ENERGY EFFICIENT EXCAVATOR FUNCTIONS BASED ON ELECTRO-HYDRAULIC VARIABLE-SPEED DRIVE NETWORK

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ABSTRACT

Electrification of mobile working machines is subject to increasing focus in both industry and academia. At this stage, focus has been the replacement of conventional internal combustion engines with cable or battery fed electric motors driving the main pump(s), and the replacement of rotary functions with electro-mechanical drive solutions. However, the linear functions remain controlled by hydraulic control valves resulting in substantial throttle losses, which in turn necessitates large battery sizes and/or low machine uptimes. Alternatively, the valve-controlled hydraulic cylinders may be replaced with electro-mechanical solutions in applications with limited forces, whereas heavy duty working machines such as medium/large excavators may benefit from standalone electro-hydraulic primary controlled drives, i.e., variable-speed standalone drives. The use of such solutions will substantially increase efficiency due to the absent/limited throttle control and the ability to share power through the electric supply/DC-bus. A main drawback is that each axis needs to be designed to meet both the maximum force and maximum speed, hence in the case of using single motor standalone drives, each motor and associated inverter needs to be designed to meet both the maximum force and maximum speed, potentially rendering these somewhat large. Alternatively, dual motor standalone drives can be applied, enabling power distribution via more motors. However, the use of numerous motors requires more extensive system integration and potentially large motor power installations considering industrially available non-specialized components. This paper presents a novel so-called electro-hydraulic variable-speed drive network, applied for actuation of three linear functions of an excavator implement. Cylinder chamber short-circuiting's and electro-hydraulic variable-speed units constitute a drive network allowing both electric and hydraulic power sharing. The drive network is realized with Bosch Rexroth A2 displacement units and eLION electric motors as its core components. Results demonstrate that the proposed drive network is realizable with similar energy efficiency as a standalone dual motor electro-hydraulic drive solution, but with less motor power and with fewer motors, displacement units and integration effort, rendering this a more sustainable and cost-efficient solution. Finally, it is shown that the proposed drive network is superior in terms of installed displacement, electric motor power and energy efficiency, compared to a separate metering valve drive supplied by a battery fed electro-hydraulic pump.

Keywords: Electro-hydraulic Drive Networks, Energy Efficiency, Excavator Implement Drives

1. INTRODUCTION

In the endeavours to improve the energy efficiency of hydraulic working machines, the current state of technology is the replacement of the conventional internal combustion engines with batteries, electric motors and inverter drives. However, inefficient valve controlled hydraulic systems remains the current standard resulting in low machine uptimes or large battery capacity requirements[1]. To negate these unfortunate features, obvious drive technologies to consider are electro-mechanical or electro-hydraulic drive solutions. The former has already been proposed for small and medium sized machines whereas the latter may be a feasible alternative in larger machines. Electro-hydraulic drive research and developments have historically been focused on standalone drives for actuation of (mainly) differential cylinders and, a fairly large number of such drives have been introduced in literature [1-5]. Even though many hydraulic systems contain more than a single cylinder, limited attention has so far been given to dedicated multi-axis electro-hydraulic drives. This field has, however, begun to emerge in the last few years ranging from solutions combining displacement units in variable-speed and/or displacement-controlled cylinders with valve-controlled cylinders in a mix [6,7] to more disruptive drive design approaches such as the so-called HHEA [8,9] and so-called electro-hydraulic variable-speed drive networks [10-12]. The latter approach generally contains a tremendous amount of possible drive architectures, with this number increasing exponentially with the number of cylinders (or motors) in a system. Electro-hydraulic variable-speed drive networks do basically not contain any control valves to realize the drive functionality and may be realized with short-circuited chambers and with the only active components being variable-speed displacement units in network configurations. The absence of control valves, consequently the absence of conceptual losses, renders such drive networks highly efficient, whereas the networked drive system and chamber short-circuits may allow for substantially reduced realistic power installations compared to other electro-hydraulic (and electro-mechanical) drive solutions.

This paper presents a novel electro-hydraulic variable-speed drive network (EDN) for actuation of the main linear functions of an excavator implement. The EDN is put in the context of a separate metering valve drive solution (SMV) supplied by a variable-speed pump feasible for electrified machines and a drive system based on dual displacement unit standalone electro-hydraulic drives (DEH's), also with no conceptual losses. Based on measured digging cycles from a representative 17-19t wheeled excavator, the basic drive requirements are established, and the main components subsequently determined with a focus on Bosch Rexroth A2 displacement units and Bosch Rexroth eLION motors. Hereafter, component loss models based on experimental measurements are included and the resulting power consumption and loss distribution is elaborated. Results imply that the EDN energy efficiency is superior to the SMV and comparable to the DEH, but with lower installed displacement and electric motor power in comparison.

