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Downsizing the electric machines of energy-efficient electro-hydraulic drives for mobile hydraulics

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Abstract. The poor energy efficiency of state-of-the-art mobile hydraulics affects the carbon dioxide released into the atmosphere and the operating costs. These crucial factors require urgent improvements that can be addressed by the electrification of fluid power. This approach has already generated electro-hydraulic drives that remove flow throttling and enable energy recovery. However, the entire power managed by the actuators of conventional systems must pass through the electric machines. This characteristic is unfeasible for medium-to-high power applications since they need electric motors and electronics with high power ratings and large onboard generation of electricity. Thus, this paper applies to a hydraulic excavator's boom the idea of splitting the power being transferred to/from the actuator between the hydraulic and electric domains (*i.e.*, a centralized hydraulic power supply is involved). The objective is downsizing the power rating of the boom's electric components while maintaining the high-power output of the hydraulic actuator. The results show the expected behavior of the hybrid excavator in terms of motion control, but only 57% of the boom's peak power is now exchanged electrically. The resulting electric machine with 61% downsizing favors the system's cost and compactness supporting the electrification process that is aligned with the low-carbon economy.

1. Introduction

Mobile hydraulics is a widely used technology in fields such as construction or earth-moving with hydraulic excavators being the most popular application. Despite the multiple system architectures developed over the last decades [1-2] environmental and economic reasons call for improvements of their energy efficiency. The average efficiency of state-of-the-art machines is, in fact, poor due to functional flow throttling in control valves and absence of energy recovery. Several examples of this unfortunate behavior are available in the literature. For instance, approximately 12.5% efficiency was calculated for the hydraulics alone of a 5-ton load-sensing excavator [3], where about 40% of the total power dissipations take place in the control valves.

Since many countries released updated climate pledges to reach "carbon neutrality" and raise the share of non-fossil energy in primary energy use, several emission regulations for combustion engines have been introduced recently (*e.g.*, EPA Tier 4 Final in the U.S.A. and Stage V in Europe). Reducing fuel consumption in mobile hydraulics is also an economic issue due to the high price of diesel. Thus, research has been conducted for hydraulic excavators to mitigate the sharp spikes that characterize the power requirements placed on the combustion engine and to enable energy recovery due to the potential savings achievable for the boom and swing actuators [4]. Using hybrid systems based on different energy storage devices is key [5]. Various layouts were introduced involving batteries and/or supercapacitors [6] or hydro-pneumatic accumulators [7], despite functional power losses in control valves are still commonplace for linear actuators. So, alternative valveless architectures were evaluated. A nonhybrid, displacement-controlled excavator required only 50% of the input energy used by the load-sensing counterpart to complete the same digging cycle [3]. Additional fuel savings around 17% were then predicted for an updated hybrid version of the same machine [8], even if the costly set-up with, at least, 4 overcenter pumps and their inefficient behavior at partial displacement are issues. Moreover, a hybrid active-passive system was developed for the boom drive to hydraulically support an electro-mechanical actuator [9] that achieves good efficiency but might experience severe limitations in heavy-duty operations [10].

A different research direction concerns the electrification of fluid power machines using speed-controlled pumps driven by electric motors to control the motion of hydraulic actuators efficiently (*i.e.*, both throttleless actuation and energy recovery take place). These individual, electro-hydraulic drives can successfully replace conventional control valves in terms of both dynamic response [11] and energy efficiency [12-13]. Passive load-holding devices that do not affect efficiency are also available when needed for safety reasons [14-15]. Electro-hydraulic drives can deal with multi-actuator systems (*e.g.*, offshore drilling applications [16]) and were already applied to the main actuators of a 9-ton excavator [17] increasing the efficiency from 5.4% (load-sensing baseline) up to 8.8-14.5% depending on the system design. However, the main downside is about the power managed by the actuators that must pass through the electric motors in conventional solutions. This approach becomes critical, or even infeasible, if the power level rises to several kilowatts; a bulky and expensive arrangement for the electric subsystem is needed (motors, drives, generator, and energy storage device) and an enormous quantity of electricity must be generated onboard. Therefore, hybrid actuators with self-contained energy storage can downsize the rated power of the electric machines [18], but additional components are needed. Downsizing the installed electric power can also be done by connecting the inlet port of speed-controlled pumps to a pressure rail supplied by a centralized pump [19-20] resulting in power being transferred to/from the actuators both hydraulically and electrically.

For this reason, this research paper applies to a hydraulic excavator the idea of splitting the power being transferred to/from the actuators between the hydraulic and electric domains (a centralized hydraulic power supply is involved). It proposes a hybrid system architecture that avoids functional power losses in control valves, recovers energy, and downsizes the power ratings of the electric components related to the boom actuator while maintaining its high-power output. This novel approach retains the efficiency benefits and control performance of conventional speed control but reduces the amount of electric power used onboard in favor of the system's cost, compactness, and power density.

2. System architectures

It is well-know that excavators often perform operations where the boom, arm, bucket, and swing actuators are operated simultaneously [21]. Providing their independent control is therefore paramount. The following system architectures achieve this goal while ensuring energy-efficient actuation that removes flow throttling, enables energy recovery, and supports the machine electrification.

2.1. Reference energy-efficient excavator

In the first system architecture considered in this paper (figure 1), the internal combustion engine (CE) drives an electric generator (G), a fixed-displacement charge pump (P_1), and a variable-displacement pump (P_2). The latter unit can be disconnected by a clutch to avoid unnecessary power losses since it supplies those valve-controlled actuators that are seldomly used (the track, blade, and boom-swing actuators). The combination of the charge pump, pressure-relief valve (RV), and hydro-pneumatic accumulator (A) creates a low-pressure line. The generator charges a supercapacitor-based battery (S) that feeds four electric motors (EMs). EM_4 directly drives the swing, while EM_1, EM_2, and EM_3 drive hydraulic motors/pumps that are part of the electro-hydraulic actuators of the boom, arm, and bucket. The pump/motor P_3 sets the boom velocity, whereas the 4/3 nonproportional direction control valve (DCV) defines the direction of the motion (its return port to reservoir contains a preloaded check valve to ensure a minimum pressure that avoids cavitation and increases the actuator's stiffness). The open-circuit configuration for the boom was chosen to favor the energy efficiency that is superior compared to the closed-circuit alternative [13] where both actuator's chambers are permanently connected to the pump/motor. The arm and bucket use two pumps/motors each because this configuration is free of instable mode switching that might take place when the high-pressure side switches from one actuator chamber to the other [22-23]. Check valves (CVs) connect the charge line to the actuators' chambers to avoid cavitation in the long transmission lines going to the hydraulic cylinders.

