. In the simulated example, the chamber pressures are low, while the cylinder speed is large, which make the problem more apparent, as the demand of charge flow is large under these conditions. At standstill, it is possible for the system to operate at low chamber pressures (no external load), without $p_R$ dropping below atmospheric pressure, but $p_R$ cannot be elevated without sufficient charge pressure. When the bootstrap reservoir extends, while the load force is low, the drop in reservoir pressure does not occur because no charge source is needed under these circumstances. The reservoir can be regarded self-charging in the retracting direction of the cylinder.

Figure 16 shows the simulated system in two...
quadrant operation, where the external load ensures a sufficient charge flow to be provided when necessary. Under these restricted conditions, the system is observed to behave as desired. The reservoir pressure is controlled within the design limits from two bar to four bar.

Figure 16: Simulation results for the proposed gasless system, operated in two quadrants.

Unlike the gas-loaded system, neither the pressure in the bootstrap nor reservoir chamber are position dependent, but depend on motion direction only.

The proposed gasless reservoir system should generally not be considered if there is a risk for lacking sufficient charge pressure. The proposed system may therefore be relevant for ECD architectures where it is possible to control the backpressure of the cylinder to a certain level, ensuring that a charge source is available at all times. Examples of such system architectures can be found in [16]–[20], for single variable-speed electric motor systems. Architectures with two electric machines can be found in [21], [22].

5. ARCHITECTURE FOR INCREASED DRIVE STIFFNESS

Bootstrap reservoirs may offer an opportunity to obtain the advantages that result from increased backpressure. In Sec. 2 the baseline layout for investigating this potential was selected as the double pump concept, given in Figure 5 (c), and redrawn in Figure 17 (a). Here the bootstrap chamber is connected to the lower line pressure via the inverse shuttle valve, allowing for elevating the backpressure, hence increasing ECD stiffness.

For the conventional pump-compensated architecture in Figure 1 (b), the pump displacement volumes \( D_{p1} \) and \( D_{p2} \) must match the area ratio of the cylinder (\( \alpha = A_B/A_A \)) in the following manner:

\[
\frac{D_{p1}}{D_{p2}} = \frac{1 - \alpha}{\alpha} \tag{11}
\]

The imbalance flow is compensated by the gas accumulator. When this is transferred to a system with an additional bootstrap reservoir as depicted in Figure 17 (a), the gas accumulator has to be dimensioned in the same way, since it has to cover the whole volume imbalance. However, a certain mismatch of pump displacements can be applied to reduce the size of the gas accumulator. For both opening directions of the inverse shuttle valve, an optimal matching ratio of the pumps can be established (Table 3), such that the imbalance flow is completely covered by the reservoir chamber of the bootstrap reservoir:

Table 3: Conditions for minimal accumulator size

<table>
<thead>
<tr>
<th>( y_1 )</th>
<th>&gt; 0</th>
<th>&lt; 0</th>
</tr>
</thead>
<tbody>
<tr>
<td>( D_{p1}/D_{p2} )</td>
<td>( \frac{1 - \alpha}{\alpha - \alpha_B} )</td>
<td>( \frac{1 - \alpha}{\alpha - \alpha_B} )</td>
</tr>
</tbody>
</table>

Since the state of the shuttle valve is governed by the direction of the load, a four-quadrant drive cannot be dimensioned to optimality with regard to accumulator size. As a compromise, the pump displacement ratio is chosen to be the mean value:

\[
\frac{D_{p1}}{D_{p2}} = \left( \frac{1 - \alpha}{\alpha - \alpha_B} + \frac{1 - \alpha}{\alpha - \alpha_B} \right)/2 \tag{12}
\]

For low bootstrap area ratios, the deviation of Eq.
(12) compared to Eq. (11) is small, resulting in manageable changes in static speed gain when the external load changes direction.

Measures of leakage compensation in order to avoid bootstrap drifting (see Sec. 2) may include a small charge pump driven by an external motor (Figure 17 (b)) or pressure compensator valves (Figure 17 (c)). The former can be controlled using a simple on/off motor switch to keep accumulator pressure inside the working range. This will automatically ensure the bootstrap reservoir piston to be within its working range. The latter can be based purely on passive elements. Both pressure compensators PC4 and PC5 control the pressure difference $p_C - p_R$. Newton’s second law for the bootstrap piston, formulated for relative pressures results in

$$p_C - p_R = (1 - \alpha_B) \cdot p_C - \frac{F_{\text{pe}}}{A_R} + \frac{M_B \ddot{z}}{A_R}$$

thus $p_C$ can be controlled via the difference $p_C - p_R$ when friction and acceleration forces are neglected.