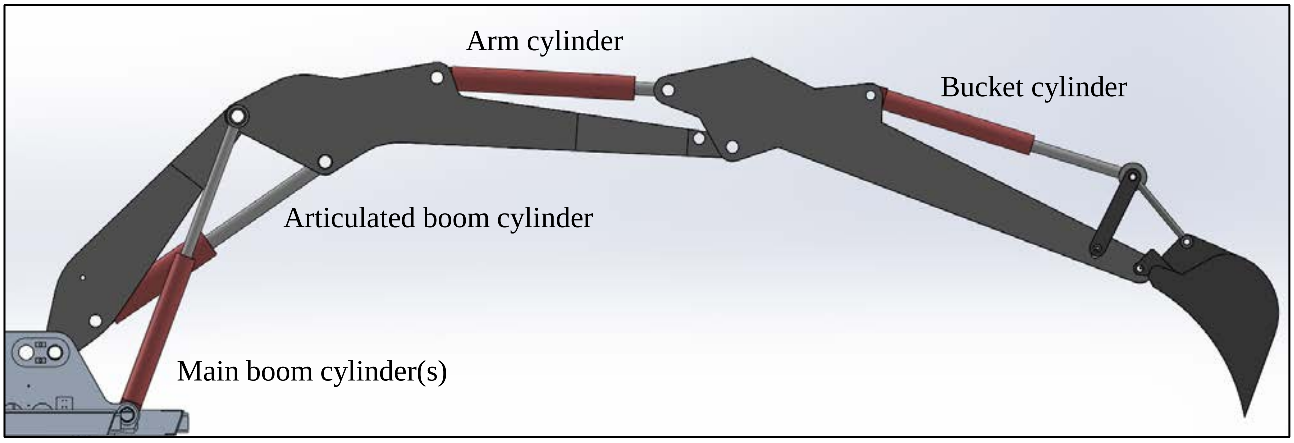

Figure 1: Illustration of excavator implement, emphasizing the cylinders in consideration.

[1] As experienced by the authors at the fair BAUMA in Munich, Germany in October 2022.

The study presented in the following is restricted to the main cylinders of the implement of a 17-19t wheeled excavator, with these being the two main boom cylinders (excluding the articulated boom cylinder), the arm cylinder and the bucket cylinder as illustrated in Figure 1.

An excavator, especially a wheeled excavator, is used for a variety of tasks which, for an industrially applicable design, should all be considered. In some of these tasks, like levelling, the controllability is the main focus and in other the energy efficiency is most important. The latter is true for digging, a predominant excavator task. For the sake of simplification only eight measured digging cycles are considered here, containing piston velocities and forces as depicted on Figure 2.

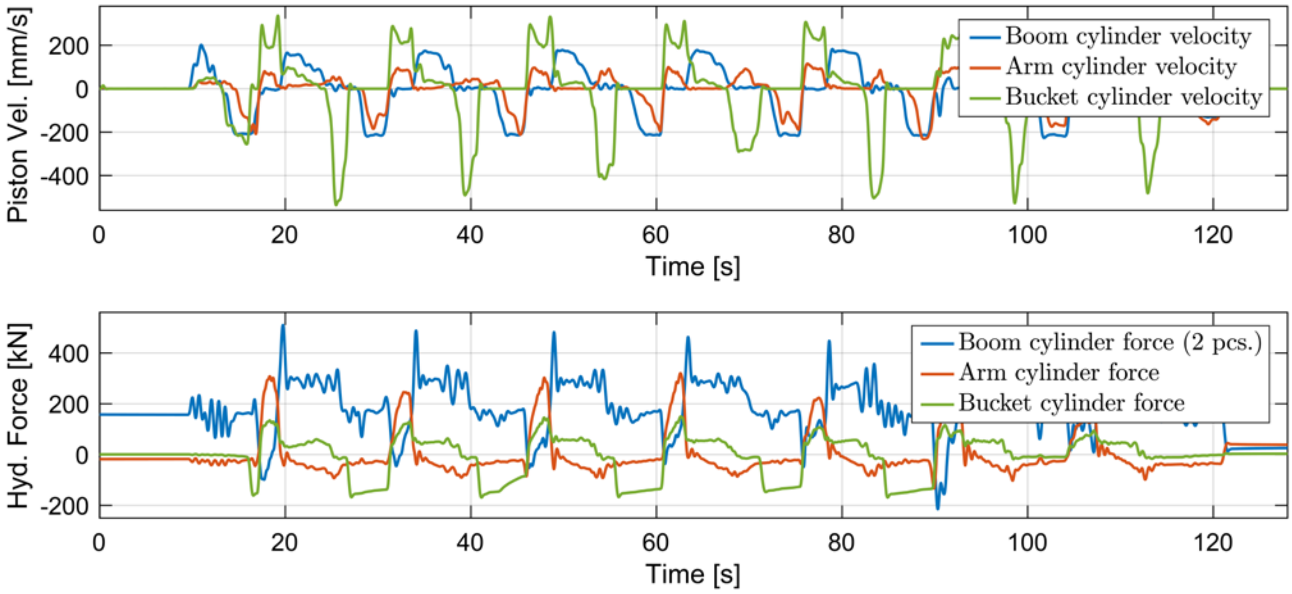

Figure 2: Load cycles for implement cylinder piston velocities and hydraulic pressure forces.

