A Power-Split Hybrid Transmission to Drive Conventional Hydraulic Valve Controlled Architectures in Off-road Vehicles: The Case of a Mini-Excavator

Mateus BERTOLIN¹, Andrea VACCÁ²

¹ Maha Fluid Power Research Center, Purdue University (Lafayette, IN, United States) mbertoli@purdue.edu
² Maha Fluid Power Research Center, Purdue University (Lafayette, IN, United States)

Received January 14, 2022 / Accepted July 7, 2022

In recent years, several disruptive solutions for off-road vehicles hydraulic implements were proposed to increase energy efficiency, many focusing on the reduction of throttling losses. However, the market has showed a limited adoption of such solutions due to high implementation costs and operator’s acceptance. Considering these limitations, this paper proposes to adopt a hybrid power-split hybrid transmission between the supply pump and the prime mover, while still using state-of-the-art hydraulic actuation. Such approach results in an independent control of the engine and pump speeds, therefore improving the efficiency of both. The paper describes the potential application of the proposed solution to a small excavator equipped with a load sensing system. The authors developed a simulation model, which was validated against experiments of the baseline machine. The model was used evaluate fuel efficiency gains achieved with the proposed solution, which are estimated in up to 16% for a truck-loading cycle.

Keywords: Hybrid, Construction Machinery, Power-Split Transmission, Load-Sensing

1. Introduction

Typical hydraulic architectures commonly found in off-road vehicles are well known for two main characteristics: precise motion control, and low energy efficiency associated with the inherent throttling losses of their valve-controlled systems. Especially in machines with multiple simultaneous actuations like excavators, losses caused by the valves are inevitable when a centralized hydraulic supply is used. This is because each pump in the machine usually supplies more than one actuator at the same time. In addition, many vehicles today available in market operate in not optimal conditions for the engine and the pumps. This happens because the faster dynamics and the high installed power capacity of the hydraulic pumps force the engine to operate at high speeds to avoid stall. Additionally, the power and torque limitation of the engine usually results in partial displacement operation of the hydraulic pumps, leading to lower volumetric efficiency. Consequently, during high-power demanding cycles the system operates with two of its most important components at low efficiency conditions.

Literature has been consistent with two approaches to improve fuel efficiency of off-road vehicles. When possible, hybridization is a good solution to improve engine efficiency and to reduce the maximum engine power requirement. In addition, a lot of focus has also been given to minimization or elimination of throttling losses with solutions such as independent metering and de-centralized displacement-controlled (DC) architectures.

Among the most successful systems proposed in literature are the ones that can achieve both a better engine management and a reduction of throttling losses. Vukovic, Sgro, and Murrenhoff⁴⁻⁶ proposed the STEAM excavator, which uses two accumulators to maintain three different pressure levels in the machine. The concept is known as common pressure rails. On/off valves are used to connected different pressure levels to each actuator chamber. A controller was developed to select the proper pressure level and chamber connection with the goal of minimizing throttling losses and the actuator motion is controlled with proportional valves. The large accumulators adopted in the STEAM allow meeting high flow demands decoupling the actuators flow request from the engine speed. A 55% increase in efficiency during an air grading cycle was reported in the mentioned work.

In a similar fashion, Heybroek and Sahlman⁷ also propose different pressure rails with large accumulators to minimize throttling losses and decouple flow request from engine speed. In their proposal, two pressure levels are used in combination with multi-chamber cylinder with 4 chambers, achieving up to 33% reduction in fuel consumption for a truck-loading cycle.

The author’s research lab has followed the de-centralized approach to eliminate completely throttling losses. It has been shown by Zimmermann, Busquets and Ivantysynova that only by replacing the state-of-the-art hydraulics with displacement control actuation it is possible to reduce fuel consumption by 40%⁸. Later, the machine was modified to a series-parallel hybrid solution and achieved extra 10% in fuel savings in simulation⁹. However, this solution keeps the DC pumps mechanically connected to the engine shaft. In this scenario, whenever the engine speed is kept constant and the flow request is low, the pumps are forced to operate at low displacements where the volumetric efficiency is low. Williamson⁴ also proposed different pressure rail configurations with the goal of minimizing throttling losses and the actuator motion is controlled with proportional valves. The large accumulators adopted in the STEAM allow meeting high flow demands decoupling the actuators flow request from the engine speed. A 55% increase in efficiency during an air grading cycle was reported in the mentioned work.

