Guidelines to Select Between Self-Contained Electro-Hydraulic and Electro-Mechanical Cylinders

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***Abstract*—This research paper presents a guideline on how to select between self-contained electro-hydraulic and electro-mechanical cylinders. An example based on the motion control of a single-boom crane is studied. The sizing process of the different off-the-shelf components is analyzed in terms of design impact when replacing a traditional valve-controlled hydraulic cylinder. The self-contained electro-hydraulic solution is the best choice when there exists a risk for high impact forces, when the required output power is continuously above 2 kW, or when installation space, weight, and cost are critical design objectives. However, the electro-mechanical solution is expected to show better controllability due to the higher drive stiffness, requires less control effort, has higher energy efficiency, and lower system complexity resulting in a more straightforward design approach.**

***Index Terms*—hydraulic systems, electric drives, self-contained acutators, linear actuators, actuation system design, component selection, electro-mechanical cylinders, electro-hydraulic cylinders, valve-controlled cylinders, load-carrying applications**

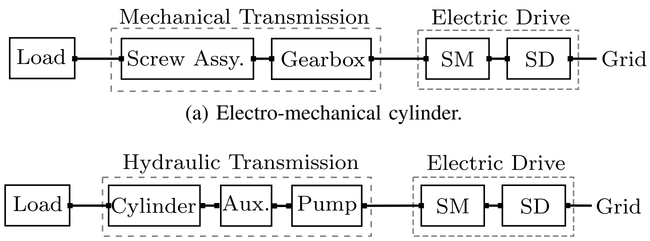
I. Introduction

Traditional valve-controlled linear hydraulic actuators can be implemented in many architectures [1]. They are characterized by reliability, high force and power capability, and excellent overload protection (e.g., shock absorption) but also by relevant power losses due to flow throttling in the valves, demanding efforts on installation and maintenance, as well as costly piping due to the centralized hydraulic power supply. Despite their disadvantages, they are still commonplace in industry.

An effort to replace these conventional hydraulic systems with electro-mechanical cylinders (EMCs) has been a popular research topic and engineering task for many years, especially in aerospace systems, to reduce the system weight, installation time, and maintenance effort [2]–[6]. However, predicting the wear life is challenging and the actuator cannot absorb shocks in standby mode, so there is a high risk of failure (e.g., jamming that would result in dangerous consequences). Therefore, these actuators are still not accepted in commercial airplanes as primary actuators for flight control. As a consequence, electro-hydraulic cylinders (EHCs) that are self-sufficient and completely sealed have been introduced. They typically contain a dedicated variable-speed pump that controls the motion of the cylinder. This technology, as explained by [7], was first introduced in aerospace systems, demonstrating that the disadvantages of traditional hydraulics can be reduced significantly. Indeed, EHC are now used for flaps control in commercial airplanes [8].

In other fields of industry, especially in low power applications (below 5 kW) where disadvantages such as increased wear, difficult overload protection, and lower load forces are accepted [9], hydraulic systems have been replaced by EMCs to increase the energy efficiency and to eliminate oil spills. Consequently, the idea of combining the advantages of EMC and hydraulics, has been further developed in the recent years to compete against EMCs [9]–[15]. For instance, Michel and Weber created a prototype and compared it against a commercial EMC with a ball screw drive [9]. The overall efficiency of the EMC was 74.6%, including 13% losses in the electric drive and 12.4% losses in the screw transmission. Comparatively, the EHC had about 25% losses in the hydraulic transmission, resulting as a valid alternative. A peculiar aspect of the EHCs, namely the thermal behavior, is still an ongoing research topic [16]–[18], as well as the durability [19]. An EHC concept comprising passive load-holding devices, sealed tank, capable of recovering energy, four quadrants operations, and suitable for power levels above 5 kW was proposed in [14], and implemented on a load carrying application in [15]. Hagen et al. present further investigations on this concept with a focus on motion performance [20] and energy efficiency [21] showing that EHCs significantly improves conventional hydraulics.

The two actuator technologies mentioned above can be used in several applications as an alternative to conventional valve-controlled actuators. However, more specific comparative analyses regarding their design impact in terms of installation space, design complexity, speed and force (power) capability, reliability, service life (durability), and cost when implemented on load-carrying applications, are still missing in the technical literature. Hence, this paper aims to present general guidelines on how to size and select the best self-contained linear actuation system for a given working cycle. The two considered actuators (Fig. 1) are the off-the-shelf heavy-duty electro-mechanical cylinder from Rexroth [22] and the self-contained electro-hydraulic concept presented in [15] using off-the-shelf components. An example based on the motion control of a single-boom crane is studied, and the sizing process of the



(a) Electro-mechanical cylinder.

(b) Electro-hydraulic cylinder.

Fig. 1: The two considered self-contained cylinders.

different off-the-shelf components is also considered.