2. ELECTRO-HYDRAULIC DRIVE NETWORK FOR EXCAVATOR IMPLEMENT

Based on the chosen measurement data (Figure 2) the proposed electro-hydraulic drive network (EDN) considered for actuation of the excavator implement cylinders is depicted in Figure 3. This drive network is a so-called minimal realizable EDN, meaning that it is realized with the minimal number of displacement units allowing the control of cylinder motions and the system pressure level. Hence, with three linear functions (noting that there are two parallel boom cylinders), there are four motor inputs and four outputs to control.

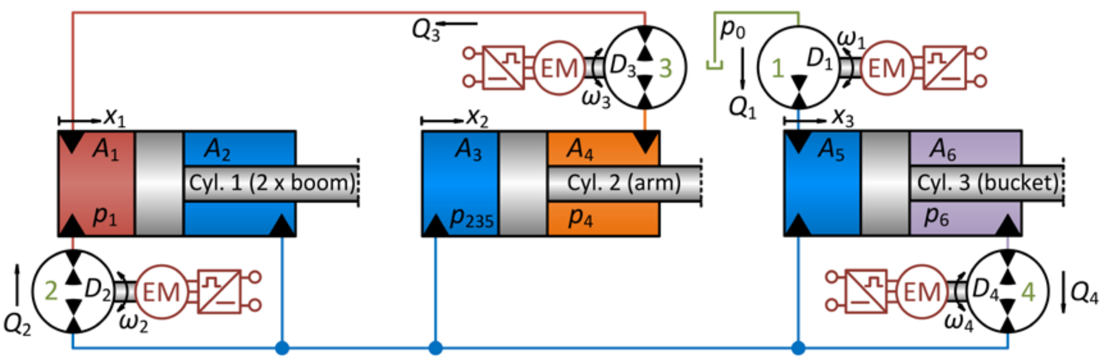

Figure 3: Proposed electro-hydraulic drive network (EDN) for excavator implements.

The proposed EDN encompasses two chamber short-circuiting's being the short-circuit of the boom rod side with arm as well as the bucket piston side chambers. The consequences of these short-circuiting's are identical pressures in the connected chambers but also that hydraulic fluid can flow unrestricted between the connected chambers. Hence, during simultaneous cylinder motions, power in terms of flow under pressure can be transmitted nearly loss free between the cylinders. The network of variable-speed displacement units (VsD's) allows to control the fluid exchange between the control volumes of the drive system. However, due to the way these are interconnected, the ability to control the drive system relies on a combined effort of all VsD's.

2.1. Component Requirements

Considering the EDN schematic of Figure 3, the flow continuity equations expressed as the pressure dynamics are given by Eqs. (1), (2), assuming identical fluid bulk modulii β.

$$\dot{p}_1 = \frac{\beta}{V_1}(Q_1 + Q_3 - A_1\dot{x}_1) \quad , \quad \dot{p}_4 = \frac{\beta}{V_4}(A_4\dot{x}_2 - Q_3) \quad , \quad \dot{p}_6 = \frac{\beta}{V_6}(A_6\dot{x}_3 - Q_4) \tag{1}$$

$$\dot{p}_{235} = \frac{\beta}{V_{235}}(Q_1 - Q_2 + Q_4 + A_2\dot{x}_1 - A_3\dot{x}_2 - A_5\dot{x}_3) \tag{2}$$

Solving Eqs. (1), (2) for Q_1, Q_2, Q_3, Q_4 under steady state conditions, Eqs. (3), (4) are obtained.

$$Q_1 = (A_1 - A_2)\dot{x}_1 + (A_3 - A_4)\dot{x}_2 + (A_5 - A_6)\dot{x}_3 \tag{3}$$

$$Q_2 = A_1\dot{x}_1 - A_4\dot{x}_2 \quad , \quad Q_3 = A_4\dot{x}_2 \quad , \quad Q_4 = A_6\dot{x}_3 \tag{4}$$

The displacement unit pressure differences for the four units are given by Eqs. (5), (6).

$$\Delta p_1 = p_{235} - p_0 \qquad , \qquad \Delta p_2 = p_1 - p_{235} \tag{5}$$

$$\Delta p_3 = p_1 - p_4 \qquad , \qquad \Delta p_4 = p_{235} - p_6 \tag{6}$$

Combining Eqs. (3)-(6) with the load cycles of Figure 2, the maximum and minimum required displacement unit flows and pressure differences appear as tabularized in Table 1.

Table 1: Max. and min. requirements for VsD flows and pressure differences of the EDN.

	Unit 1	Unit 2	Unit 3	Unit 4
Max. flow:	111 [l/min]	198 [l/min]	50 [l/min]	98 [l/min]
Min. flow:	-196 [l/min]	-227 [l/min]	-101 [l/min]	-155 [l/min]
Max. Δp:	273 [bar]	266 [bar]	217 [bar]	107 [bar]
Min. Δp:	20 [bar]	-240 [bar]	-346 [bar]	-367 [bar]

The requirements of Table 1 will be used for component sizing in Section 4.