This overall design approach for the excavator's hydraulics is convenient to electrify the machine due to the reduced number of components and the simple system design and control. Nevertheless, all the actuators' power must be managed by the electric machines that need to ensure sufficient power ratings to meet the peak requests. Such a solution might work fine for small excavators, but it becomes critical when the power level increases to several kilowatts. Bulky electric motors and costly electronic drives are, in fact, necessary. The electric power must be generated onboard requiring the installation of a powerful electric generator driven by the combustion engine and of a performant electro-chemical energy storage device. Those drawbacks pose a challenge to the dissemination of this technology: thus, they will be addressed by the system architecture presented in the following subsection.

2.2. Modified energy-efficient excavator

The second system architecture studied in this paper, that is a novel solution, resembles the previous one. The main actuators in figure 2 (arm, bucket, and swing) are still fully controlled by electric machines, while the boom's drive gets the support of a centralized, hydraulic power supply. The high-power demand of this actuator dictates, in fact, an unfortunate sizing of the electric motor EM_1 in conventional electro-hydraulic systems like the one in figure 1. Thus, the common rail can be connected to one port of the boom's pump/motor P_3 via the electrovalve EV_2. This feature can significantly lower

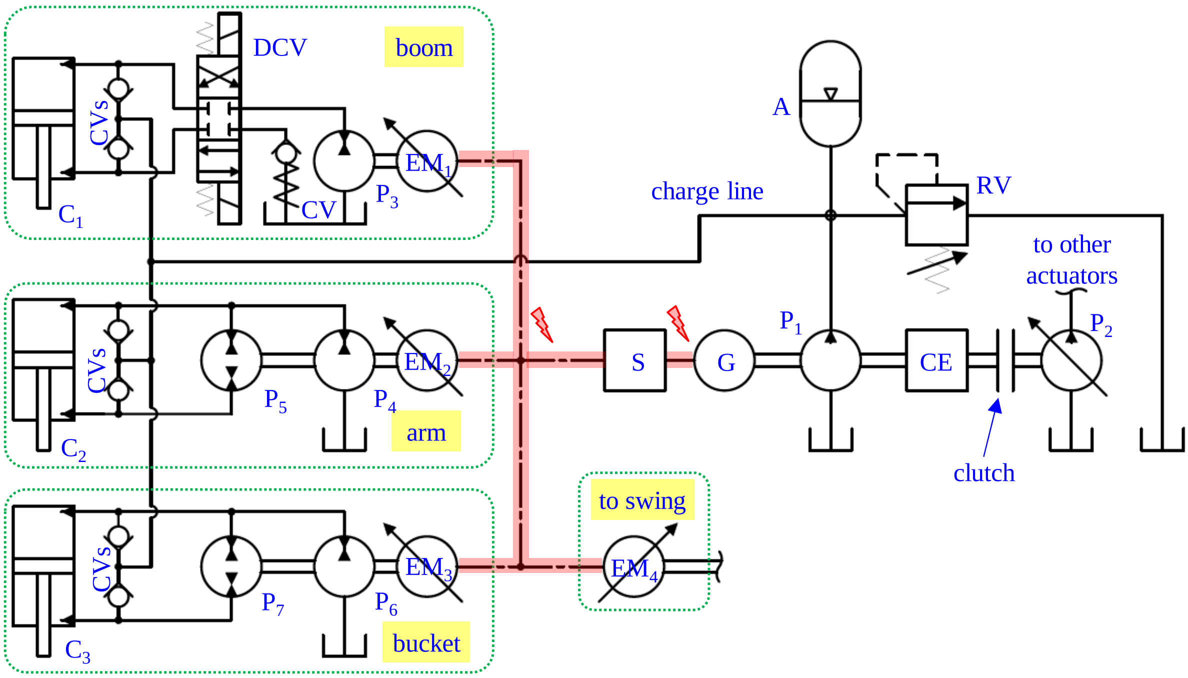

Figure 1. Simplified architecture of the reference excavator without a centralized power supply.

the maximum torque applied to EM_1 requiring, therefore, a lower power rating of this machine. The pump P_1 maintains the pressure in the common rail within an approximately constant range dictated by the intervention of the pilot-operated pressure-relief valve (RV_1). Its pilot stage is engaged/disengaged by the pressure switch (PS) that commands the vent valve inside RV_1. If the vent valve is de-energized, the pump's flowrate feeds the common rail that is also connected to a hydro-pneumatic accumulator (A_1) supporting the boom's flow requests (the electrovalve EV_1 isolates it during standstill to avoid leakages). In this system, the accumulator A_1 is not meant to sustain the effort of the combustion engine, even if this approach is possible (*i.e.*, the pump P_1 should be an overcenter unit, while RV_1 would only be a safety element). A small pump (P_8) replenishes the low-pressure charge line for the electro-hydraulic actuators. A compact accumulator (A_2) and a pressure-relief valve (RV_2) are part of this subsystem. The electric generator (G) and the supercapacitor (S) remain unchanged.

Therefore, the novel system architecture in figure 2 combines electric and hydraulic actuation in a complementary manner. The speed-controlled pump/motor of the boom actuator is connected to a centralized common rail, so that a substantial amount of power is transferred to the boom actuator hydraulically. The electric motor EM_1 still controls the motion of the boom, but it only manages a reduced amount of power (*i.e.*, its power rating decreases and less electric power is needed onboard).