Pressure compensator PC4 extracts a flow from cylinder chamber A to the bootstrap chamber when a positive load force is available and the pressure difference $p_C - p_R$ is below a critical pressure level. PC5 throttles the pump outflow during cylinder retraction to a pressure level just above $p_C$, such that a flow portion is deviated to the bootstrap chamber via an orifice, when the pressure difference is too low. Both pressure compensators have symmetric pilot areas. Note, that this valve arrangement does not allow leakage compensation under pulling loads while extending the main cylinder.

### 5.1. Simulation results

For a simulation study, a system with the presented valve based leakage compensation approach is parameterised according to the parameters given in the Nomenclature. The governing equations are presented in Appendix A. The simulation study performs the same load cycle as shown in Sec. 3.2, of which results are given in Figure 18.

The simulation results show that the leakage compensation structure is capable of keeping the bootstrap cylinder position within working range. Comparative studies without the compensation measures revealed that under the given load and pump leakage model, a bootstrap reservoir of the chosen size would hit the end stop within less than 5 seconds. The lowest cylinder pressure never drops below 20 bar while the reservoir/accumulator pressure remains within a range of two to four bar.

It is worth noting that the chosen gas accumulator volume is 1 L, compared to a minimum required volume of 2 L in the conventional system. The total volume of the bootstrap reservoir is 0.6 L, while the main cylinder takes a fluid volume of 1.6 L.

### 6. DISCUSSION

This paper presents a conceptional study, investigating how bootstrap reservoirs may be incorporated in the design of electro-hydraulic compact drives, to improve compactness and/or stiffness of the drive systems.

Focus has been placed on investigating the general applicability of the presented three concepts via a simulation study. Here, simplifying assumptions, such as first order valve dynamics and simple cylinder friction models have been applied. This means that the paper serves as an initial investigation of the possibilities arising when combining bootstrap reservoirs and ECD, rather
than a complete design guide. Therefore, it is emphasized that further analysis is needed prior to realization of the presented concepts. This includes an analysis of the dynamic requirements to the pressure compensators, as well as an investigation of the extent to which the required charge flow decreases the energy efficiency of the entire system.

7. CONCLUSION

Gas accumulators are conventionally used as volume compensators in electro-hydraulic compact drives (ECD’s). To stay within a narrow reservoir pressure range, a considerable gas volume is required, compromising system compactness. This paper investigates how ECD compactness may be improved by incorporating a bootstrap reservoir in the design. Based on a systematic derivation of the solution space, three improvement potentials have been identified: Reservoir downsizing, avoidance of gas volume and increased drive stiffness by elevating cylinder backpressure. Three architectures, each representing one of the three improvement potentials, have been selected for a conceptual study.

It is found that the area ratio of the bootstrap reservoir piston as well as the magnitude of friction forces severely affects the downsizing potential. A linear analysis further shows that for some architectures, system dynamics is strongly influenced by incorporating the bootstrap reservoir. To reduce this influence a thorough analysis of the system dynamics is therefore encouraged. If avoidance of the gas volume is desired, a sufficient external load force is needed to define a reservoir pressure greater than the surroundings. In the case of elevated backpressure, additional circuitry needs to be implemented for compensation of pump leakage. The exact functioning of these structures may depend on the load cycle, especially when they are hydraulically piloted.

Simulation results confirm the functionality of the proposed system architectures, and shows that a potential for downsizing/avoiding the gas volume, as well as increasing the ECD stiffness is present.

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**REFERENCES**


[16] L. Schmidt, M. Groenkjaer, H. C. Pedersen, and
APPENDIX A – SIMULATION MODELS

Simulations models are presented in the following, while parameters can be found in the Nomenclature.

System architecture from Sec. 3

Considering Figure 19 (a), the system incorporating a gas-loaded bootstrap reservoir is modelled using Eq. (14) to (31). The pressure gradients of $p_A$, $p_B$, $p_R$, and $p_{GB}$ are given as:

\[
\dot{p}_A = \frac{\beta_A}{V_A}(Q_A - Q_{CA} - A_A\dot{x}) \tag{14}
\]

\[
\dot{p}_B = \frac{\beta_B}{V_B}(A_B\dot{x} - Q_B - Q_{CB}) \tag{15}
\]

\[
\dot{p}_R = \frac{\beta_R}{V_R}(Q_L + Q_{CA} + Q_{CB} - A_R\dot{z}) \tag{16}
\]

\[
p_{GB} = \dot{p}_C = \frac{p_{AGB}K}{V_G} A_c \ddot{z} \tag{17}
\]

The variable volumes $V_A$, $V_B$, $V_R$ and $V_G$ are described as:

\[
V_A = V_{GA} + x A_A \quad , \quad V_B = V_{GB} + (L_z - x) A_B \tag{18}
\]

\[
V_R = V_{GR} + z A_R \quad , \quad V_{GB} = V_{GB1} \frac{p_{AGB}}{p_{AGB1}} \frac{1}{\kappa} \tag{19}
\]

$p_{AGB1}$ is the initial gas pressure at volume $V_{GB1}$, with $\kappa$ being the polytropic process constant.