II. Sizing Principles

Both actuation systems consist of an electric drive and a transmission system that converts rotary motion to linear motion (Fig. 1).

The electric drive includes the servo-drive (SD) that is connected to the electrical three-phase grid controlling the rotational speed of the servo-motor (SM) and some accessory components that are not considered in detail in this study. The electric drive can be chosen based on the following steps [23]:

1) determine the drive requirements such as the torque, speed, power, control performance, and interfaces;
2) select the power unit/motor combination;
3) identify the control unit performance and interface (i.e., communication standards, input/outputs, and safety features);
4) define the firmware function (e.g., open-loop or closed-loop controller architecture);
5) select the accessories (e.g., mains filters and chokes, brake resistors/units, capacity modules, cables, and software).

The following part is the sizing of the servo-motor for both the EMC and the EHC. The servo-drive is selected to deliver a continuous output current i_{cont} such that:

$$i_{cont} \geq \frac{\tau_{cont}}{\tau_0} i_0, \tag{1}$$

where τ_{cont} is the required continuous torque, and τ_0 and i_0 are the motor's continuous torque and current at standstill, respectively. The continuous operating characteristic S1 (60K) according to EN 60034-1 is used when sizing the electric drive.

A. Electro-Mechanical Cylinder

The mechanical transmission system includes the screw assembly (i.e., ball screw or roller screw), the gearbox (optional), and the motor attachment (i.e., direct via flange and coupling, or via timing belt side drive). The main components of the EMC are illustrated in Fig. 1a. According to common industrial practice [22], the EMC is designed based on the following steps:

1) select the type of screw assembly and its dimensions (i.e., diameter and lead) based on the required cylinder stroke length, the average power and the dynamic load requirements of the considered working cycle, and the desired service life;
2) select the motor and gearbox combination based on the desired control performance, maximum speed, and continuous torque requirements.

These aspects are addressed in detail in the following paragraphs.

1) The screw assembly: The type of screw assembly (SA), diameter (d_0), lead (l), gear reductions (i_g), and the size of the electric motor must be selected to meet the minimum requirements without being overly conservative by stating the desired working cycle (i.e., the desired motion and resulting load profile) as accurately as possible. First, to avoid overheating of the mechanical transmission system, the average output power of the application must not exceed the permissible transmitted power of the screw assembly

$$\dot{E}_{emc,p} \geq \dot{E}_{c,avg}. \tag{2}$$

Secondly, the screw assembly is selected such that its given dynamic load capacity (C) is at least five times greater than the resulting average equivalent dynamic load ($F_{c,avg}$) to satisfy the nominal life calculation

$$L_{10,emc} = \frac{\left(\frac{C}{F_{c,avg}}\right)^3 l \cdot 10^3}{\dot{x}_{c,avg} \cdot 3.6}, \tag{3}$$

where $\dot{x}_{c,avg}$ is the average absolute linear velocity of the actuator.

2) The motor and gearbox: The motor and gearbox combination is selected such that the conditions addressed in (4)-(8) are satisfied.

The maximum speed of the electric machine is constrained by:

$$n_{emc,max} \geq \dot{x}_{c,max} \frac{i_g \cdot 60}{l}, \tag{4}$$

where $\dot{x}_{c,max}$ is the maximum required linear velocity.

The ratio of the load moments of inertia (J_R) serves as an indicator for the control performance of a motor/controller combination according to:

$$J_R = \frac{J_{mech}}{J_m} \leq \begin{cases} 6.0, & \text{handling} \\ 1.5, & \text{processing} \end{cases}, \tag{5}$$

where J_{mech} is the moment of inertia of the mechanical transmission system and the external load referred to the motor side, and J_m of the electric motor and brake unit (i.e., the passive load-holding device if needed). Industrial experience has shown that a suitable moment of inertia ratio will result in high control performance for different applications (e.g., handling or processing applications) [22].

The permissible force of the transmission system is constrained by:

$$F_{emc,p} \geq F_{c,max}, \tag{6}$$

where $F_{c,max}$ is the the maximum resulting load force acting on the cylinder.

The permissible torque of the transmission system is constrained by:

$$\tau_{emc,p} \geq \frac{F_{c,max}}{\mu_{emc}} \frac{l}{2\pi \cdot i_g}, \tag{7}$$

where μ_{emc} is the efficiency of the mechanical transmission.