With this approach, DC pumps and engine speeds can be controlled independently, leading to fuel consumption improvements of up to 60% when compared to the baseline Load-Sensing system or extra 11% when compared to the previous version of the hybrid system.
Despite the reduction in fuel consumption achieved in the past mentioned works, it must be considered that the market has showed a limited acceptance of DC based solutions. Besides implementation costs, there are challenges associated with reproducing all the machine requirements and expert operator feelings that are achieved by the current state-of-the art hydraulic architectures, such as load-sensing and open-center systems.

To overcome these limitations, this paper proposes a solution that can be considered as less disruptive when compared to the above-mentioned work, but that is able to achieve reduction in fuel consumption with minimal changes in the system dynamic response to operator commands and without compromising the market acceptance of the hydraulic system architecture. The rationale is to design a system with high engine-hydraulics integration with limited implementation costs. In addition, the proposed solution aims at keeping the same motion control quality and user feeling of the state-of-the-art hydraulic implements.

The paper structure is as follows. In the next section, the reference machine is introduced and followed by a description of the proposed solution. Then, the modeling approach used to model the baseline load-sensing system is described. Next, the reference working cycle used in this study is presented and simulation results are validated against measurements. Following the model validation, the design of the proposed transmission is discussed, and the simulation approach used to model the transmission is presented. Afterwards, the proposed power-management controller is presented, followed by a discussion on the possible integration of the swing actuator in the transmission. Lastly, simulation results are presented and discussed, followed by conclusions.

2. Nomenclature

- $\theta_{\text{cmd}}$: Desired primary unit displacement.
- $k_p$: PGT standing gear ratio.
- $M_{\text{cond}}$: Desired primary unit torque.
- $M_{\text{p}}$: Charge pump torque.
- $M_{\text{c}}$: Commanded engine torque.
- $M_{\text{e}}$: LS pump effective torque.
- $n_1$: Primary unit speed.
- $n_{\text{ref}}$: Secondary unit reference speed.
- $n_{\text{carrier}}$: Carrier gear speed in rpm.
- $n_s$: Engine speed.
- $n_{\text{ring}}$: Ring gear speed in rpm.
- $n_{\text{sun}}$: Sun gear speed in rpm.
- $P_{\text{c,cmd}}$: Commanded engine power.
- $P_{\text{hp}}$: Pressure in the transmission high-pressure side.
- $P_{\text{ls}}$: Pressure in the load-sensing line.
- $P_{\text{out}}$: LS pump outlet pressure.
- $Q_{\text{A}}$: Flow from/to the control valve from the A line.
- $Q_{\text{B}}$: Flow from/to the control valve from the B line.
- $Q_s$: LS pump effective flow.

3. Reference Machine

The reference machine used in this study is a Bobcat 435 mini-excavator, which in its baseline version uses a post-compensated load-sensing (LS) system with a single pump, like the one shown in Fig. 1. The main machine parameters are shown in Table 1.

<table>
<thead>
<tr>
<th>Machine Parameter</th>
<th>Symbol</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine Power</td>
<td>$P_e$</td>
<td>36 kW</td>
</tr>
<tr>
<td>Operating Weight</td>
<td>—</td>
<td>5 ton.</td>
</tr>
<tr>
<td>Engine Speed</td>
<td>$n_e$</td>
<td>2,400 rpm</td>
</tr>
<tr>
<td>LS Pump Displacement</td>
<td>$V_p$</td>
<td>48 cm$^3$/rev</td>
</tr>
<tr>
<td>LS Pump Margin</td>
<td>$s$</td>
<td>23 bar</td>
</tr>
<tr>
<td>LS Compensator Margin</td>
<td>$s_c$</td>
<td>4 bar</td>
</tr>
</tbody>
</table>

In this architecture, the highest user pressure is picked-up by the LS line through each actuator pressure compensator. The pump adjusts its displacement to keep its outlet pressure at the level of the LS line plus a fixed margin ($s$). Pressure compensators ensure a constant pressure differential across each flow control valve. This pressure differential is equal to $s-\delta$, and ensures that the flow is directly proportional to the valve opening area, therefore eliminating any load interference on actuator speed. A fixed orifice bleeds out any trapped pressure in the LS line, while a relief valve in the same line limits the pump maximum pressure. Pilot lines and local actuator relief valves are omitted in Fig. 1 for clarity.