The nonlinear motion dynamics is described by:

\[
\dot{x} = M_x^{-1}(p_A A_x - p_B A_B - K_s (x - s_{50}) - B_x \dot{x} - F_{pz}) \tag{20}
\]

\[
\dot{z} = M_z^{-1}(p_A A_R - p_C A_c - F_{pz}) \tag{21}
\]

$x$ and $z$ are the cylinder and bootstrap piston positions, with $M_x$, $M_z$, and $F_{pz}$, $F_{pz}$, being mass and friction forces. $s_{50}$, $K_s$ and $B_x$ are spring pre-tension, spring and damping coefficients of the modelled load.

Figure 19: Circuit architectures of the three systems investigated in this paper, including quantity designations.

The friction forces $F_{Fx}$ and $F_{Fz}$ is found as:

\[
F_{Fx} = \tanh(\gamma \dot{x}) F_{Cx} + B_{Fx} \dot{x}, F_{Fz} = \tanh(\gamma \dot{z}) F_{Cz} + B_{Fz} \dot{z} \tag{22}
\]
are Coulomb and viscous friction coefficients. $Q_{CA}, Q_{CB}$ are modelled by the orifice equation as:

$$Q_{CA} = \begin{cases} K_{Q1}|y|\sqrt{|p_a - p_r|}\text{sign}(p_a - p_r), & y \geq 0 \\ 0, & y < 0 \end{cases}$$

$$Q_{CB} = \begin{cases} K_{Q1}|y|\sqrt{|p_a - p_r|}\text{sign}(p_b - p_r), & y > 0 \\ 0, & y \leq 0 \end{cases}$$

with $y_{\text{norm}}$, normalised valve opening and $K_{Q1}$ flow gain. The valve opening $y_{\text{norm}}$ are modelled as a first order dynamic system with time constant $\tau_{\text{v}}$ and input $\ddot{y}_{\text{v}}$, which is calculated based on valve cracking $(P_{CR1})$ and full open $(P_{OP1})$ pressure:

$$\ddot{y}_{\text{v}} = \frac{p_{D1} - p_b}{\tau_{\text{v}}}$$

$$\dot{y}_{\text{v}} = \frac{\ddot{y}_{\text{v}} - y_{\text{v}}}{\tau_{\text{v}}}$$

$Q_A, Q_B, Q_L$ are pump flows modelled by the Wilson model, using pump displacement $D$ and laminar leakage coefficient $K$:

$$Q_A = \omega D - K(p_A - p_b) - K(p_A - p_r)$$

$$Q_B = \omega D - K(p_A - p_b) + K(p_B - p_r)$$

$$Q_L = K(p_A - p_b) + K(p_b - p_r)$$

$\omega$ is the motor shaft speed modelled as a second order dynamic system

$$\ddot{w} = (\ddot{w} - \omega)\omega^2 - 2\zeta \omega \dot{w}$$

with eigenfrequency $\omega_0$ and damping ratio $\zeta$, with $\ddot{w}$ being the system input.

Finally $\beta_i$ is the effective bulk modulus of the fluid air mixture with $\beta_P$ the bulk modulus of the pure fluid, $\sigma$ the volumetric air content at atmospheric pressure and $m$ the pressure dependent bulk modulus parameter:

$$\beta_i = \frac{(1 - \epsilon)\left(1 + \frac{m(p_i - p_{\text{atm}})}{\beta_P}\right)^{\frac{1}{m}} + \epsilon\left(\frac{p_{\text{atm}}}{\beta_i}\right)^{\frac{1}{m}}}{(1 - \epsilon)\left(1 + \frac{m(p_i - p_{\text{atm}})}{\beta_P}\right)^{\frac{m-1}{m}} + \frac{\epsilon\left(\frac{p_{\text{atm}}}{\beta_i}\right)^{\frac{1}{m}}}{\beta_P}}$$

