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

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***Abstract*—This research paper presents guidelines 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 a risk for high impact forces is present, when the required output power level lies continuously above 2 kW, or when installation space, weight, and cost are critical design objectives. However, the electro-mechanical solution is expected to show more controllability due to the fact that it has higher levels of drive stiffness, and energy efficiency as well as lower system complexity. This solution also requires less effort to control the actuator's linear motion accurately. All of these factors result 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**

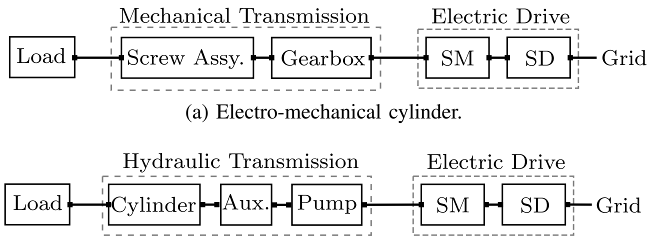
I. Introduction

Traditional valve-controlled linear hydraulic actuators can be implemented in many architectures [1]. They are characterized not only by reliability, high force and power capability, and excellent overload protection (e.g., shock absorption) but also relevant power losses due to flow throttling in the valves, demanding efforts needed for installation and maintenance, and costly piping due to centralized hydraulic power supplies. Yet in spite of their disadvantages, these actuators are still commonplace in industry.

Over the course of several years, efforts have been made to replace these conventional hydraulic systems with electro-mechanical cylinders (EMCs). This has been a popular research topic and engineering task, especially in aerospace systems. To date, a great deal of work has been done to reduce the system's weight, installation time, and maintenance requirements [2]–[6]. However, predicting this system's service life is challenging, and there is a considerable risk of failure (e.g., jamming, which can result in dangerous consequences) since the actuator cannot absorb shocks in standby mode. Consequently, these actuators are still not accepted in commercial airplanes as primary actuators for flight control. Alternatively, electro-hydraulic cylinders (EHCs) that are self-sufficient and completely sealed have been introduced. These cylinders typically contain a dedicated variable-speed pump that controls the cylinder's motion. This technology, as explained by [7], was first introduced in aerospace systems, demonstrating the idea that the disadvantages of traditional hydraulics can be significantly reduced. 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 in order to increase energy efficiency and eliminate oil spills. Subsequently, the idea of combining EMC's advantages and hydraulics has been further developed in recent years to compete against EMCs themselves [9]–[15]. For instance, Michel and Weber created a prototype and compared it with a commercial EMC containing 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, proving it to be a valid alternative to the EMC. One unusual aspect of the EHCs, namely their thermal behavior, remains an ongoing research topic [16]–[18], as does their durability [19]. An EHC concept comprised of passive load-holding devices, a sealed reservoir, four quadrants operations, capable of recovering energy, and suitability 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 of this concept with a focus on motion performance [20] and energy efficiency [21], showing that EHCs significantly improve 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 lacking 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 actuators being considered (Fig. 1) are the off-the-



(a) Electro-mechanical cylinder.

(b) Electro-hydraulic cylinder.

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

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 a single-boom crane's motion control is studied, and the sizing process of the 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 a number of accessory components which have not been considered in detail in this study. The electric drive can be chosen based on the following steps [23]:

1) determine the drive requirements, including 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 section shows the sizing of the servo-motor for both the EMC and EHC. The servo-drive has been selected to deliver a continuous output current i_{cont} so 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 has been 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 either flange and coupling or timing belt side drive). The main components of the EMC are illustrated in Fig. 1a. According to common industrial practice [22], the EMC has been 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, average power and dynamic load requirements of the considered working cycle in addition to 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. They should state the desired working cycle (i.e., desired motion and resulting load profile) as accurately as possible. First, to avoid overheating the mechanical transmission system, the application's average output power must not exceed the screw assembly's permissible transmitted power

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

Secondly, the screw assembly is selected in order that its given dynamic load capacity (C) will be at least five times greater than the resulting average equivalent dynamic load ($F_{c,avg}$), which will 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 in order that the conditions addressed in (4)-(8) will be 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 an appropriate 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 level 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 continuous 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 using 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 being 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, which include load-holding valves, pilot-operated check valves used for balancing the differential flow of the single-rod cylinder due to uneven areas, anti-cavitation valves, and pressure relief valves used to counteract 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 servo coupling. For more details about the considered hydraulic circuit, Padovani et al. [15] have described its functioning and experimental testing.

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

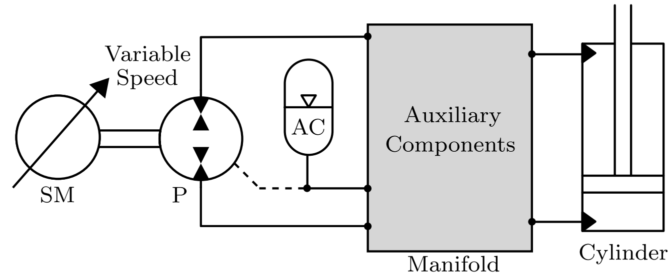


Fig. 2: A possible system architecture for self-contained electro-hydraulic cylinders (simplified schematic) [14].