4. Proposed Solution

The proposed solution consists in using a power-split transmission between the engine and the load-sensing pump, like shown in Fig. 2. The idea is to improve the overall system fuel consumption, without major changes to the hydraulic actuation of the baseline machine. In addition, another solution including a hybrid swing integrated in the transmission hydraulic path using secondary control will also be introduced and analyzed later in this paper. Despite the proposed solution can be applied with any centralized fluid power architecture, this work demonstrates its application on an excavator equipped with a LS system. It should be noted that a LS system is used here exclusively because it was the original configuration of the baseline machine, for which data was available. However, implementation with an open-center system is also possible.

The widely known power-split transmission has the characteristic of a series hybrid, in which engine and pump speeds are completely decoupled, but with a higher efficiency, since most of the power is still transmitted mechanically through the planetary gear train (PGT). Another advantage is on the size of the transmission units needed. While a series hybrid transmission would have both units sized for maximum system power, the selected power-split system requires smaller units since it relies on mechanical power transmission as well.

The planetary gear used has three shafts. One can be connected to the engine, one the primary hydraulic unit and another connected to the main user, in this case is a LS pump.

4.1 Transmission Power Flow

At the PGT, the torque relationship between the shafts is constant. On the other hand, the speed relationship between the three shafts is given by
where \( i_0 \) is the PGT standing gear ratio. Therefore, The PGT gear configuration enables independent speed control of 2 out of 3 shafts, as clear from the expression stated in equation (1). Consequently, power can be split between two paths according to the ratio of the selected speeds. In this way, while part of the engine power is transmitted to the LS pump through the mechanical path, the remaining power is transmitted through the hydrostatic path of the transmission. The ratio between inlet and outlet flows in the transmission hydraulic path determines the accumulator state-of-charge, allowing for energy storage/release.

With this approach, the primary unit is used to load the engine with the desired torque, while the secondary unit uses secondary control to regulate the speed of the LS pump. The engine speed is then controlled by adjusting the engine throttle, with a setpoint determined by a power-management controller. Therefore, compared to the original system the engine is operated in a much steadier way, with nearly constant torque and speed, while the accumulator and the secondary unit handle the severe load transients that occur during a digging operation. In addition, the output shaft speed can be quickly adjusted over time such that the LS pump is forced to continuously operate close to its maximum displacement, hence increasing its volumetric efficiency. Additionally, when the mechanically transmitted power is higher than the demand from the LS pump, the secondary unit serves as hydrostatic brake, also charging the accumulator. Additional relevant information on the behavior of hydraulic power-split transmissions can be found in these references [9], [10].

Besides the freedom in engine speed selection, the proposed architecture also brings the typical benefits of hybrid systems, such as the potential for engine downsizing. The added energy storage device allows the designer to use a smaller engine such that it is constantly operated closer to its maximum torque, where engine efficiency is maximum, while peak power demand is achieved with a hydraulic boost from the hydraulic accumulator.

5. Simulation Model

The design of the proposed transmission is based on a numerical simulation model that can reproduce the behavior of the baseline LS system and achieve a good estimate of its power consumption. Such model is used to characterize the loads that would be seen by a power-split transmission, and to provide important information in the design process. In addition, it is also used to obtain a fair comparison between proposed and baseline systems.

5.1 Load-Sensing Implements Model

The Load-Sensing system model is developed in Matlab/Simulink and it can be divided into six different sub-models as shown in the top-level block diagram presented in Fig. 3. Since the kinematic model has been presented and validated in previous publications [11], [12], its description will be left out of this paper, but the reader is encouraged to refer to mentioned references for more details. The operator model is a PI controller that generates valve position commands based on a reference cycle profile. Additional inputs to the model include the external load \( F_{ext} \), which is estimated based on measured pressures, and the measured engine speed \( n_e \). The operator model is a PI controller that generates valve position commands based on a reference cycle profile. Additional inputs to the model include the external load \( F_{ext} \), which is estimated based on measured pressures, and the measured engine speed \( n_e \). The hydraulic valves are modeled with the standard orifice equation and linear first and second-order models are used to account for components dynamics. For brevity, details of the modeling equations are left out of this paper but can be found in the previous work by the authors [13].
6. Reference Cycle and Model Validation

To verify the validity of the model of the hydraulic system, comparisons on measured kinematic quantities and pressures of the simulated machine are presented. A 90 degrees truck-loading cycle performed by an expert operator has been selected. In this cycle, the machine digs into the ground, rotates 90 degrees while raising the boom and dumps the bucket content into a truck. The rationale for the selection of this cycle is twofold: first, it has been widely used both in academia and industry for fuel consumption evaluation of excavators. Second, it reproduces the most demanding operating conditions for the machine, in which four actuators are used simultaneously with maximum power demand.