3. BENCHMARK DRIVE SYSTEMS FOR EXCAVATOR IMPLEMENT

The two benchmark drive systems are chosen from their suitability in electrified machinery applications, their energy efficiency perspectives and ability to control piston motions as well as the lower system pressure. These benchmark drive systems are described in the following.

3.1. Benchmark Drive System # 1 & Main Component Requirements

The first benchmark drive system relies on separate metering control as depicted in Figure 4 and is here denoted SMV. Besides being a separate metering valve control drive system, this also includes flow regenerative valves on the boom and arm functions, and it is supplied by an electro-hydraulic variable-speed pump unit realizable with electric load-sensing function.

The SMV valve flows are given by Eqs. (9)-(11) where p_{lim} = 1 [bar] and $\dot{x}_{lim}$ = 10 [mm/s], with the pump flow given by Eq. (12).

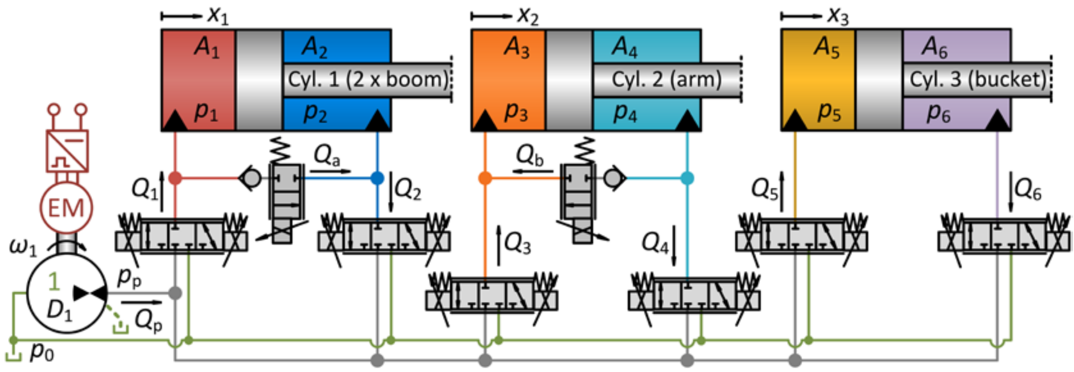

Figure 4: Separate metering valve drive system with electro-hydraulic supply (SMV).

$$Q_1 = A_1\dot{x}_1 + Q_a, Q_2 = A_2\dot{x}_1 + Q_a, Q_3 = A_3\dot{x}_2 - Q_b, Q_4 = A_4\dot{x}_2 - Q_b, Q_5 = A_5\dot{x}_3, Q_6 = A_6\dot{x}_3 \quad (9)$$

$$Q_a = \begin{cases} -A_2(\dot{x}_1 + \dot{x}_{lim}) & \text{for} \quad \dot{x}_1 < -\dot{x}_{lim},\ p_1 - p_2 > p_{lim} \\ 0 & \text{else} \end{cases} \quad (10)$$

$$Q_b = \begin{cases} A_4(\dot{x}_2 - \dot{x}_{lim}) & \text{for} \quad \dot{x}_2 > \dot{x}_{lim},\ p_4 - p_3 > p_{lim} \\ 0 & \text{else} \end{cases} \quad (11)$$

$$Q_p = \bar{Q}_1 + \bar{Q}_2 + \bar{Q}_3 + \bar{Q}_4 + \bar{Q}_5 + \bar{Q}_6 \quad (12)$$

$$\bar{Q}_1 = \begin{cases} Q_1 & \text{for} \quad Q_1 > 0 \\ 0 & \text{for} \quad Q_1 < 0 \end{cases}, \quad \bar{Q}_5 = \begin{cases} Q_5 & \text{for} \quad Q_5 > 0 \\ 0 & \text{for} \quad Q_5 < 0 \end{cases}, \quad \bar{Q}_3 = \begin{cases} Q_3 & \text{for} \quad Q_3 > 0 \\ 0 & \text{for,} \quad Q_3 < 0 \end{cases} \quad (13)$$

$$\bar{Q}_2 = \begin{cases} 0 & \text{for} \quad Q_2 > 0 \\ -Q_2 & \text{for} \quad Q_2 < 0 \end{cases}, \quad \bar{Q}_4 = \begin{cases} 0 & \text{for} \quad Q_4 > 0 \\ -Q_4 & \text{for} \quad Q_4 < 0 \end{cases}, \quad \bar{Q}_6 = \begin{cases} 0 & \text{for} \quad Q_6 > 0 \\ -Q_6 & \text{for} \quad Q_6 < 0 \end{cases} \quad (14)$$

The pump outlet pressure is controlled via electric load sensing, adjusted according to Eq. (15), where p_{po} = 7 [bar] is the pressure overhead allowing for valve control under relevant loads.

$$p_p = \max(p_1, p_2, p_3, p_4, p_5, p_6) + p_{po} \quad (15)$$

From Eqs. (9)-(15) and the load cycles in Figure 2, the ideal maximum and minimum pump flow and pump pressure difference appear as outlined in Table 2.