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 buckling criteria;
2) size the hydraulic pump/motor unit based on the displacement and speed required to deliver the required flow dictated by the actuator's desired motion profile;
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 pressure levels in the reservoir;
5) size the load-holding valves so that the pressure drop will be kept minimal to maintain an efficient throttleless 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 oil filter so that the reservoir pressure will be kept below the pressure limits of the hydraulic components to ensure proper functionality;
7) size the pressure-relief valves based on the maximum allowed pressure levels of the hydraulic components and on the force limitations of the cylinder in order to satisfy the buckling criteria.

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

1) Hydraulic cylinder: The buckling factor is an 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) Hydraulic 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 unit. 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 provided 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 the viscosity influence and the flow influence (swivel angle), respectively.

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 pressure levels. Lastly, based on the maximum and minimum ambient temperature (T_{max} and T_{min}), the gas precharge pressure at the minimum ambient temperature ($p_{0,(T_{min})}$) is calculated as:

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

5) Auxiliary valves: The flow capacity of the load-holding valves must be greater than Q_{max}, while the flow balancing valves are 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 in order that the highest reservoir pressure (i.e., the pressure connected to the pilot line when motion is not desired) cannot open the pilot-operated check valves, according to:

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

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

III. Design Analysis

This section describes the design analysis carried out for the purpose of selecting suitable components for the EMC and 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] has been 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 to be one of the most relevant commercial EMCs for load-carrying applications (see, for instance, [24] for a more detailed description of the EMC-HD). Consequently, the same manufacturer has also been 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.

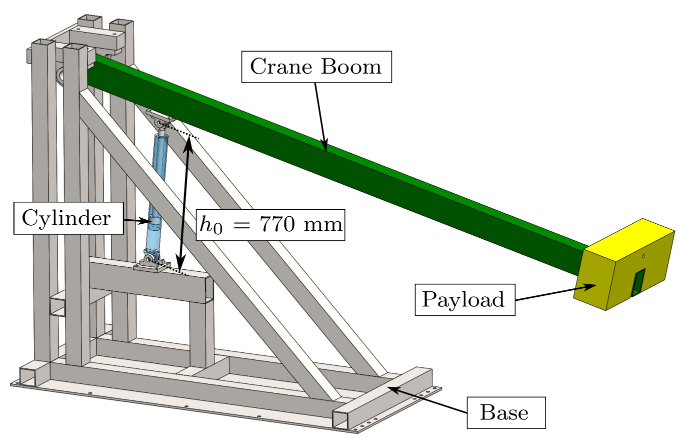


Fig. 3: The single-boom crane and the original hydraulic cylinder.

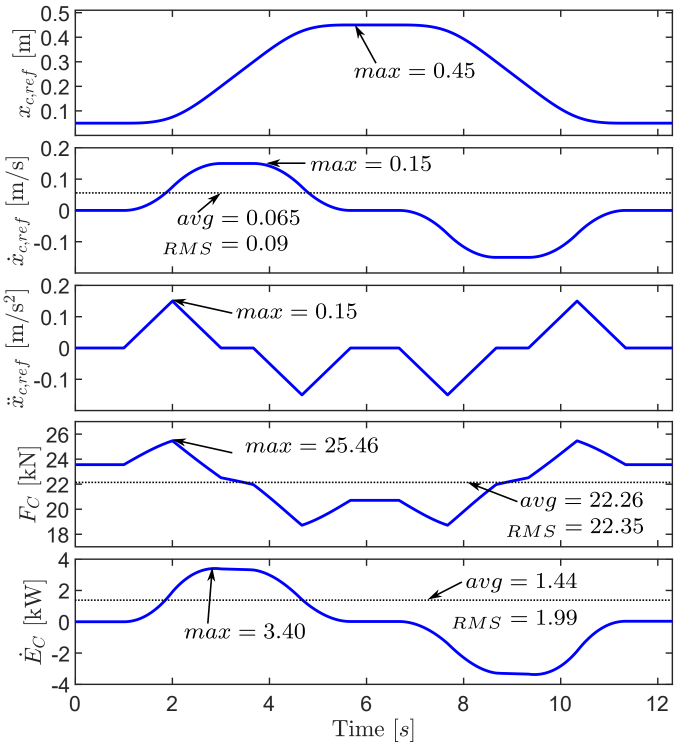
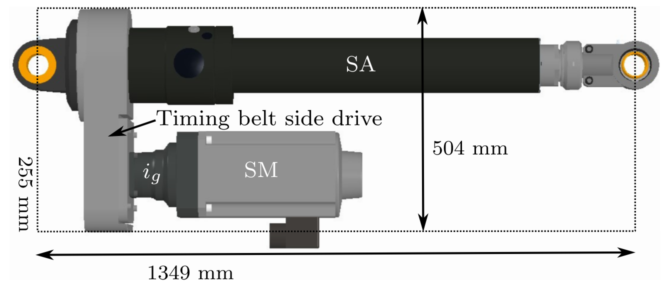
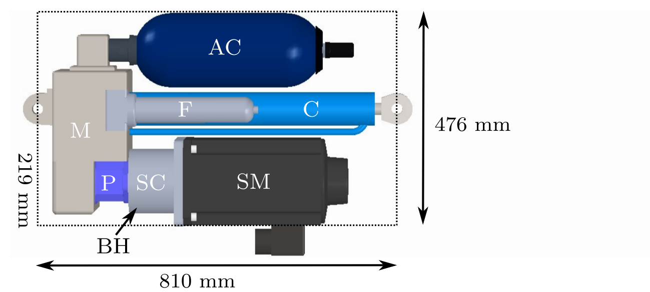