The torque ratio must verify

$$\tau_R = \frac{\tau_{emc,r}}{\tau_{emc,cont}} \leq 1.0, \tag{8}$$

where $\tau_{emc,r}$ is the rated torque (i.e., the maximum continuously torque available at the continuous speed) of the servo-motor identified from the motor's characteristics curve provided by the manufacturer and $\tau_{emc,cont}$ is the continuous driving torque

$$\tau_{emc,cont} = \frac{F_{c,_{RMS}}}{\mu_{emc}} \frac{l}{2\pi \cdot i_g}, \tag{9}$$

where $F_{c,_{RMS}}$ is the root mean square (RMS) value of the cylinder's force profile. If an accurate load profile is not available, the manufacturer [22] proposes to use an empirical value $\tau_R \leq 0.6$ instead of $\tau_R \leq 1.0$. Lastly, the rated torque is considered at the continuous speed

$$n_{emc,cont} = \dot{x}_{c,_{RMS}} \frac{i_g \cdot 60}{l}, \tag{10}$$

where $\dot{x}_{c,_{RMS}}$ is the RMS of the cylinder's velocity profile.

B. Electro-Hydraulic Cylinder

In place of the mechanical transmission in the EMC, the EHC includes a hydraulic (hydrostatic) transmission with a fixed-displacement hydraulic pump/motor unit (P) driving the hydraulic cylinder and arranged in a closed-circuit configuration (Fig. 2). The hydraulic unit operates as a pump when the cylinder's piston is extending and as a motor when the piston is retracting — allowing the electric drive to regenerate power. The hydraulic auxiliary components such as load-holding valves, the pilot-operated check valves used for balancing the differential flow of the single-rod cylinder due to unequal areas, anti-cavitation valves, and the pressure relief valves against over-pressurization are placed in a manifold and installed directly on the hydraulic cylinder. A bladder-type accumulator (AC) represents the sealed reservoir, and a low-pressure return filter is connected to the circuit. Lastly, the electric motor is mounted directly to the manifold by a bell housing and a servo coupling. For more details about the considered hydraulic circuit, Padovani et al. [15] described its functioning and experimental testing.

The self-contained electro-hydraulic cylinder is designed based on the following steps:

1) size the cylinder's stroke capability according to the requirements of the application and size the piston and rod diameter based on the maximum load force and the buckling criteria;
2) size the pump/motor unit based on the displacement and speed required to deliver the demanded flow dictated by the actuator's desired motion profile;

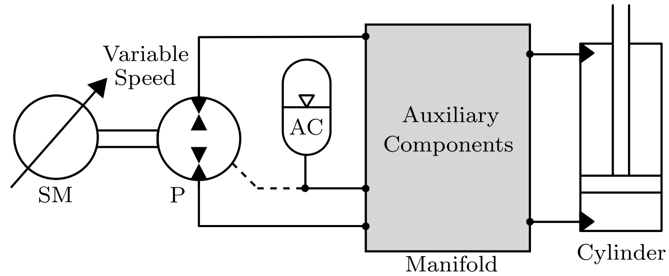


Fig. 2: A possible system architecture for self-contained electro-hydraulic cylinders (simplified schematic) [14].

3) size the electric drive in terms of maximum and continuous speed, torque, and current requirements;
4) size the hydro-pneumatic accumulator based on the exchange volume and the desired maximum and minimum pressures in the reservoir;
5) size the load-holding valves such that the pressure drop is kept minimal to maintain an efficient throttle-less system (i.e., increased throttling may result in the need for an oil cooler) and select a proper pilot ratio and cracking pressure to ensure functionality;
6) size the flow balancing valves and the oil filter such that the reservoir pressure is kept below the pressure limits of the hydraulic components to ensure proper functionality;
7) size the pressure-relief valves based on the maximum allowed pressures of the hydraulic components and on the force limitations of the cylinder to satisfy the buckling criteria.

Details about the above-mentioned procedure are shared in the following paragraphs.

1) Hydraulic cylinder: The buckling factor is the essential design parameter when sizing the hydraulic cylinder. According to existing regulations (DNVGL-CG-0128), the accepted criterion to avoid buckling is to use a safety factor $f_s = 4$. The cylinder dimensions can be chosen according to standards (e.g., ISO 6022) based on the minimum rod diameter

$$d_r \geq \sqrt[4]{\frac{64 f_s h_{max}^2 F_{c,max}}{\pi^3 E}}, \tag{11}$$

and minimum piston diameter

$$d_p \geq \sqrt{\frac{4}{\pi} \frac{F_{c,max}}{p_{p,ref} - p_{ac,max}\varphi}}, \tag{12}$$

where h_{max} is the maximum length of the cylinder in the extended position, E is the elasticity modulus (206 GPa for steel), $p_{p,ref}$ is the desired piston-side pressure (i.e., the desired operating pressure of the hydraulic pump), $p_{ac,max}$ is the highest accumulator pressure, and φ the ratio between rod and piston areas.