Table 2: Max. and min. requirements for the VsD flow and pressure difference of the SMV.

	Unit 1
Max. flow:	250 [l/min]
Min. flow:	0 [l/min]
Max. Δp:	394 [bar]
Min. Δp:	46 [bar]

3.2. Benchmark Drive System # 2 & Main Component Requirements

The second benchmark drive system is based on dual pump electro-hydraulic standalone drives (DEH's) and is depicted in Figure 5. The DEH do not contain any conceptual losses, rendering this one of the most efficient variable-speed electro-hydraulic drives introduced in literature. This drive system can share the electric supply across all VsD's and has the ability to control the lower chamber pressure level on all cylinders.

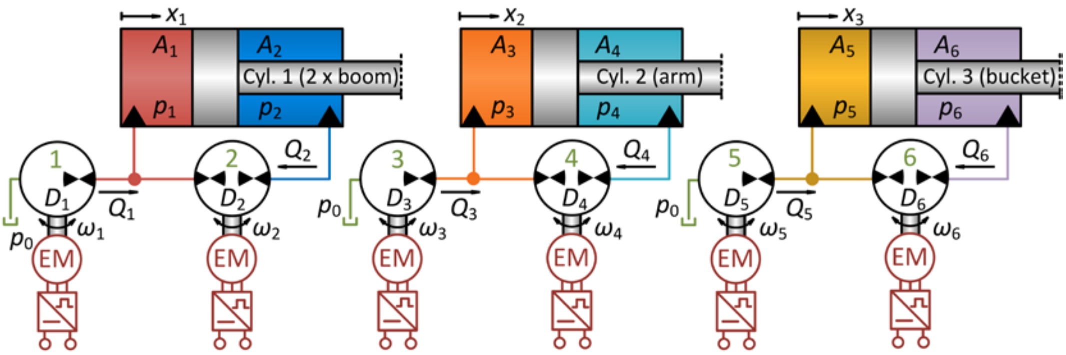

Figure 5: Drive system based on dual pump electro-hydraulic standalone drives (DEH's).

The displacement unit flows and pressure differences for the DEH's are given by Eqs. (16)-(19).

$$Q_1 = (A_1 - A_2)\dot{x}_1\,,\;\; Q_2 = A_2\dot{x}_1\,,\;\; Q_3 = (A_3 - A_4)\dot{x}_2 \tag{16}$$

$$Q_4 = A_4\dot{x}_2\,,\;\; Q_5 = (A_5 - A_6)\dot{x}_3\,,\;\; Q_6 = A_6\dot{x}_3 \tag{17}$$

$$\Delta p_1 = p_1 - p_0\,,\;\; \Delta p_2 = p_1 - p_2\,,\;\; \Delta p_3 = p_3 - p_0 \tag{18}$$

$$\Delta p_4 = p_3 - p_4\,,\;\; \Delta p_5 = p_5 - p_0\,,\;\; \Delta p_6 = p_5 - p_6 \tag{19}$$

Combining Eqs. (16)-(19) with the load cycles of Figure 2, the ideal required maximum and minimum displacement unit flows and pressure differences appear as shown in Table 3.

Table 3: Max. and min. requirements for the VsD flows and pressure differences of the DEH.

	Q1	Q2	Q3	Q4	Q5	Q6
Max. flow:	94 [l/min]	117 [l/min]	35 [l/min]	50 [l/min]	78 [l/min]	98 [l/min]
Min. flow:	-105 [l/min]	-131 [l/min]	-70 [l/min]	-101 [l/min]	-124 [l/min]	-155 [l/min]
Max. Δp:	306 [bar]	286 [bar]	273 [bar]	253 [bar]	184 [bar]	164 [bar]
Min. Δp:	20 [bar]	-240 [bar]	20 [bar]	-198 [bar]	20 [bar]	-367 [bar]

4. IMPLEMENT DRIVE COMPONENT SIZINGS

The sizing of components is conducted according to the maximum and minimum displacement unit flows and pressure differences obtained for the EDN, SMV and DEV.

4.1. Sizing of Displacement Units

In all cases, the displacement units considered are Bosch Rexroth A2 bent axis pumps or motors[2], due to their proven application history. It is assumed that the lower pressure level of any control volume is controlled to 20 [bar]. For any displacement unit not connected to a vented fluid reservoir, the suction pressure conditions are not violated for any of the considered units and may therefore operate in all four quadrants. Hence, displacement units not connected to fluid reservoirs are chosen as A2FM hydraulic motors. Displacement units that are connected to a fluid reservoir are subject to suction restrictions and may be operated in two quadrants. Hence, A2FO hydraulic pumps are more feasible than A2FM's. Also, cavitation may occur when fluid is pumped from a vented reservoir to a pressurized control volume. Consequently, A2FO sizing's are based on the maximum positive flow requirements. Furthermore, flow losses are not included in the sizing, and hence the sizing is based on nominal shaft speeds but with an upper limit of 6000 [rpm].