3. System modeling

High-fidelity simulations have been performed using Simcenter Amesim®. The two models involved in this study reflect the architectures introduced before. They are developments of an experimentally validated model representative of a 7-ton excavator [24-25] that has been further updated with more advanced Amesim blocks [20]. A representative comparison of the two layouts is feasible and accurate because the same modeling approach has been adopted in both architectures under investigation. A brief description of these dynamic models is given below.

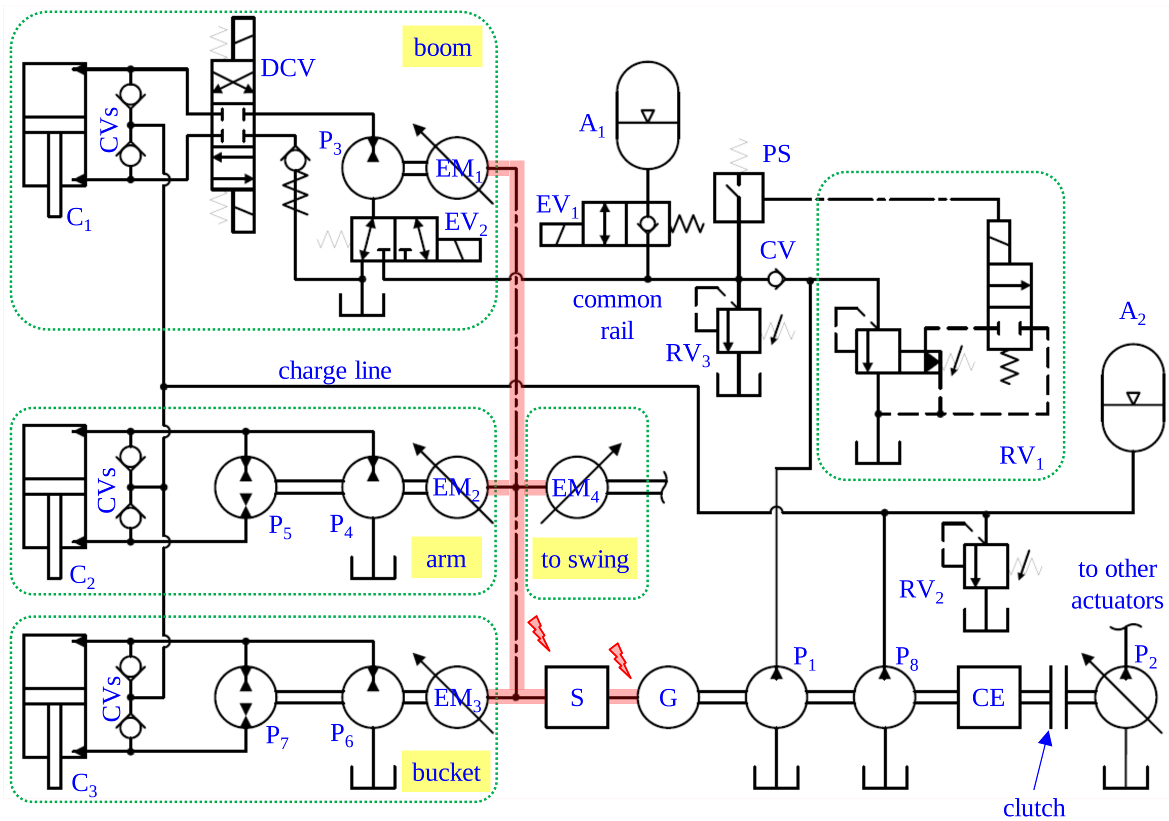

Figure 2. Simplified architecture of the modified excavator with a centralized power supply.

3.1. Reference energy-efficient excavator

With reference to figure 3, the arm kinematics has been simulated using the 2D Mechanical library. The model computes the digging force giving a realistic load condition due to the soil/bucket interaction and the adjustable filling factors of the bucket. The model also considers the variation of the cabin inertia as a function of the arm position. The engine model includes the map of the fuel consumption. Its speed varies according to the wide-open-throttle curve as a load is applied since the engine is set to high-idle at about 2500 rev/min. The electric power supply is simulated by a generator model with constant efficiency that is enabled by a logical switch according to the lower and upper threshold limits of the supercapacitor's state of charge. A quasi-static model simulates this energy storage device (transients and thermal effects are neglected). The linear actuators of the boom and arm are simulated by tailor-made supercomponents that account for the end-position cushioning. The swing model includes the hydraulic motor with integrated gearbox. Finally, the electro-hydraulic drives are simulated by the supercomponents shown in figure 4.

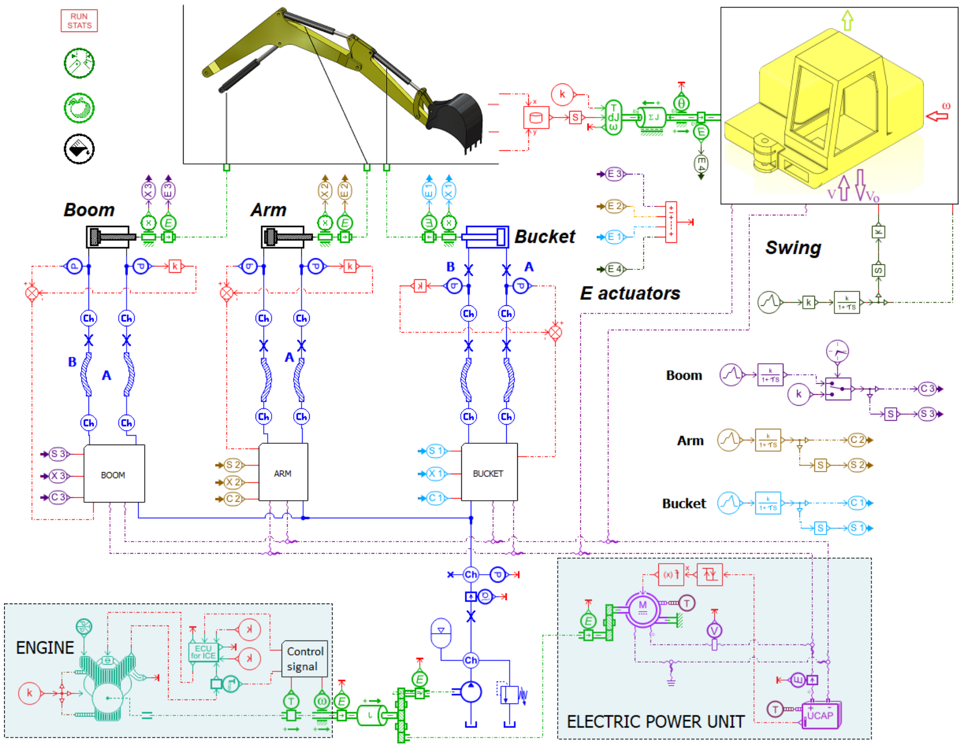

Figure 3. Amesim model of the reference excavator without a centralized power supply.