2) Pump/motor unit: When the size of the cylinder's piston area (A_p) is known and the maximum required velocity of the actuator ($\dot{x}_{c,max}$) is given, then the maximum flow demand

$$Q_{max} = A_p \dot{x}_{c,max}, \tag{13}$$

is used to calculated the required displacement (D_p) of the hydraulic machine. This step, based on the nominal operating speed (n_{nom}) and the estimated volumetric efficiency (η_{vol}) of the selected pump/motor unit, is determined according to:

$$D_P \geq \frac{Q_{max}}{n_{nom}} \frac{60}{\eta_{vol}}. \tag{14}$$

The lifetime of the axial piston machine (i.e., the roller bearings are the limiting factor) is estimated based on data given by the manufacturer as:

$$L_{10,ehc} = L_{10N} a_{23} C_0, \tag{15}$$

where L_{10N} is the nominal bearing life at nominal operating data and a_{23} and C_0 are the correcting factors for respectively the viscosity influence and the flow influence (swivel angle).

3) Servo-motor: The conditions addressed in (16) and (17) must be satisfied when sizing the electric machine.

They are the maximum speed

$$n_{ehc,max} \geq \frac{Q_{max}}{D_P} \frac{60}{\eta_{vol}}, \tag{16}$$

and the motor's continuous rated torque

$$\tau_{ehc,r} \geq \tau_{ehc,cont} = \frac{D_P}{2\pi} \frac{F_{c,RMS}}{A_p} \frac{1}{\eta_{mh}\eta_c}, \tag{17}$$

where $\tau_{ehc,cont}$ is the continuous driving torque. It is assumed that the rod-side pressure is insignificantly low, but the mechanical-hydraulic efficiency of the hydraulic pump/motor unit (η_{mh}), and the efficiency of the hydraulic cylinder (η_c) should be included for a conservative sizing. Lastly, the rated torque is considered at the continuous speed

$$n_{ehc,cont} = \frac{A_p \dot{x}_{c,RMS}}{D_P} \frac{60}{\eta_{vol}}. \tag{18}$$

4) Accumulator: The size of the bladder accumulator is derived according to the required effective gas volume

$$V_{ac,0} \geq C_a \frac{\Delta V}{\left(\frac{p_{0,(T_{min})}}{p_{ac,min}}\right)^{\frac{1}{\kappa}} - \left(\frac{p_{0,(T_{min})}}{p_{ac,max}}\right)^{\frac{1}{\kappa}}}, \tag{19}$$

where $C_a = p_{ac,max} / p_{ac,min}$ is the adiabatic correction factor, $\kappa = 1.4$ is the adiabatic exponent, ΔV is the total exchanged volume (i.e., the cylinder's differential volume $(A_p - A_r)x_{c,max}$ + 20% overhead), $p_{ac,max}$ and $p_{ac,min}$ are the desired maximum and minimum pressures. Lastly, based on the maximum and minimum ambient temperature (T_{max} and T_{min}), the gas precharge pressure at minimum ambient temperature ($p_{0,(T_{min})}$) is calculated as:

$$p_{0,(T_{min})} = 0.9 p_{ac,min} \left(\frac{T_{min}}{T_{max}}\right). \tag{20}$$

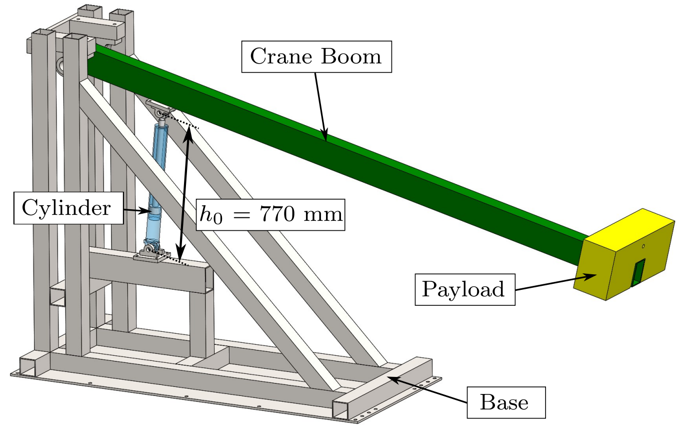


Fig. 3: The single-boom crane and the original hydraulic cylinder.

5) Auxiliary valves: The flow capacity of the load-holding valves must be greater than Q_{max}, while the flow balancing valves is sized based on the highest differential flow

$$\Delta Q_{max} = Q_{max} \left(1 - \frac{A_r}{A_p}\right). \tag{21}$$

The cracking pressure (p_{cr}) and pilot ratio (α_p) of the load-holding valves must be selected such that the highest pressure in the pilot line (i.e., the reservoir pressure) cannot open the pilot-operated check valves, according to:

$$p_{cr} > \alpha_p p_{max}. \tag{22}$$

Furthermore, a vented pilot-operated check valve is preferable to ensure complete opening of the load-holding valves [15]. For a detailed description of the auxiliary valves, their function, and selected components, see [15].