The corresponding choices of displacement units chosen for the three drive systems are tabularized in Table 4 along with actual geometric displacements to be installed. Here, the EDN and DEH are subject to the lowest installed displacement, in both cases are more than 9 [%] lower than the SMV.

Table 4: Summary of chosen displacement units and total displacements for the three drive systems.

Drive	Unit 1	Unit 2	Unit 3	Unit 4	Unit 5	Unit 6	Total Displ.
SMV	A2FO 180	-	-	-	-	-	180,0 [ccm]
DEH	A2FO 45	A2FM 23	A2FO 12	A2FM 23	A2FO 32	A2FM 28	163,5 [ccm]
EDN	A2FO 56	A2FM 56	A2FM 23	A2FM 28	-	-	163,7 [ccm]

4.2. Sizing of Electric Motors

Based on the displacement unit sizes of Table 4, the ideal shaft speed and torque requirements for the electric motors can be established for the three drives as specified in Table 5.

Table 5: Required motor speeds and torques for the three drive systems.

SMV	Unit 1					
$\max(\lvert n \rvert)$:	1384 [rpm]	-	-	-	-	-
$\max(\lvert \tau \rvert)$:	1129 [Nm]	-	-	-	-	-
DEH	Unit 1	Unit 2	Unit 3	Unit 4	Unit 5	Unit 6
$\max(\lvert n \rvert)$:	2294 [rpm]	5709 [rpm]	5822 [rpm]	4398 [rpm]	3867 [rpm]	5505 [rpm]
$\max(\lvert \tau \rvert)$:	222 [Nm]	104 [Nm]	53 [Nm]	93 [Nm]	94 [Nm]	165 [Nm]
EDN	Unit 1	Unit 2	Unit 3	Unit 4		
$\max(\lvert n \rvert)$:	3486 [rpm]	4010 [rpm]	4398 [rpm]	5505 [rpm]	-	-
$\max(\lvert \tau \rvert)$:	244 [Nm]	240 [Nm]	127 [Nm]	165 [Nm]	-	-

The flow and torque losses are not included in the following electric motor sizing examples and hence these are sized considering the motor S2 torques as the maximum design torques and the shaft speeds well below the maximum motor speeds. Considering the eLION EMS1 motor portfolio[3] and the specifications in Table 5, the motor choices are presented in Table 6 along with the resulting total rated motor power.

[2] Based on data sheets "RE 91071/2021-05-17, Bosch Rexroth AG" and "RE 91401/06.2012, Bosch Rexroth AG".
[3] RE98771/2022 04 26, Bosch Rexroth AG.

The deviations in the total rated (motor) power results especially from either relatively high required torques or the use of a relatively large number of VsD's. In case of a high maximum motor torque, the associated rated power tends to be correspondingly large as well, attributed the associated speed range. Similarly, the use of a relatively large number of VsD's tends to result in a relatively large total installed torque overhead, with this increasing with number of units used. A reasonable level of installed torque and power is therefore achieved with few units and with a reasonable ratio between required speed and torque for each unit.

Table 6: Choice of eLION motors EMS1 along with the resulting total rated (motor) power to be installed.

Drive	Unit 1	Unit 2	Unit 3	Unit 4	Unit 5	Unit 6	Total rated power
SMV	EMS1-16	-	-	-	-	-	74 [kW]
DEH	EMS1-10	EMS1-10	EMS1-10	EMS1-10	EMS1-10	EMS1-10	54 [kW]
EDN	EMS1-10	EMS1-10	EMS1-10	EMS1-10	-	-	44 [kW]

In summary, the total rated motor power for the EDN is 18,5 [%] lower than the DEH and 40,5 [%] lower than the SMV, owed to the fewer units applied compared to the DEH and, indirectly, to the substantially lower torque requirement compared to the SMV.

4.3. Tank Sizing Considerations

Besides the component sizes, hence the drive sustainability in terms of material usage, another important aspect is the sizing of the fluid reservoir/tank. The EDN, as well as the DEH, do not rely of throttle control, and hence the fluid degasification and fluid cooling requirements are substantially reduced, as the "throttling" taking place in these drive systems are associated only with cross-port leakage and drain flows, whereas it for the SMV is the full pump flow that is throttled. A rule of thumb suggests that the tank size should be chosen as three times the average (throttled) flow to allow for heat dissipation and degasification[4]. Using this rule, the theoretical SMV tank size may be obtained as Eq. (20). Applying the same rule for the EDN (and the DEH), and accounting for the cylinder piston volumes, the theoretical tank size is obtained as Eq. (21).

$$V_{tank,SMV} = 3\ \mathrm{mean}(Q_p) \tag{20}$$

$$V_{tank,EDN} = 3\ \mathrm{mean}(\Sigma Q_D + \Sigma|Q_L|) + (A_1 - A_2)x_{1\max} + (A_3 - A_4)x_{2\max} + (A_5 - A_6)x_{3\max} \tag{21}$$

Using the loss models described in Section 5, the resulting theoretical tank volumes are outlined in Table 7, suggesting significant reductions in the EDN and DEH tank volumes compared to the SMV, hence also in the required amount of fluid in the system. Conventional tank designs used in mobile machinery are often optimized in various ways, and hence the tank volume could be as low as half of the volume proposed for the SMV in Table 7. However, even in such a case, the proposed theoretical volumes for the EDN and DEH are still reduced by more than 80 [%].