Based on the available data, it is chosen to look at the actuators measured positions and pump outlet pressure. In short, by comparing measured and simulated positions, it is guaranteed that the pump is delivering a similar flow rate in both experiments and simulation. Additionally, the validation of the pump outlet pressure ensures that the loads and throttling losses are well captured with the model. Therefore, it is inferred that the developed model is representative of the LS pump power consumption.

Fig. 4 Simulated actuators positions with LS model

7. Planetary Gear Selection and System Sizing

7.1 Engine Downsizing

Although the proposed system allows any engine to be operated at the desired combination of torque/ and speed, fuel efficiency can be further improved when smaller engines are used. This because a downsized engine will tend to operate more often close to its peak torque, where fuel efficiency is higher and because the hydraulic accumulator can be used to energy storage/release. Therefore, the starting point in the design process is to evaluate the required engine size as well as the optimal engine torque/speed, such that the transmission can be designed to operate the engine at the desired operating point.

To evaluate the required engine size, the developed model is used to evaluate the typical LS pump power demand over a highly demanding cycle, as shown in the top plot of Fig. 5. From these results, it is clear the large variations in power demand over time and
therefore it is possible to size the engine to provide only the average power demand and use the energy storage device to handle the heavy transients. For the cycle in analysis, the average demand is 22.8 kW, which is 37.5% lower than the original engine rated power of 36.5 kW. However, such a large engine size reduction is not desired for three reasons: first, there is no guarantee that the cycle in analysis - even if it was an aggressive expert operator cycle - is the most demanding a machine will ever face in real life, meaning different conditions and operators could require a larger average power. Second, this analysis considers only the LS power demand and does not leave any extra power to accelerate the engine, for example should its speed drop during a sudden load increase. Lastly, other functions in the machine like the tracks usually require a constant power supply, and therefore it is not always possible to rely on the accumulator to achieve a higher power supply.

The aforementioned reasons leave the room for the question on how to properly select the engine size. To answer this, a proposed reasonable approach can be derived from the engine fuel consumption map. Diesel engines typically operate with higher efficiency (or break-specific fuel consumption) near its peak torque, and not near its peak power. Therefore, an engine can be selected to deliver the average power of the studied cycle at its peak efficiency point. This ensures optimal engine operation for a typical cycle, while also leaving some room to operate at higher power demands should the operator require it. Nevertheless, this approach still requires ensuring all the machine requirements are met, such as the power requirements of functions with constant demand like the tracks.

A typical Diesel engine fuel consumption map is shown in Fig. 6. In this work, this map is scaled to the specifications of the original engine in the baseline machine. The Figure shows that the peak engine power is at 2300 rpm, while the peak efficiency (and peak torque) is at around 1800 rpm. Therefore, assuming the smaller engine will have similar characteristics, it is desired to select an engine capable of providing 22.8 kW at 1800 rpm, resulting in an engine with a maximum power of 29.2 kW, a 20% reduction when compared to the baseline machine. Evidently, different engines will have different fuel consumption maps and the proposed approach may not be as straightforward as presented here, requiring some iterations and analysis of different engines. Nevertheless, it is reasonable to expect that other engines will share similar characteristics. This because according to the work by Vukovic, Leifeld and Murrenhoff diesel engines fuel consumption will vary similarly with respect to the operating points in different engines.

Besides operating the engine at its peak efficiency, the proposed approach also leaves the possibility of achieving higher power demands. The selected engine can achieve up to 28% more power than what is seen on average in the analyzed cycle, therefore ensuring that more demanding cycles can also be met, and that the accumulator can be quickly charged. In addition, the reduction in power availability to the tracks is not as significant.