III. Design Analysis

This section describes the design analysis carried out to select suitable components for the EMC and the EHC; the single-boom crane depicted in Fig. 3 is used as the case study. The heavy-duty version (EMC-HD) from Rexroth (Fig. 5a) [22] is chosen as an exemplary drive due to its specific characteristics in terms of accuracy, dynamics, controllability, and heavy load capability. According to the survey carried out in [24], this type of electro-mechanical linear actuators was considered as one of the most relevant commercial EMC for load carrying applications (see, for instance, [24] for more detail description of the EMC-HD). Consequently, the same manufacturer is also chosen for the main components of the EHC (i.e., the servo-drive [25], the servo-motor [26], the axial piston machine [27], and the accumulator [28]), as illustrated in Fig. 5b.

A. Actuator Requirements

The nonlinear model of the single-boom crane is described using the Newton's second law for the actuator dynamics

$$M_{eq}(x_c)\ddot{x}_c = F_c - F_{eq}(x_c), \tag{23}$$

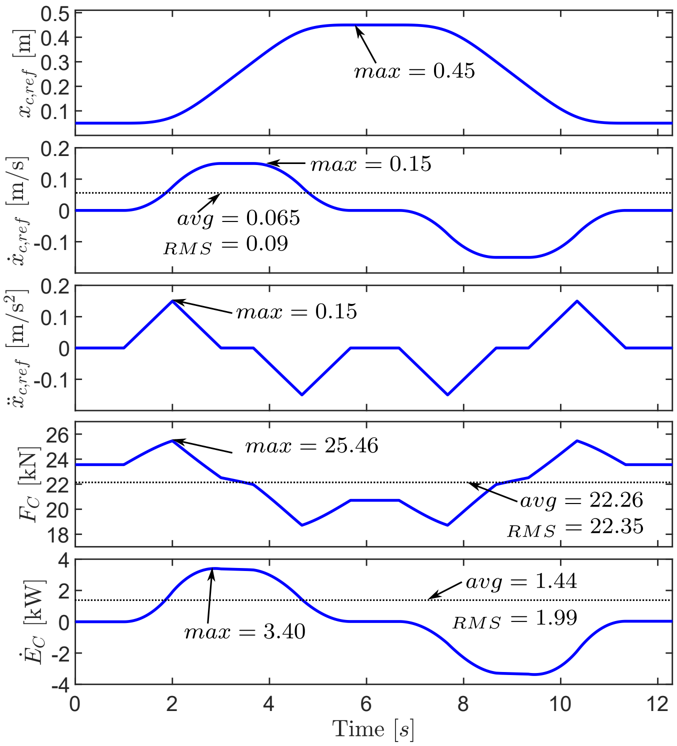


Fig. 4: Key magnitudes of the considered working cycle.

where $\ddot{x}_c$ is the acceleration of the equivalent mass (M_{eq}), F_c is the mechanical force delivered by the linear actuator and F_{eq} is the equivalent gravitational force as a function of the piston position (x_c).

The motion profile generator presented in [29] is used to generate motion reference signals (i.e., desired position ($x_{c,ref}$), velocity ($\dot{x}_{c,ref}$), and acceleration ($\ddot{x}_{c,ref}$) of the linear actuator). Further on, the required cylinder force and output power are defined by the following equations

$$F_c = M_{eq}(x_{c,ref})\ddot{x}_{c,ref} + F_{eq}(x_{c,ref}), \tag{24}$$

$$\dot{E}_c = F_c \dot{x}_{c,ref}. \tag{25}$$

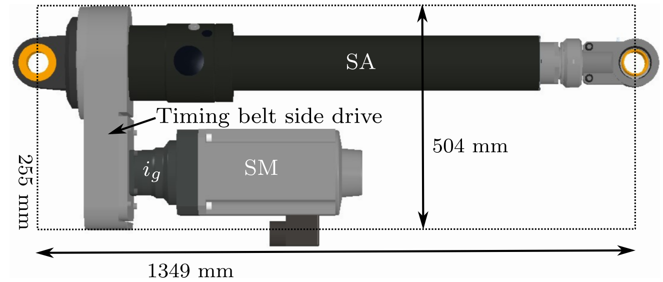
The identified maximum motion and load requirements that the two linear actuation systems must satisfy for the considered working cycle are are highlighted in Fig. 4 together with the average and RMS values.

B. Selection of Electro-Mechanical Components

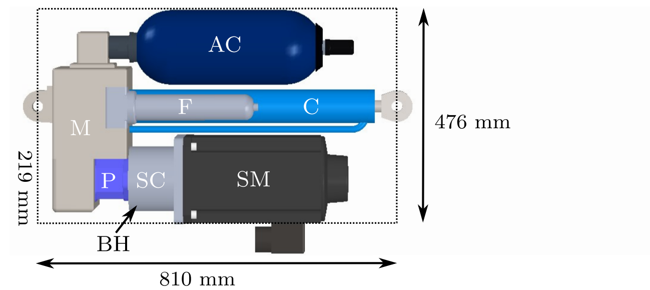
The screw assemblies that satisfy the permissible average power (2) and have a dynamic load capability $C > 5 \cdot F_{c,avg}$ for the considered working cycle are listed in Tab. I.