Table 7: Ideal fluid reservoir/tank sizes and relative reduction compared to SMV.

	EDN	DEH	SMV
Ideal fluid reservoir/tank size	30,7 [l]	29,4 [l]	330,4 [l]
Reduction relative to SMV	↓ 90,7 [%]	↓ 91,1 [%]	-

[4] https://www.powermotiontech.com/hydraulics/reservoirs-accessories/article/21882642/fundamentals-of-hydraulic-reservoirs (assessed 26. October 2023).

5. ENERGY EFFICIENCIES & LOSS DISTRIBUTION

Having decided on the main component sizes of the proposed EDN and the benchmark drive systems SMV and DEH, their energy efficiency and loss distributions are considered in the following as well as the loss models applied for the analyses.

5.1. Loss Models Used in Case Studies

The loss model used as reference for the displacement units is based on measured loss maps for the A2FMM 32 [ccm/rev], assuming this also representative for the A2FO units. The loss map used as reference for the eLION EMS1's is based on measured losses of an EMS1-20 type motor. In all cases, the measured data has been smoothened to mitigate the impact of outlier measurement points, and the hydraulic losses extrapolated for pressure differences above 350 [bar], hence representing approximate loss measures. The approximate loss diagrams are shown in Figure 6.

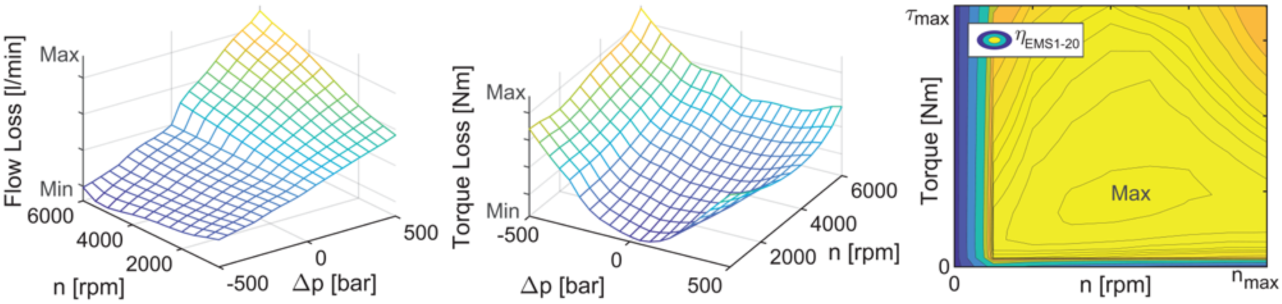

Figure 6: Approximate loss diagrams. **Left plot**: 2Q flow loss map for A2FMM 32. **Center plot**: 2Q torque loss map for A2FMM 32. **Right plot**: Power loss map for eLION EMS1-20 component series motor.

Regarding the A2FM, the measured flow losses account for both cross-port leakage flow and drain flows. It is assumed that the flow losses are evenly distributed in the reference measurements and that these can be scaled to relevant displacement unit sizes using scaling laws [2,10]. The scaling of the EMS1 assumes that the efficiency map is invariant with respect to the motor size i.e., that any motor has the same efficiency map as the reference EMS1-20, with the axes scaled to the max. torque and speed of the motor in consideration. Furthermore, for the EMS1 motors it is assumed that the efficiency map is valid for all four quadrants. The total inverter losses $P_{inv,loss}$ and DC-bus losses $P_{dc,loss}$ are estimated according to Eqs. (22), where R_i is the electrical resistance of the i^{th} inverter, R_{dc} is the electrical DC-bus resistance, $U_{dc} = 700$ [Vdc] the DC-bus voltage, $I_{inv,nom,i}$ the nominal current of the i^{th} inverter, and $I_{dc,nom} = \sum_{i=1}^{n} I_{inv,nom,i}$ the nominal DC-bus current. Finally, the inverter and DC-bus efficiencies are assumed to be $\eta_{inv} = 0{,}98$ [-], $\eta_{dc} = 0{,}98$ [-] and the battery efficiency assumed to be $\eta_{bat} = 0{,}90$ [-].