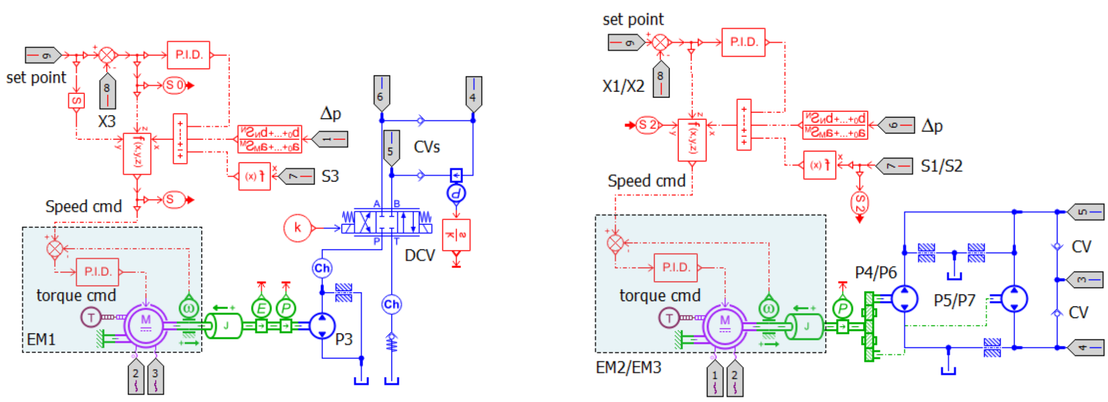

Figure 4. The electro-hydraulic drives of the boom (left), arm, and bucket for the reference layout.

Table 1. Main parameters of the reference layout.

Parameter	Value	Parameter	Value
Displacement charge pump P_1	6 cm^3/rev	Maximum power EM_1 (boom)	55 kW
Displacement pump P_3 (boom)	36.7 cm^3/rev	Maximum power EM_2 (arm)	21 kW
Displacement pump P_4 (arm)	10.6 cm^3/rev	Maximum power EM_3 (bucket)	21 kW
Displacement pump P_5 (arm)	16.1 cm^3/rev	Maximum power EM_4 (swing)	60 kW
Displacement pump P_6 (bucket)	16.1 cm^3/rev	Maximum power generator G	30 kW
Displacement pump P_7 (bucket)	23 cm^3/rev	Nominal voltage supercapacitor S	350 V
Diameter (bore/rod) actuator C_1	110/80 mm	Capacitance supercapacitor S	20 F
Diameter (bore/rod) actuator C_2	95/70 mm	Specific power supercapacitor S	12.3 kW/kg
Diameter (bore/rod) actuator C_3	85/55 mm	Pressure setting relief valve RV	5 bar
Maximum power engine CE	53 kW	Volume accumulator A	6 L
Torque time constant EMs	0.1 s	Preload pressure accumulator A	0.1 bar

The electric motors contain a functional model where power losses and torque limits are defined with look-up tables depending on the operating conditions. These machines are reversible and can work either as generators or as motors, where the same maps have been used in both scenarios. The models of the hydraulic pumps/motors consider the mechanical-hydraulic and the volumetric efficiencies (the latter is simulated by equivalent gaps). All valves are simulated by considering realistic flow-pressure drop characteristics, namely the check valves and the DCV in the boom's drive (left-hand side of figure 4). Finally, the transmission lines going to the actuators resemble the ones of the original 7.5-ton, load-sensing excavator [24]: the system has been sized to meet the performance of this commercialized valve-controlled machine taken as the starting point for this study. Table 1 lists the main parameters, where the maximum power of the electric motors refers to the S1 duty curve for continuous operations.

3.2. Modified energy-efficient excavator

The second Amesim model shares the abovementioned approach for the arm kinematics, combustion engine, electric power unit, actuators, and electro-hydraulic drives of the arm and bucket. The boom drive is modified to include the support of the centralized, hydraulic power supply (figure 5). Its model considers both the volumetric and mechanical-hydraulic efficiency of the pump as a function of pressure

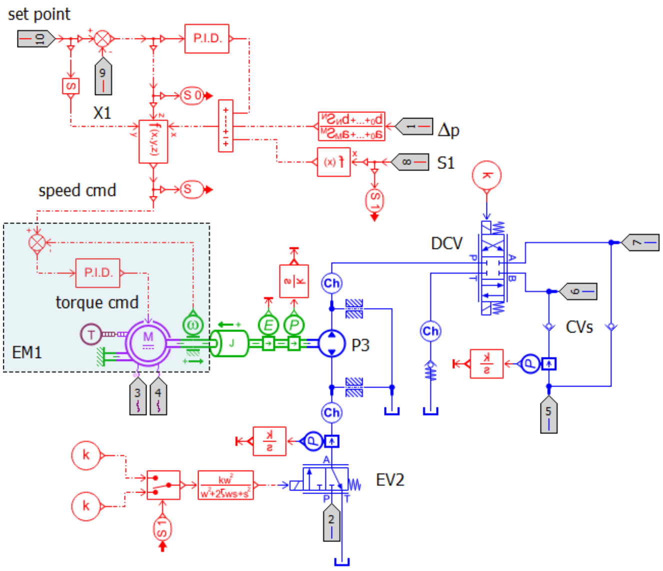

Figure 5. The electro-hydraulic drive of the boom for the modified layout.

and speed. The power dissipations in the valves are also accounted. The pressure-relief valve considers a flow-pressure characteristic of a typical component with nominal flow rate 200 L/min. The check valves can manage comparable flow rates and their flow-pressure characteristics are taken from manufacturer's catalogues. The pressure switch, simulated by a trigger, supplies the activation signal to the vent valve.