TABLE I: Suitable screw assemblies.

SA	$\mathbf{d_0}$ [mm]	C [kN]	$\mathbf{F_P}$ [kN]	$\mathbf{L_{10}}$ [h]	$\mathbf{h_0}$ [mm]
105	50	117	56	11,873	1274
125	63	131	62	16,804	1349
150	80	307	115	217,267	1586



(a) Electro-mechanical cylinder.



(b) Electro-hydraulic cylinder.

Fig. 5: Installation size of the two self-contained cylinders.

The diameter of the screw, the dynamic load capacity, the maximum permissible force (F_p), the calculated nominal life (L_{10}) derived in (3), and the initial length (h_0) of the cylinder when fully retracted are also listed. All the suitable assemblies are ball screws with a lead of $l = 20$ mm, and a limited permissible continuous transmitted power of $\dot{E}_{emc,p} = 2$ kW to ensure that the mechanical transmission will not overheat during continuous operation.

The initial length of the EMC-HD 150 is too long to fit inside the base of the single-boom crane (Fig. 3). Also, only the timing belt side drive is considered since the direct mounting adds additional installation length. Hence, the suitability, when considering the conditions mentioned earlier in (4)-(8) for the available motor and gear combinations from the catalog [22], is considered for the ball screw assembly with a diameter of 50 mm (HD 105) and 63 mm (HD 125), and further analyzed in Tab. II. Three suitable configurations result from the analysis mentioned above. Further on, the configuration, including the smallest (i.e., the cheapest) servo-motor requiring the smallest servo-drive, is chosen. The selected EMC includes the HD 125 ball screw assembly [22], the MSK100B-0300 servo-motor with integrated load-holding brake [26] mounted on a timing belt side drive together with the gearbox resulting in an overall gear ratio $i_g = 4.5$. The chosen combination requires an continuous current $i_{cont} \geq 11.8$ A. Hence, the IndraDrive C HCS02.1E-W0054 [25] with $i_{cont} = 22$ A is selected, resulting in a nominal power (i.e., installed power) of $\dot{E}_{in,nom} = 7.5$ kW.

C. Selection of Electro-Hydraulic Components

The dimensions of the hydraulic cylinder are already given by the real application (Fig. 3) used as a case study. The

TABLE II: Analysis of the available electro-mechanical cylinder configurations with timing belt side drive.

SA HD	**SM** MSK	$\mathbf{i_g}$ [22] [-]	μ [22] [-]	$\tau_\mathbf{p}$ [22] [Nm]	$\dot{\mathbf{x}}_\mathbf{max}$ [22] [m/s]	$\mathbf{n_{cont}}$(10) [rpm]	$\tau_\mathbf{cont}$(9) [Nm]	$\tau_\mathbf{r}$ [26] [Nm]	$\mathbf{i_{cont}}$(1) [A]	$\mathbf{m}$ [22] [kg]	$\mathbf{J_R}$(5) [-]	$\tau_\mathbf{R}$(8) [-]	**OK?**
105	071D	1.5	0.87	136.1	1.00	405	54.5	16	28.4	103	30.9	3.41	No
105	071D	4.5	0.83	47.7	0.33	1215	19.1	13	9.9	111	**3.85**	1.47	No
105	071D	7.5	0.83	28.6	0.20	2025	11.4	10	5.9	111	**1.61**	1.14	No
105	100B	1.5	0.87	136.1	1.00	405	54.5	25	33.9	120	**3.92**	2.18	No
105	101D	1.5	0.87	136.1	1.00	405	54.5	41	33.4	127	6.50	1.33	No
105	101E	1.5	0.87	136.1	1.00	405	54.5	53	32.4	141	**4.77**	1.03	No
125	071D	7.5	0.83	43.5	0.20	2025	11.4	10	5.9	158	**1.75**	1.14	No
125	100B	1.5	0.87	164.9	0.80	405	54.5	22	33.9	168	**4.40**	2.18	No
125	100B	4.5	0.83	72.5	0.33	1215	19.1	21	11.8	175	**0.54**	**0.91**	**Yes**
125	101D	1.5	0.87	164.9	0.80	405	54.5	41	33.4	175	7.29	1.33	No
125	101D	4.5	0.83	72.5	0.33	1215	19.1	28	11.7	183	**0.90**	**0.68**	**Yes**
125	101E	1.5	0.87	164.9	0.80	405	54.5	53	32.4	188	**5.35**	1.03	No
125	101E	4.5	0.83	72.5	0.33	1215	19.1	24	11.3	196	**0.66**	**0.79**	**Yes**

TABLE III: The selected components for the EHC.