$$P_{inv,loss} = \sum_{i=1}^{n} R_i \frac{P_{m,i}^2}{U_{dc}^2}, P_{dc,loss} = R_{dc} \frac{P_{inv,loss}^2}{U_{dc}^2}, R_i = \left(\frac{1}{\eta_{inv}} - 1\right) \frac{U_{dc}}{I_{inv,nom}}, R_{dc} = \left(\frac{1}{\eta_{dc}} - 1\right) \frac{U_{dc}}{I_{dc,nom}} \quad (22)$$

5.2. Loss Distribution & Power Consumption

The overall average loss distribution of the three drive systems, using the load cycles of Figure 2, are tabularized in Table 8 along with the average and peak power consumption. The deviation in losses and power consumption of the EDN and DEH relative to the SMV drive are shown in Table 9. From this it is evident that significant reductions in all losses are achieved with DEH and EDN drives compared to the SMV drive, except for the displacement unit friction measures which are higher, attributed the use of more displacement units and the associated specific loads.

Table 8: Average losses and average and peak power consumption.

Drive Type	Avg. Hydraulic Losses	Avg. Friction Losses	Avg. Electric Losses	Avg. Battery Losses	Peak Power Consumption	Avg. Power Consumption
SMV	15,6 [kW]	1,3 [kW]	2,8 [kW]	4,0 [kW]	116,7 [kW]	39,8 [kW]
DEH	1,2 [kW]	2,8 [kW]	2,1 [kW]	2,4 [kW]	83,7 [kW]	18,3 [kW]
EDN	1,2 [kW]	2,5 [kW]	2,0 [kW]	2,5 [kW]	84,6 [kW]	18,1 [kW]

Furthermore, it is notable that the average energy consumption by the DEH and EDN drives are reduced by more than 54 [%] compared to the SMV. If the load cycles considered here are assumed generally representative for the implement function, then for an 8 hour shift the SMV drive would require a 318 [kWh] battery, whereas the DEH and EDN drives would only require 146 [kWh] and 145 [kWh] batteries, respectively.

Table 9: Relative differences in losses and power consumption of the DEH and EDN compared to SMV.

Drive Type	Avg. Hydraulic Losses	Avg. Friction Losses	Avg. Electric Losses	Avg. Battery Losses	Peak Power Consumption	Avg. Power Consumption
DEH	↓ 92,3 %	↑ 115,4 %	↓ 25,0 %	↓ 40,0 %	↓ 28,3 %	↓ 54,0 %
EDN	↓ 92,3 %	↑ 92,3 %	↓ 28,6 %	↓ 37,5 %	↓ 27,5 %	↓ 54,5 %

Finally, it is notable that the drive energy efficiencies, in terms of the ratio between the average battery power and the average hydraulic piston power (≈ 9,9 [kW]), are 24,8 [%], 54,0 [%] and 54,5 [%], for the SMV, DEH and EDN drive systems, respectively. Hence, the DEH and EDN drive systems are subject to energy efficiency increases of ≈118 [%] and ≈120 [%], respectively.

6. CONCLUSION

A novel electro-hydraulic variable-speed drive network is proposed, specifically intended for actuation of the cylinder functions in excavator implements, namely the main boom cylinders, the arm and bucket cylinders. The proposed drive network encompasses the short circuiting of the boom cylinder rod sides and the arm and bucket cylinder piston sides. Consequently, the three-cylinder system contains four effective control volumes, which are actuated by four variable-speed displacement units in a network configuration. The proposed drive network is placed in context of a separate metering drive system supplied by a variable-speed fixed displacement pump as well as a dual pump electro-hydraulic standalone drive system. Main component requirements are established, and components subsequently chosen from the Bosch Rexroth A2 hydraulic pump and motor series, and the eLION electric motor series. Steady state models relying on measured component losses are established, and the power consumption and loss distribution evaluated. The resulting key measures in terms of total energy efficiency, power consumption, total installed displacement and electric motor power are evaluated, and the relative differences for the proposed drive network compared to the benchmark drive systems are outlined in Table 10.

Table 10: Relative differences in key figures of EDN compared to the DEH and SMV drive systems.

Drive Type	Avg. Total Energy Efficiency	Avg. Power Consumption	Total Installed Displacement	Total Installed Motor Power
EDN vs. DEH	↑ 0,9 %	↓ 1,1 %	↑ 0,1 %	↓ 9,1 %
EDN vs. SMV	↑ 119,2 %	↓ 54,5 %	↓ 11,6 %	↓ 40,5 %

It is found that the proposed drive network may be realized with substantially less installed electric motor power and similar energy efficiency compared to the dual pump electro-hydraulic standalone benchmark drive system. Furthermore, compared to the separate metering valve drive system the proposed drive network may be realized with less installed displacement, and significantly less installed electric motor power, and with more than twice the energy efficiency. Also, the proposed drive network may be realized with an ideal tank volume reduction of more than 80 [%] compared to the separate metering valve drive system due to the substantially reduced fluid degasification requirements. The results emphasize the potential significance of electro-hydraulic drive networks in the ongoing electrification trends and efficiency improvements of hydraulic working machines.

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