Table 2 lists the main parameters of the modified system architecture. The other key magnitudes not mentioned here keep the same value given in table 1 (*e.g.*, the power ratings of the electric motors dedicated to the arm, bucket and swing, or the pump displacements used in all the electro-hydraulic drives). It is worth mentioning that the maximum installed power of the boom's electric motor is now 21 kW due to the support of the centralized power supply (this is a 61% reduction compared to the value of 55 kW representative of the reference excavator).

4. System simulation

The dynamic models introduced above have been used as follows to compare the system architectures.

4.1. Working cycle and control algorithm

A trench digging cycle based on the JCMAS standard has been used to compare the systems' performance. The chosen soil type is the one labelled as GW by the Unified Soil Classification System (*i.e.*, "well graded gravel, sandy gravel, with little or no fines" with density of 2141 kg/m^3). It generates the maximum forces of the bucket and, therefore, the more severe load conditions. Figure 6 depicts the displacement of the actuators during one entire cycle. It should be noted that the boom velocity increases when the swing rotation is enabled; this feature derives from the valve-controlled excavator due to the intervention of the flow boost valve [24]. Significant positions of the excavator are shown on the right side of the figure, where the slope of the trench and the relative position of the bucket are also visible.

Closed-loop position control has been enforced to ensure fair comparisons. The swing's electric actuator only requires a PI-controller acting on the position error combined with a feedforward command based on the desired swing speed. The electro-hydraulic drives (boom, arm, and bucket) use the same control structure with the addition of artificial damping obtained via pressure feedback. The latter feature is key because the drives' energy-efficient nature dictates extremely low damping ratio for their version

Table 2. Main parameters of the modified layout.

Parameter	Value	Parameter	Value
Maximum power EM_1 (boom)	21 kW	Volume accumulator A_1	10 L
Displacement pump P_1	42 cm^3/rev	Preload pressure accumulator A_1	35 bar
Low-setting pressure switch PS	60 bar	High-setting pressure switch PS	70 bar

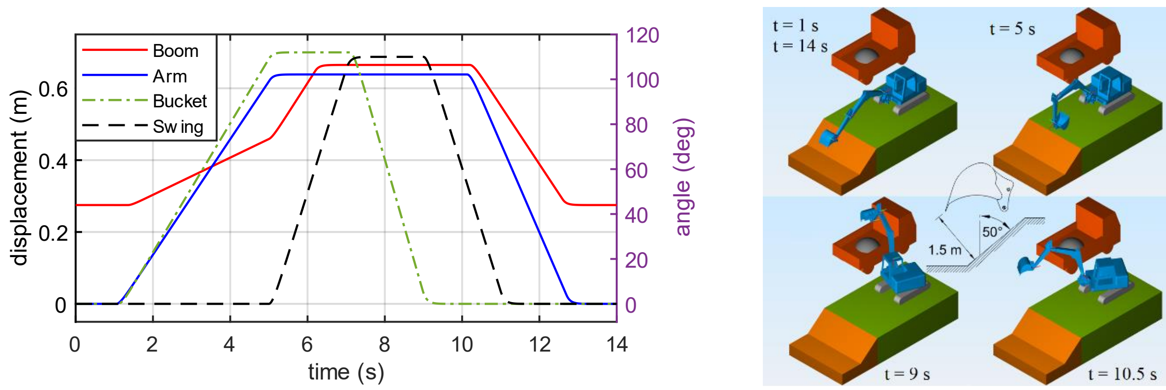

Figure 6. Actuators' position (left) and 3D rendering of the excavator during one digging cycle.

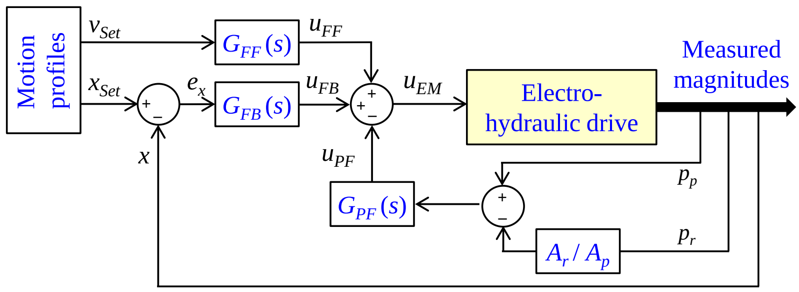

Figure 7. Motion control algorithm for the electro-hydraulic drives (boom, arm, and bucket).

without compensation [26]. Therefore, tracking of the commanded actuator position (x_{Set}) is obtained by generating a suitable speed command for the electric motor (figure 7). The final speed command (u_{EM}) is defined by combining three terms:

- The position feedback $G_{FB}(s)$ that is a PI-controller with proportional gain K_P = 14 rev/(min·mm) and integrative gain K_I = 9 rev/(min·mm·s)

$$G_{FB}(s) = \frac{u_{FB}(s)}{e_x(s)} = K_P + \frac{K_I}{s} \quad ; \tag{1}$$

- The velocity feedforward $G_{FF}(s)$ that recalls the commanded actuator velocity (v_{Set}), the pump displacement (D), and the actuator's area of the load-carrying chamber (A)

$$G_{FF}(s) = \frac{u_{FF}(s)}{v_{Set}(s)} = \frac{A}{D} \; ; \tag{2}$$

- The pressure feedback $G_{PF}(s)$ that uses both the piston-side pressure (p_p) and the rod-side one (p_r) multiplied by the area ratio of the actuator (A_r/A_p). It is a first-order high-pass filter [27] with cut-off frequency ω_{PF} = 3 rad/s and gain k_{PF} = 6.5 rev/(min·bar)

$$G_{PF}(s) = k_{PF} \cdot \frac{s}{s + \omega_{PF}} \quad . \tag{3}$$

4.2. Reference energy-efficient excavator

The JCMAS digging cycle given in figure 6 has been simulated three times in a row leading to figure 8 that focuses on the hydraulic domain of the excavator.