	Characteristics	**Requirements**	**Size**	**Model**
C	A_p [m^2]	–	0.0033	PMC
	A_r [m^2]	–	0.0024	25CAL
	η_{vol}	–	0.9	
	m_c [kg]	–	20.1	
SM	n_{max} [rpm]	2968	4500	MSK
	τ_r [Nm]	15	18	100B-0300
	m_{sm} [kg]	–	34	[26]
SD	$\dot{E}_{in,nom}$ [kW]	3.82	4	HCS02.1E
	i_{cont} [A]	9.3	11	W0028 [25]
P	D_P [$\frac{cm^3}{rev}$]	8.74	10.6	A10FZG
	n_{nom} [rpm]	2968	3600	[27]
	η_{vol}	–	0.95	
	η_{mh}	–	0.84	
	m_p [kg]	–	9	
AC	V_0 [L]	5.32	5.9	HAB6
	m_{ac} [kg]	–	20	
	m_{oil} [kg]	–	10	ISO VG 46
F	m_f [kg]	–	2	50LEN0100 [30]
M	m_m [kg]	–	29.6	Custom
BH	m_{bh} [kg]	–	3.5	PSG-200
SC	m_{sc} [kg]	–	1.5	Rotex-GS28

hydraulic cylinder has piston diameter $d_p = 65$ mm, rod diameter $d_r = 35$ mm, initial length when the cylinder is fully retracted $h_0 = 770$ mm, and stroke length $x_{c,max} = 500$ mm.

Based on the identified minimum requirements according to the sizing principles presented in Section II, the components' sizes are selected from the catalogs and presented in Tab. III. The following values are used when sizing the accumulator: $p_{max} = 1.0$ bar, $p_{min} = 0.5$ bar, $T_{max} = 50$ °C, and $T_{min} = -20$ °C. The mass of the mineral oil, filter (F), manifold (M), bell housing (BH), and servo coupling (SC) are also included.

The nominal bearing life of the axial piston machine is, according to the diagram provided by the manufacturer, equal to 30,000 h for nominal operating pressures below 200 bar. The viscosity correcting factor is $a_{23} = 2.5$ for the considered hydraulic fluid (i.e., ISO VG 46), and the flow correcting factor is $C_0 = 1$ for a fixed displacement unit. Derived in (15), the estimated life of the pump/motor unit is 75,000 h. According to the manufacturer, values above 30,000 h are not accurate because other factors influence the lifetime of the axial piston machine.

IV. Comparative Analysis and Discussion

Both the considered EMC and EHC can continuously perform the required motion in the considered case study. However, the final selection depends on additional design objectives, such as installation space, weight, service life, ingress protection classification, energy efficiency, power density, control performance, and cost. Environmental factors such as temperatures, ingress protection, and potentially explosive atmospheres (ATEX), are omitted in the considered design requirements.

In regards to the controllability, this study does not evaluate the motion tracking performance (accuracy) since that would require an experimental investigation to have a fair comparison. However, it is generally known that the driving stiffness of EMC is significantly high compared to the EHC [9]. To accurately control the linear speed and position of the EMC, the standard control architecture, including a position controller in cascade with the speed controller (FOC), implemented in the servo-drive using the angular encoder initially included on the servo-motor as feedback is sufficient. For the EHC, instead of using the angular encoder as position feedback, an additional linear transducer (position sensor) must be implemented on the hydraulic cylinder for feedback to supervise to cylinder's piston position. The EHC's position controller can be implemented on an embedded controller, providing a reference speed signal to the servo-drive [20], or a more complex control firmware [31] can be installed on the servo-drive. Furthermore, previous research related to a similar EHC [20] shows that the uncompensated hydraulic system suffers from very low damping. Consequently, extra sensors are needed to implement active damping (e.g., pressure sensors for pressure feedback), that is necessary to achieve satisfactory motion performance.

Based on the selected components and on the maximum values (force and power) of the working cycle (Fig. 4),

TABLE IV: Performance characteristics of the self-contained cylinders compared to the valve-controlled cylinder.

Characteristics	VCC	EMC	EHC
Length [mm]:	770	1349 (+75.2%)	810 (+5.2%)
Volume [L]:	4.8	173.4 (+3513%)	84.4 (+1658%)
Installed mass: [kg]:	30	175 (+483%)	130 (+333%)
Installed power [kW]:	10.5	7.5 (-29%)	4 (-62%)
Force pr mass [N/kg]:	852	146 (-83%)	197 (-77%)
Power density [W/L]:	708	19.6 (-97%)	40.3 (-94%)
Energy efficiency [%]:	22	60 (+173%)	57 (+159%)
Max force [kN]:	82.5	62 (-25%)	82.5 (0%)
Max power [kW]:	5.1	2 (-61%)	2.4 (-53%)
Max service life [h]:	30,000	16,804 (-44%)	30,000 (0%)
Enclosure rating:	IP65	IP65	IP65

Tab. IV compares relevant characteristics of the two designed self-contained drive systems relative to the valve-controlled cylinder (VCC) presented in [20], [21], [32]. It is assumed that the valve-controlled system can deal with the required piston velocity of the motion profile. The energy efficiencies experimentally identified in [21] for a similar EHC and the considered VCC driving the same application with a similar motion cycle is used for comparison. For the EMC, the efficiency of the electric drive identified in [9] is used in combination with the efficiency of the selected mechanical transmission system according to [22]. As a side note, the energy efficiency of the two self-contained cylinders can be significantly increased if the energy regenerated when lowering the crane boom is recovered [21].