$i = \{A, B, R, C\}$

**System architecture from Sec. 4**

Considering Figure 19 (b) the gasless systems is modelled. The pressure dynamics, are given as:

$$\dot{P}_A = \frac{B_0}{V_A} (Q_A - Q_{CA} - A_A\dot{x} - Q_{CHA})$$

The motion dynamics, friction forces, shuttle valve opening and flows are modelled as the previous system using Eq. (20) to (25). The opening of the pressure compensator valve are modelled as a first order dynamic system given in Eq. (26), with the input $\ddot{y}_{\text{v}}$, and valve flows described by:

$$\ddot{y}_{\text{v}} = \frac{p_{D1} - p_b}{\tau_{\text{v}}}$$

$$\ddot{y}_{\text{v}} = \frac{\ddot{y}_{\text{v}} - y_{\text{v}}}{\tau_{\text{v}}}$$

The variable volumes of $V_A, V_B$ and $V_R$ are found by Eq. (18), (19). $V_C$ are found as:

$$V_C = V_{\text{GC}} + \left(L_{\text{sz}} - z\right)A_C$$

The motion dynamics, friction forces, shuttle valve opening and flows are modelled as the previous system using Eq. (20) to (25). The opening of the pressure compensator valve are modelled as a first order dynamic system given in Eq. (26), with the input $\ddot{y}_{\text{v}}$, and valve flows described by:

$$\ddot{y}_{\text{v}} = \frac{p_{D1} - p_b}{\tau_{\text{v}}}$$

$$\ddot{y}_{\text{v}} = \frac{\ddot{y}_{\text{v}} - y_{\text{v}}}{\tau_{\text{v}}}$$

Finally, the pump flows, shaft dynamics and bulk modulus are described by Eq. (26) to (31).

**System architecture from Sec. 5**

The modelling of the ECD depicted in Figure 19 (c) is carried out in line with the previously described models. Pressure gradients are given by:

$$\dot{P}_A = \frac{B_0}{V_A} (Q_{1A} + Q_{2A} - Q_{CA} - Q_{BA} - A_A\dot{x})$$

$$\dot{P}_B = \frac{B_0}{V_B} (A_B\dot{x} - Q_{CB})$$

$$\dot{P}_P = \frac{B_0}{V_P} (Q_S - Q_R - Q_{LP} - Q_{1B})$$

$$\dot{P}_C = \frac{B_0}{V_C} (A_C\dot{x} + Q_{CA} + Q_{CB} + Q_{LP} + Q_{BA})$$

$$\dot{P}_R = \frac{B_0}{V_{\text{ACC}}} (\frac{p_{\text{atm}}}{P_{AR}})^{\frac{1}{m}} (Q_R + Q_L - Q_S - A_R\dot{x})$$

Pump flows $Q_{1A}, Q_{1B}, Q_{2A}, Q_{2B}$ are calculated in
analogy to Eq. (27) to (29), with leakage parameters $K_1$ and $K_2$. Flows through the pressure compensators ($Q_{BA}, Q_{BR}$) are calculated in analogy with Eq. (38), using flow gains $K_{Q4}$ and $K_{Q5}$ and spool opening dynamics using Eq. (26) and (37). Motion dynamics, inverse shuttle valve behaviour and bulk modulus are modelled as in previous sections, while orifice flow is modelled by:

$$Q_{LP} = \begin{cases} K_{Q2} \sqrt{p_p - p_C} & p_p > p_C \\ 0 & p_p \leq p_C \end{cases}$$

(45)

**APPENDIX B – LINEAR MODEL**

For the system shown in Figure 7 on page 5 a linearized model is derived considering Eq. (14) to (16) and Eq. (20), (21), by assuming the inverse shuttle valve ideal, thus combining the reservoir chamber and the B-chamber in a single continuity equation. By neglecting shaft dynamics and pump leakage, $Q_A = Q_B = Q$ is the system input. Assuming bulk modulus, chamber volumes, external load, gas pressure ($p_G$) and Coulomb friction constant at the linearization point, linear equations can be formulated as:

$$\dot{p}_A = \frac{\beta_{A0}}{V_{A0}} (Q - A_A \dot{x})$$

(46)

$$\dot{p}_B = \frac{\beta_{B0}}{V_{B0}} (A_B \dot{x} - A_R \dot{z} - Q)$$

(47)

$$\dot{x} = M^{-1}_G (p_A A_A - p_B A_B - B_{px} \dot{x})$$

(48)

$$\dot{z} = M^{-1}_G (p_B A_R - B_{pz} \dot{z})$$

(49)

Change variables are given as capital letters and variables evaluated at the linearization point with an indexed “0”. From the linearized equations the transfer function from $Q(s)$ to the selected output may be established.