The position tracking - plot a, c, e, and g - results always good. For the boom actuator, that is the most challenging due to the high-inertia load, the position error - plot a - is predominately well within ±10 mm. The pressure variations in all the actuators' chambers are relatively smooth due to the addition of artificial damping - plot b, d, and f - despite the abrupt changes dictated by the working cycle (this aspect emerges from the sudden variations of the electric motors' speed in plot h).

The presence of the charge line avoids cavitation in the low-pressure side of the hydraulic cylinders; this low-pressure line, ideally at 5 bar, is necessary due to the long transmission lines connecting the hydraulic components located in excavator's upper carriage to the actuators. For these reasons, the machine behaves as expected and completes the operations successfully. Other key results that support this statement will be analyzed in the sequel.

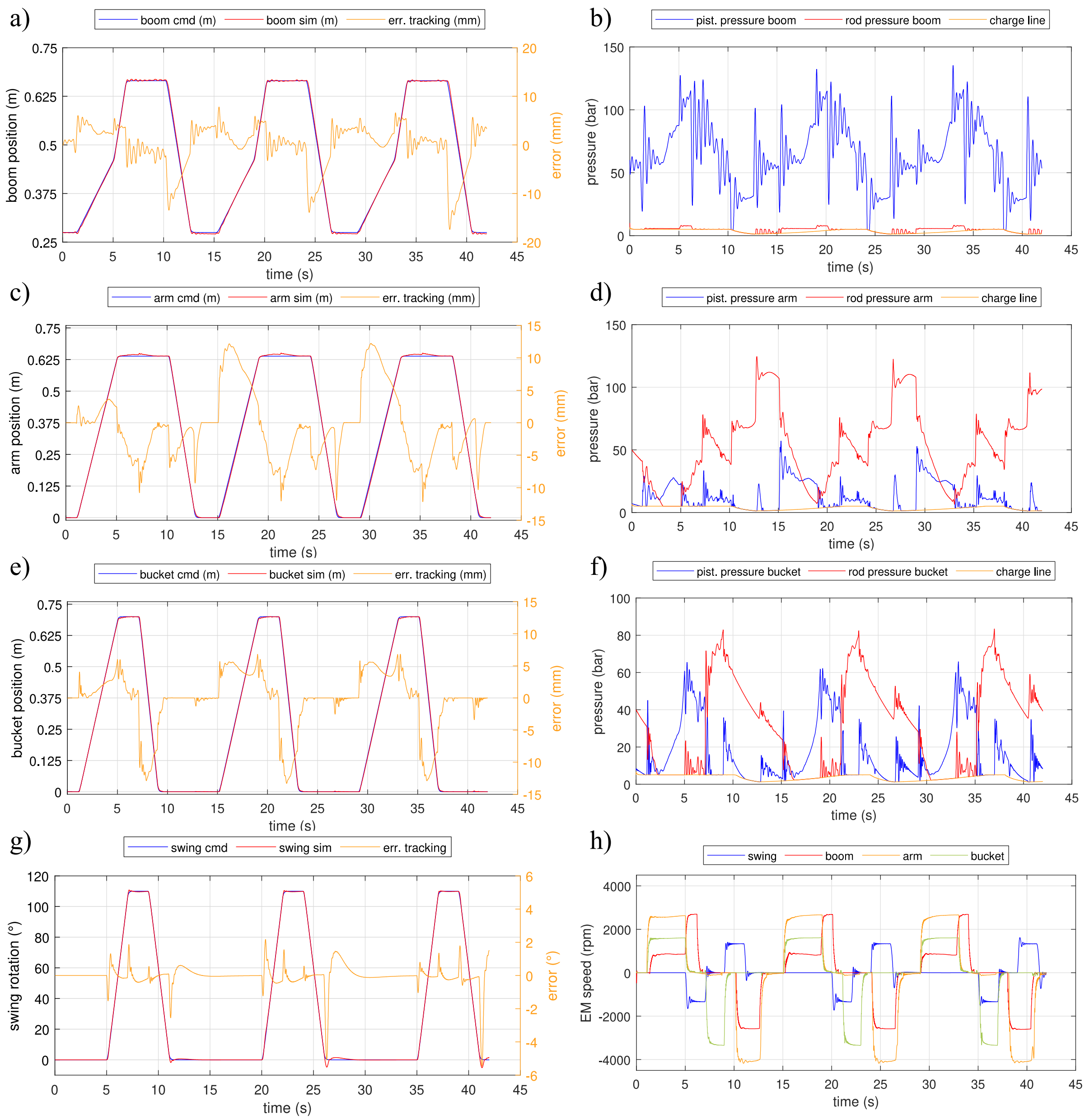

Figure 8. Simulation of the reference excavator (without the centralized power supply) while performing three sequences of the JCMAS digging cycle (cmd = input command, sim = simulated).

4.3. Modified energy-efficient excavator

Several JCMAS digging cycles have been simulated also for the excavator with the modified layout (figure 9). Both position tracking and pressure variations are unchanged with respect to the reference excavator (only the results concerning the boom are reported - plot a and b - since it has a different system architecture). The power managed by the boom drive - plot c - is now split between the hydraulic and the electric domain. This feature is possible because the centralized hydraulic supply has been added. It maintains an almost constant pressure in the common rail between 60 and 70 bar - plot d - by engaging/disengaging the centralized pump. The common rail is then connected to the inlet port of the boom's pump only when the boom actuator is extending (*i.e.*, the fractional spool position of the DCV - plot c - equals one). As a result, the boom's electric motor receives support during piston extension, that is the most demanding operation, while the potential energy is recovered only electrically during