Fig. 4: Key magnitudes of the considered working cycle.



(a) Electro-mechanical cylinder.



(b) Electro-hydraulic cylinder.

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

A. Actuator Requirements

The nonlinear model of the single-boom crane is described using the Newton's Second Law of Motion for the actuator dynamics:

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

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 highlighted in Fig. 4 along 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.

The diameter of the screw, dynamic load capacity, maximum permissible force (F_p), calculated nominal life (L_{10}) derived in (3), and the initial length (h_0) of the fully retracted cylinder 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). Further, only the timing belt side drive has been 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], has been 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 have resulted from the analysis mentioned above. Subsequently, the configuration, including the smallest (i.e., the cheapest) servo-motor requiring the smallest servo-drive, has been 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 along with the gearbox and resulting in an overall gear ratio $i_g = 4.5$. The chosen combination requires a

TABLE I: Suitable screw assemblies.

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

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]	**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**

continuous current $i_{cont} \geq 11.8$ A. Hence, the IndraDrive C HCS02.1E-W0054 [25] with $i_{cont} = 22$ A has been 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 have already been given by the real application (Fig. 3) used as a case study. The hydraulic cylinder has a 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 in accordance with the sizing principles presented in Section II, the components' sizes have been selected from the catalogs and presented in Tab. III. The following values have been 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.

According to the diagram provided by the manufacturer, the nominal bearing life of the axial piston machine is 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 hydraulic 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.

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 [28]
	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

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, including 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), have been omitted in the considered design requirements.

Regarding controllability, this study does not evaluate motion tracking performance (accuracy) since that would require completing an experimental investigation in order to make a fair comparison between the two factors. However, it is generally known that the driving stiffness of EMC is significantly high compared to that of the EHC [9]. To accurately control the linear speed and position of the EMC, it is sufficient to have 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. In the case of 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 the 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]; alternately, 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), which is necessary to achieve satisfactory motion performance.

Based on the selected components and maximum values (force and power) of the working cycle (Fig. 4), 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 motion profile's required piston velocity. The energy efficiencies experimentally identified in [21] for a similar EHC, along with the considered VCC driving the same application with a similar motion cycle, are used for comparison. As regards 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 not only does the EHC take less space, but it also weighs less, making it more suitable for applications where conventional hydraulic cylinders are used, including 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 hydraulic 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 that of 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); if this occurs, 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. Based on 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, in order 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 here. This comparison is intended to act as a general guideline for choosing between EMC or EHC.

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

TABLE V: Advantages and disadvantages of 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 to the conventional VCC. Results that are based on general knowledge from the technical literature are denoted by *, while results that are related to ongoing research are 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 continuous output power	– –	–
Safety	Passive load-holding	-?	0
	Fail-safe*	–	0
	Overload protection*	–	+
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 electro-hydraulic cylinder, have been investigated in this paper; whose 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 has been chosen as the baseline for applying these drives. The sizing process to select available off-the-shelf components has been illustrated, and relevant characteristics of the systems have been discussed. The following main aspects are, therefore highlighted:

- The electro-hydraulic drive shows several benefits over the electro-mechanical counterpart, for example 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 around a 50% lower initial cost. In the case of 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. Regarding the electro-hydraulic solution, the rated torque of the servo-motor is a 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 a higher level of controllability due to higher drive stiffness and energy efficiency as well as lower system complexity. All of these factors result, in a more straightforward design approach.

This analysis also shows that the electro-hydraulic solution is the best choice when a risk for high impact forces exists, the required output power is continuously above 2 kW, and when minimal installation space, weight, and cost are key design objectives. Future work in this field will include efforts to make the sizing procedure automated in terms of component selection.

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