7.2 Planetary Gear Configuration Selection
With the engine size and preferred engine speed defined, it is desired to select the transmission components. A challenge when designing any power-split transmission is to select the most appropriate configuration, meaning to which one of the PGT shafts should engine, primary and secondary units be connected. Such selection is usually done such that the primary unit speed is as close as possible to the full mechanical point, where the primary unit speed is zero. Therefore, in this condition, most of the power is still be transmitted mechanically through the PGT. The rationale for that approach is based on the fact that mechanical power transmission tends to be more efficient than using a hydrostatic path, where the power flow is subject to total efficiencies of 2 hydrostatic units. In the case of the proposed application, the job of placing the full mechanical point is simplified given the fact that the baseline machine runs with a constant engine speed. In this way, assuming no changes will be made to the LS pump size, it is straightforward to select a configuration that will place the full mechanical point at an engine speed of 1800 rpm and a pump speed of 2400 rpm. This can be done in an iterative process using equation (1) in which different gear ratios and configurations can be plugged in until the solution with the primary unit speed as close to zero as possible is found for the given operating condition. It should be noted that different combinations of configurations and gear ratios may lead to similar primary unit speeds. In that case, the PGT that requires smaller hydrostatic units is
preferred, which in the studied case led to the selection of the configuration shown in Fig. 2. In this case, the carrier is connected to the engine, the sun is connected to the primary unit and ring is connected to the load. The sizing procedure for the hydrostatic units and accumulator is left out of this paper for brevity, but the reader is invited to refer to this previous publication\(^{(8)}\) for details on the sizing method. However, it is important to highlight that the system is sized such that the power of the original system can always be delivered to the LS pump, even when the accumulator pressure is minimum. Consequently, the proposed powertrain will have higher power availability than the baseline machine. The final sizing is presented in Table 2. The table contains also other key parameters of the system that will be detailed in the next sections.

Table 2 Sizing results

<table>
<thead>
<tr>
<th>Machine Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine Power</td>
<td>29.2 kW</td>
</tr>
<tr>
<td>Planetary Gear Standing Ratio</td>
<td>−2.5</td>
</tr>
<tr>
<td>Primary Unit Displacement</td>
<td>12 (\text{cm}^3/\text{rev})</td>
</tr>
<tr>
<td>Secondary Unit Displacement</td>
<td>21 (\text{cm}^3/\text{rev})</td>
</tr>
<tr>
<td>HP Accumulator Volume</td>
<td>6 L</td>
</tr>
<tr>
<td>HP Accumulator Pre-Charge</td>
<td>180 bar</td>
</tr>
</tbody>
</table>

8. Engine and Transmission Models

To model the transmission, empirically derived loss models were used to evaluate the losses of each hydrostatic unit given its speed, fractional displacement, and pressure differential. Similarly, a loss map provided by a partner company was also used to estimate the PGT losses. The provided loss model considers both meshing losses, which are a function of the power transmission between gears and churning losses. Therefore, PGT losses are a function of torque and speed of at least two of its axes. Temperature is assumed constant for losses evaluation purposes. The modeling approach can be found with more details in the previous works by the authors\(^{(8),(13)}\). Although a validation of the transmission model is not available, the approach follows validated modeling strategy used by Kumar and Ivantysynova\(^{(16)}\). Lastly, the engine fuel consumption was obtained with the fuel consumption map presented in Fig. 6.

9. Supervisory and Power-Management Controller

The proposed architecture has the advantage of independent control of engine torque, speed, and the LS pump speed. The purpose of the supervisory power-management controller is to control these states such that the state-of-charge of the accumulator is kept within the desired range, while ensuring that the LS pump can deliver the required flow in the most efficient way possible.

The controller is designed such that the LS pump shaft speed is varied over time. The rationale for that is twofold: first, it forces the pump to operate at higher displacements, making it more efficient. Second, it allows an increase in the speed of the primary unit, which results in a quicker charge of the accumulator during low power demand moments of the cycle. To achieve that, it is proposed to use a closed-loop pressure control to determine the desired output shaft speed. Therefore:

\[
 n_{2,\text{ref}} = k_{2,1}(p_{\text{ref}} - p_2), \quad p_{\text{ref}} = p_2 + s 
\]

where \(k_{2,1}\) is a proportional gain. It is worth mentioning that the proposed system still considers a typical LS pump with hydro-mechanical pressure feedback. In this scenario, the proposed approach has the following advantages:

(a) With the proper gain selection, the reference speed will always be close to the minimum required to ensure that \(p_2/p_{\text{ref}}\), which then forces the LS pump to operate at higher displacements.