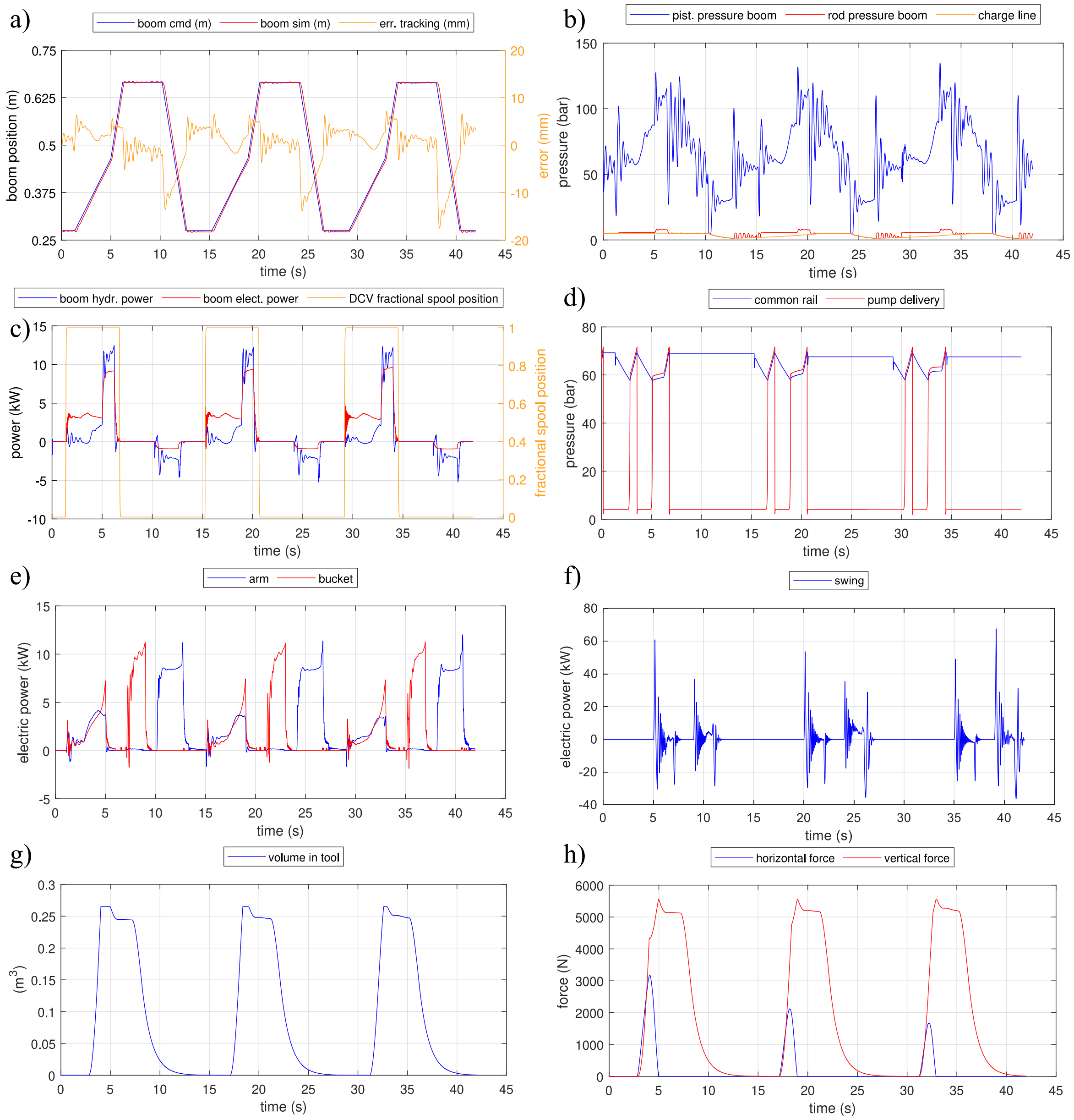

Figure 9. Simulation of the modified excavator (a centralized power supply is used) while performing three sequences of the JCMAS digging cycle.

piston retraction (the negative portions of the hydraulic power in plot c are wasted in the DCV due to flow throttling when returning fluid to the reservoir).

Other important magnitudes are portrayed in the same figure to offer an overview of the system; the amount of power - plot e and f - managed by the arm, bucket and swing actuators, the volume of material being displaced by the bucket - plot g - as well as the horizontal and vertical forces - plot h - acting on the bucket tip that are generated by the interaction with the soil (all of them are indistinguishable from the reference excavator). One can therefore conclude that also the modified machine behaves properly.

4.4. Comparing the reference versus the modified energy-efficient excavator

After confirming the functioning of both systems architectures, comparing their performance is crucial. Figure 10 shows that the position tracking of the boom actuator is identical - plot a - despite the major reduction of the power requested to the electric motor from about 21.6 kW down to 12.4 kW - plot b - that is a 43% reduction. Such an aspect indicates that a much less powerful machine can handle the operations due to the support of the centralized (hydraulic) power supply; this downswing was already performed in the current simulation since the same motor used for the arm and bucket drives was selected (*i.e.*, 21 kW of maximum installed power as opposed to 55 kW for the motor needed in the reference excavator). It is worth noticing that the total energy at the pump's shaft - plot c - that is required by the boom's drive is slightly lower for the modified architecture because the pump is now working in slightly more favorable conditions. The efficiency of the boom's drive is slightly improved since the total energy delivered to the load is indistinguishable in both layouts (not reported for brevity). The resulting change of the system behavior is also reflected on the mechanical power - plot d - requested by the centralized pump (the charge pump requires an almost constant power close to 0.3 kW). Thus, the total power delivered by the combustion engine - plot e - fluctuates a bit in the modified excavator, as well as its speed - plot f - since the engine is commanded at high-idle.