The quantities in Tab. IV and the illustration in Fig. 5 clearly show that the EHC takes less space and weights less, making it more suitable for applications where conventional hydraulic cylinders are used, such as for the considered single-boom crane. Hence, the EHC performs better in terms of force per mass (35% better) and power density (106% better). The EMC can move 0.15 m/s faster than the EHC and permits faster acceleration (114%), where the pump/motor unit is the limiting component. Morover, the EMC has 5% better energy efficiency than the EHC.

The estimated service life of the EHC is almost twice compared to the EMC. In general the EMC has low overload protection compared to hydraulic alternatives. This is the case for the mechanical system since it is very stiff and cannot absorb any additional external impact force. Consequently, the screw transmission may fail (jamming) and the EMC must be replaced to resume operations. The EMC has a higher price on the electric drive that has more installed power than the EHC. From experience, the cost of the EMC's screw assembly is believed to be about twice as expensive as the hydraulic transmission system, due to the high price resulting from low production numbers.

Finally, to evaluate the impact of moving from conventional hydraulic cylinders to self-contained cylinders, a comparison between EMC and EHC concerning conventional valve-controlled systems (without considering the space occupation, cost, and energy losses of the centralized hydraulic power unit) [32] is presented in Tab. V. Additional general knowledge from the technical literature is also included. This comparison is intended to be a general guideline for choosing between EMC or EHC.

TABLE V: Advantages and disadvantages when replacing valve-controlled cylinders with self-contained solutions. Five different grades are used, ranging from (– –) to (+ +), with (– –) being the worst, and 0 representing similar performance as the conventional VCC. Results that are based on general knowledge from the technical literature is denoted by *, while results that are related to ongoing research is denoted by ?.

Category:	Criterion:	EMC	EHC
Design	Compactness	– –	–
	Force per mass	– –	–
	Power density	– –	–
	Design complexity	+	–
	Enclosure protection	0	0
	Control effort*	+	0
	Scalabillity*	– –	–
	Cost	– –	–
Operation	Impact absorption*	– –	0
	Reliability*	– –	0?
	Energy efficiency	++	++
	Thermal absorption	– –	–?
	Accuracy*	++	+
	Drive stiffness*	++	–
	Max force	–	0
	Max velocity	++	+
	Max acceleration	++	–
	Max cont. output power	– –	–
Safety	Passive load-holding	0	0
	Fail-safe*	–	0
	Overload protection*	–	0
Application	Installed power	+	++
	Oil spill risk*	++	+
	Maintenance effort*	++	+?
	Durability*	– –	0?
	Commissioning effort	++	+

V. Conclusions

Two self-contained linear actuator technologies, namely an electro-mechanical and an electro-hydraulic cylinder, are investigated in this paper; the focus is on their design when replacing a traditional valve-controlled hydraulic cylinder. After explaining why they are potential alternatives for many state-of-the-art applications, a single-boom crane is chosen as the baseline to apply these drives. The sizing process to select available off-the-shelf components is illustrated and relevant characteristics of the systems are discussed. The following main aspects are, therefore highlighted:

- The electro-hydraulic drive shows several benefits over the electro-mechanical counterpart such as 20% higher continuous power capability, 47% less installed electric power, 79% longer expected service life, 33% higher maximum force capability, 25% less overall mass, and 40% less installation length.
- The electro-hydraulic cylinder is more robust against impact forces, and is expected to have about 50% lower initial cost. For working cycles requiring a continuous

transmitted power above 2 kW, there are no available configurations of the considered electro-mechanical actuator due to the limitations on the permissible continuous power being transmitted by the screw assembly. For the electro-hydraulic solution, the rated torque of the servo-motor is the limiting factor. However, asynchronous induction machines are available when higher torque is needed, i.e., above 180 Nm, as pointed out in [33].

- The electro-mechanical solution is expected to show better controllability due to higher drive stiffness, requires less control effort, has higher energy efficiency, and allows for lower system complexity resulting in a more straightforward design approach.

This analysis also shows that the electro-hydraulic solution is the best choice when there exists a risk for high impact forces, when the required output power is continuously above 2 kW, and when minimal installation space, weight, and cost are key design objectives. Concerning future work, effort will be placed on making the sizing procedure automated in terms of component selection.

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