(b) Since two independent controllers act on different control inputs, the proposed controller achieves a faster response when guaranteeing good tracking of the desired pump outlet pressure.

(c) E-LS can be implemented without any changes to the LS pump, since the supervisory controller can automatically reduce the pump margin by forcing the LS pump to flow saturate.

(d) The only sensors required for implementation are LS pressure and pump outlet pressure.

To ensure that the accumulator is properly charged, the supervisory controller also changes the commanded power to the engine. Using the derived simulation model, the following set of rules is proposed:

\[
 P_{\text{cmd}} = \begin{cases} 
 5 \text{ kW}, & P_{\text{pp}} > 310 \text{ bar} \\
 17 \text{ kW}, & 275 \leq P_{\text{pp}} \leq 310 \text{ bar} \\
 27 \text{ kW}, & P_{\text{pp}} < 275 \text{ bar} 
\end{cases}
\]

The idea behind this approach is to command an engine power above the average requirement for two reasons. First, because the engine now also needs to provide power to account for the transmission losses. Second, because it guarantees a quick recharge of the accumulator ensuring power availability. Once the pressure reaches a certain level, then the power command is lowered to avoid constant high-pressure in the HP line, which can increase the volumetric losses in the units. On the other hand, it should be said that this simplified approach is most-likely not optimal, and improvements could be made. For example, dynamic programming...
can be used to find the optimal control solution which can be used as reference to derive a new formulation for equation (3), therefore ensuring near-optimal control of the accumulator state-of-charge.

Once the desired engine power is selected, an online optimization algorithm is used to find the optimal combination of torque and speed that delivers the request. Based on the desired torque engine torque, the required primary unit is torque is

\[ M_{1, \text{cmd}} = \frac{M_{1, \text{cmd}} - M_{e}}{i_0 - 1} \]  \hspace{1cm} (4)

From the desired unit torque, an inverse loss model is used to obtain the displacement \( \beta_{1, \text{cmd}} = f(M_{1, \text{cmd}}, n_1, \rho_{\text{hp}}) \) needed to achieve the required torque. To avoid sudden changes in the engine load, the displacement command signal to the primary unit is filtered with a low-pass filter. In addition, the displacement of the secondary unit is limited in the case the pressure in the HP line drops below the minimum threshold of 200 bar. A top-level block diagram of the control approach is shown in Fig. 7.

10. Implementation with Hybrid Swing

Another benefit of hybridization is the storage of recovered energy, which is not possible with traditional LS actuation. Based on the proposed architecture, a straightforward solution is to integrate the swing motor in the hydraulic path of the transmission, as presented in Fig. 8. The swing is selected since it is the only actuator that can be switched to secondary control. The benefits of such approach are twofold: first, it allows the recovery of kinetic energy from the swing system. Second, it eliminates the throttling losses associated with the swing function in the LS line. Therefore, it is proposed to replace the fixed displacement swing motor with a variable displacement motor with secondary control. To quantify the benefits of the idea, the previously described model was modified to include the swing motor in the hydrostatic path of the transmission.

11. Results

Before evaluating the fuel consumption improvements of the proposed system, it appropriate to verify that it can deliver the same performance of the baseline machine. Ideally, this would be done experimentally by evaluating the machine productivity in tons/liter, but in this simulation study this is done by evaluating how well the proposed architectures follow the baseline cycle. The simulated outlet pump pressures are shown in Fig. 9 from which is seen that for most of the cycle, the LS pump outlet pressure is the same. This shows that even though the pump speed is being varied over time, it does not change the ability of the LS pump to track the desired load-pressure and therefore no significant changes in operator perception should be seen.

The discrepancies seen at 104s and 114s are most likely due to small differences in the simulated operator command, which can differ from simulation to simulation and may cause different chambers to connect to the LS line. Nevertheless, the results in Table 3 show that all three simulated systems present a nearly identical performance in terms of delivered energy (obtained by integration of positive actuator power), therefore demonstrating also, good cycle tracking in all cases.