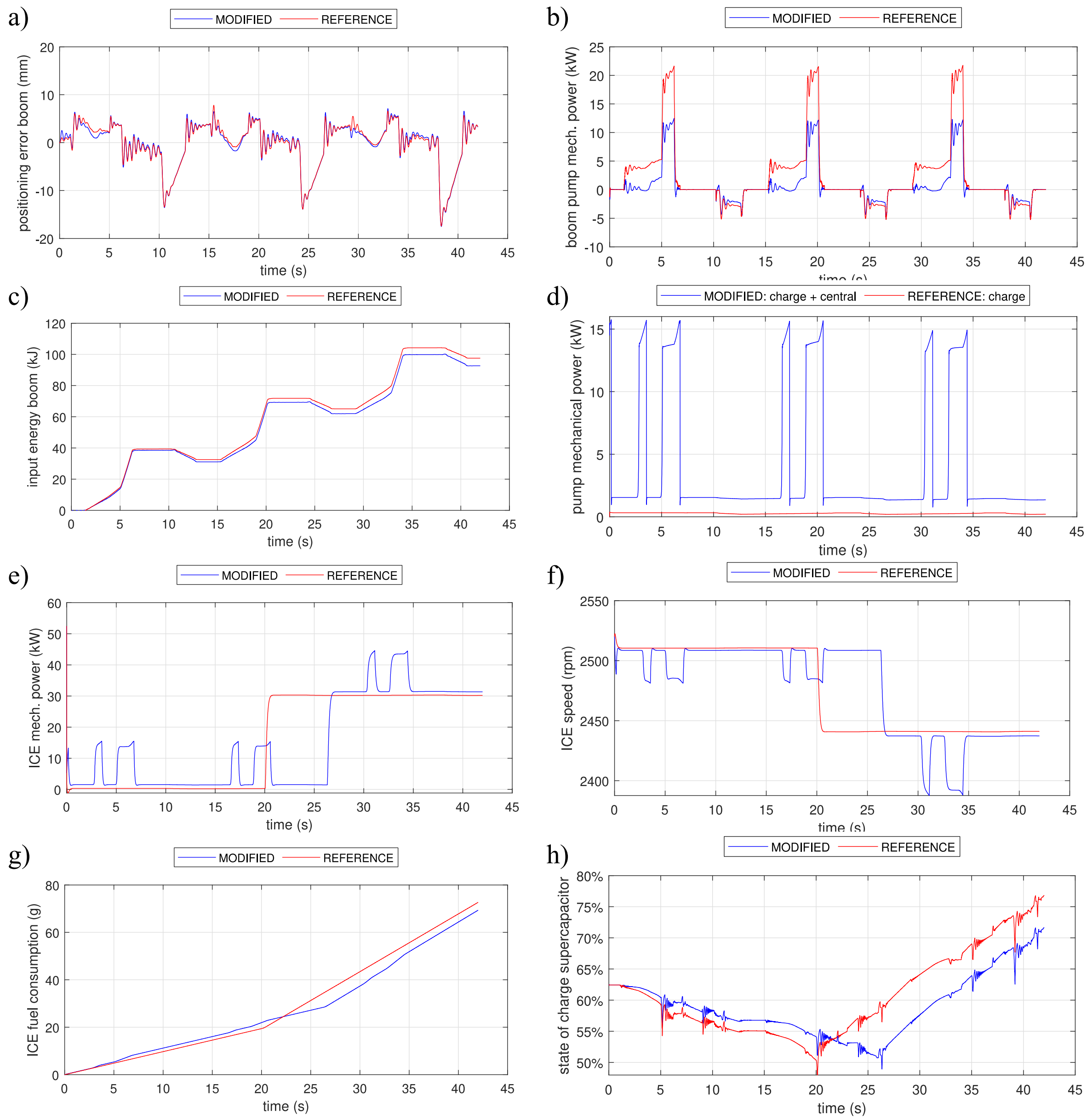

Figure 10. Comparison of the two system architectures during three JCMAS digging cycles.

The impact on the fuel consumption - plot g - is in favor of the modified excavator even if the centralized pump adds additional losses (the machine's fuel consumption is essentially the same when the electric generators are not running between 0 and 20 s). Finally, the CE drives also the generator that ensures a state-of-charge of the ultracapacitor - plot h - within 50% and 85%. The same size of the energy storage device was chosen for both arrangements; it is, therefore, clear that the modified architectures require less electric energy that is an important benefit considering that the electricity is generated onboard.

5. Conclusions

This research paper addresses the critical aspect of enhancing the machines' energy efficiency in mobile hydraulics. It presents and validates a novel layout of a hybrid, electro-hydraulic excavator that avoids flow throttling, recovers energy, and splits the power being transferred to/from the boom actuator between the hydraulic and electric domains. The latter characteristic is essential to enable the electrification of medium-to-high power applications. More specifically, this study delivers the following main results supported by simulations of a demanding JCMAS digging cycle repeated multiple times in a row:

- The reference version of the hybrid excavator is considered initially. It is based on three individual electro-hydraulic drives that are only powered by electric motors (boom, arm, bucket actuators) and a fully-electric actuator (swing). The system can track the commanded positions in closed-loop with sufficient precision and smooth behavior in terms of pressure oscillations. However, the boom actuator dictates a high-power demand with a spike up to almost 22 kW, while the arm and bucket actuators manage lower power levels always within 12 kW at the shaft of the electric motors. Such a scenario dictates an unfavorable sizing of the boom's electric motor that is largely oversized for most of the time and requires relevant generation of electricity onboard.
- The modified version of the hybrid excavator is, therefore, proposed. It resembles the reference layout with the exception of the boom actuator. The dedicated electric motor is now supported by a centralized (hydraulic) power supply driven by the combustion engine. The position tracking of the modified excavator is undistinguishable when compared to the reference machine, while the fuel consumption is slightly reduced. Most importantly, the peak power demand of the boom's electric motor is now lowered to 12.4 kW (only 57% of the reference case), since the hydraulic power supply provides the remaining power needed to complete the operations. This feature leads to a more convenient arrangement with less electric power being installed and lower generation of electricity onboard.

In conclusion, this paper shows that significant downsizing of electric machines is feasible in mobile hydraulics (*e.g.*, 61% downsizing for the boom's electric motor in a compact excavator) without affecting the machine performance and with minimum modifications to the system layout. Such a result is encouraging especially for applications with medium-to-high power levels. It favors the system's cost and compactness supporting the electrification process mentioned above that is aligned with the low-carbon economy.

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