A comparison between the two proposed architectures is shown in Fig. 10 and Fig. 11. In Fig. 10, it is seen that some swing kinetic energy is possible, although it is not that significant due to unit losses and friction. Perhaps, more benefits would be seen in machines with larger inertia, but nevertheless it is clear from the simulated pressures that recovery occurs, especially at 140s and at 152s when the pressure is clearly higher in the system with integrated swing, even though both systems were using the same power-management strategy. Another important aspect to be highlighted in the transmission pressures is that both hybrid systems started and ended the cycle with nearly the same pressure. Therefore, no free energy coming from an initial condition in the accumulator pressure was used to accomplish
the cycle and reduce fuel consumption estimation. A significant change in engine operating conditions allows the prime mover to always operate at peak efficiency. The average displacement of the supply pump is also higher due to the implemented controllers, thereby increasing their efficiency. Nevertheless, it should be noted that these gains are not that significant given the fact that the pump in the baseline LS system also

The LS pump speed is shown in Fig. 11. When the swing motor is in use, the pump speed is lower in the system with integrated swing (i.e., at 110s and 116s) therefore demonstrating the ability of the proposed controller to reduce properly slow-down the pump speed under low flow demand. It must also be pointed out that the speed controller used is able to track the reference speed, despite low inertia and sudden load changes. Nevertheless, some spikes in speed are still seen (i.e., at 120s) when the LS pressure suddenly decreases, quickly reducing the load in the shaft.

A summary of the simulation results is shown in Table 3. Results show that the proposed transmission can reduce the fuel consumption in up to 11.2%. This value can be extended up to 16.4% in the analyzed cycle when the swing is integrated in the hydrostatic path of the transmission, while respecting the same expert operator cycle.

The results can be supported by Fig. 12 which shows the engine operating points, with the contours representing break specific fuel consumption.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Baseline</th>
<th>PS Hybrid</th>
<th>PS+Swing Hybrid</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine Downsizing</td>
<td>—</td>
<td>−20%</td>
<td>−20%</td>
</tr>
<tr>
<td>Average LS Pump Disp.</td>
<td>85.3%</td>
<td>93.7%</td>
<td>90.2%</td>
</tr>
<tr>
<td>Delivered Work</td>
<td>388.6 kJ</td>
<td>377.0 kJ</td>
<td>388.2 kJ</td>
</tr>
<tr>
<td>Fuel Cons.</td>
<td>91.27 g</td>
<td>81.08 g</td>
<td>76.36 g</td>
</tr>
<tr>
<td>Change in Fuel Cons.</td>
<td>—</td>
<td>−11.2%</td>
<td>−16.4%</td>
</tr>
</tbody>
</table>

Fig. 10 Simulated transmission pressure differential and swing speed

Fig. 11 Simulated transmission speeds for power-split hybrid (top) and power-slit + swing hybrid (bottom)

Table 3 Summary of simulation results

Fig. 12 Simulated engine operation of baseline system (top), Power-split Hybrid (middle) and Power-split + Swing hybrid (bottom).
operated at a high average displacement. Lastly, the integration of the swing in the transmission allows some energy recovery of the kinetic energy available, while also eliminating the associated throttling losses.

12. Conclusions and Future Work

This work introduced a novel hydraulic architecture by combining an efficient power-split transmission with state-of-the-art hydraulic actuation for high engine/hydraulics integration in an excavator. A simulation model has been developed and validated for the baseline LS system of the reference machine, a 5 ton mini excavator. The model was then used to design and evaluate the benefits of the proposed transmission in simulation.

Simulation results showed the feasibility of developing a design with high integration between engine and hydraulics for off-road vehicles. By allowing the engine to operate at its peak efficiency, up to 11.2% improvement in fuel consumption can be achieved, while still maintaining the reliability and user experience of state-of-the-art hydraulics. The proposed architecture should also result in a lower up-front cost when compared to more radical approaches. It should be noted that, even though this analysis considered a load-sensing architecture, similar approach could also be followed with different architectures such as open-center systems.

In addition, the proposed system can be especially attractive to engine manufacturers, as it would allow an engine design focused on a single point of operation. In this scenario, higher improvements can be expected.

Another advantage of the proposed system is the possibility of swing integration, which increases efficiency improvements by reducing throttling losses and allowing energy recovery. When such approach is used, improvements of 16.4% in fuel consumption are estimated for the reference cycle.

As future work, it can be studied the impact of the proposed transmission on cooling requirements and its effects on the system fuel savings. In addition, Different accumulator charging strategies can be tested to improve even further the system efficiency.

References