

Evaluation of a New Energy Recycling Hydraulic Lift Cylinder for Forwarders

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Abstract

In mechanized forestry, much of the work is conducted by use of cranes, and recovering potential energy is a possible method to reduce energy consumption when using cranes for lift work. The objective of this study was to evaluate the capacity of a new »Energy-efficient hydraulic lift cylinder« (EHLC), which has a secondary cylinder built into its piston rod, to store potential energy from lowering the boom in the form of pressurized hydraulic oil in an accumulator and using the stored energy in the next boom lift. The EHLC was mounted on a forwarder, and manipulated to enable its use also as a standard cylinder. We then compared the EHLC and a standard cylinder in terms of function and energy consumption during repetitive boom lifts and lowerings. With the tested settings the EHLC saved up to approximately 9.4% of the energy consumed during the first part of boom lifts and up to 3.2% of the total lift energy. With possible further adjustments, such as optimization of the accumulator size, enlargement of the assisting cylinder diameter, and enhancement of the accumulator pressurization, but most importantly reduction in internal leakage, the current EHLC could have commercial potential.

Keywords: weight-balancing, fluid dynamics, fluid mechanics, timber loader, mobile hydraulic lift devices, Boyle's law, counterweight

1. Introduction

Increases in energy costs and environmental concerns have intensified efforts to improve energy efficiency recently, both generally and specifically in engineering research (e.g. European Union 2014, United Nations 2014). Notably, several recent studies have addressed possible methods to improve the productivity of cranes used in harvesters, planting machines and forwarders (Lindroos et al. 2008, Jundén et al. 2013, Laine and Rantala 2013, Ersson et al. 2014, Ortiz Morales et al. 2014). Cranes are used primarily for lift work, in many mobile and stationary applications, which involves relocating objects in such a manner that their potential energy changes. Consequently, recovering potential energy can probably be used to reduce the energy required for the work (e.g. Liang and Virvalo 2001a, Sun and Virvalo 2003, Rydberg 2005, Sun and Virvalo 2005, Virvalo and Sun 2005, Lin et al. 2010, Lin and Wang 2012, Minav et al. 2012, Noréus et al. 2013, Wang et al. 2013). Forwarder cranes are designed to provide large lifts and heights, partly at the

cost of slow horizontal movements (Malmberg 1981, Gerasimov and Siounev 1998, 2000, Virvalo and Sun 2005). Knuckleboom cranes are normally used on forwarders, and consist of a system of hydraulic cylinders and mechanical levers, i.e., a swivelling crane pillar, pivoting mid and outer booms, and an extension boom (Gerasimov and Siounev 2000). While boom is often used as a synonym for crane, that usage is avoided in this paper to avoid confusion between the system and its components (cf. Lindroos et al. 2008).

The action of a vertical lift is mainly executed by an angular change in the joint between the crane pillar and mid-boom, which causes the outer and extension booms to rise. However, in this paper the outer and extension booms are treated as rigid parts of the mid-boom. Thus, boom is used hereafter as a collective term for the mid-, outer and extension booms, with the understanding that lifts are executed via action at the joint between the crane pillar and mid-boom.

During forwarder work logs are collected in the forest and carried to roadside landings. Typically, for-

warder productivities range from 11 to 25 m³ per productive machine hour, and are negatively correlated with transportation distance, while positively correlated with sizes of both logs and loads (Eriksson and Lindroos 2014). Logs are lifted while loading and unloading the forwarder's bunk. The total time required to forward a load (for which loading and unloading collectively account for about 60%) is typically 45 minutes (Manner et al. 2016). Typically, loading and unloading will require approximately 30 and 20 lifts with a full grapple, respectively (Manner et al. 2013, Manner et al. 2016).

The boom on a standard forwarder crane is lifted with a single acting cylinder. Pressurized oil is directed into the cylinder, creating a force that causes the piston rod to lift the boom. To lower the boom, the pressurized oil from the cylinder is released into the non-pressurized reservoir, and no energy is recovered. However, there are several possible methods to save energy, for example, through weight-balancing, a common principle for recovering potential energy during load lowering for cranes and elevators. In a weight-balancing system, some of the potential energy is recovered and stored during load lowering, and then used to assist the next load lift. The potential energy recovery process creates a braking force that reduces the load lowering speed. Thus, the weight-balancing is a trade-off between the additional lift force and braking force. In a fully balanced system, ignoring energy losses through friction, it is theoretically possible to recover almost all of the potential energy, allowing a load to be lifted and lowered with minor energy input. Other examples of the weight-balancing technique for mobile devices are the use of counterweights or coil springs (e.g. Gawlik and Michałowski 2008, Deepak 2012, Lin et al. 2013).

The few available energy recovering lift applications for forest machine cranes are typically based on the use of a hydro-pneumatic accumulator tank (accumulator), a common weight-balancing technique (e.g. Liang and Virvalo 2001a, Liang and Virvalo 2001b, Sun and Virvalo 2003, Sun and Virvalo 2005, Virvalo and Sun 2005). An accumulator consists of a vessel and bladder which separates an inert gas (e.g. nitrogen) from hydraulic oil. The flow of pressurized oil into the accumulator charges the accumulator as the sealed inert gas compresses according to Boyle's law and, similarly, the flow of oil out from the accumulator discharges the accumulator and releases the stored energy (during boom lift and boom lowering, respectively, in crane work). Fast charging and discharging are some of the advantages of accumulators for energy storage (Hui and Junqing 2010, Minav et al.

2012, Van de Ven 2013). Moreover, accumulators reduce pressure spikes in the hydraulic system (e.g. Malmberg 1981, Ingvast 1989, Kim et al. 2013, Van de Ven 2013). Experiments show that 21–59% of potential, kinetic or rotational energy can be recovered by using accumulators (Zhang 2011, Ho and Ahn 2012). However, an energy recovery system based on an accumulator also has limitations. Notably, in most systems the accumulator pressure must exceed the pressure in the hydraulic circuit to enable reuse of the stored energy (cf. Einola 2013), but the accumulator pressure might decrease below this pressure due, for example, to internal leakage somewhere, which is likely to occur in all hydraulic systems (Manring 2005). Thus, leakage compensation to re-pressurize the accumulator is required. Another limitation is the low energy storage capacity in relation to their size (e.g. Van de Ven 2013).

2. Materials and methods

2.1 The energy-efficient hydraulic lift cylinder (EHLC)

Technical principles and claims for a flawlessly functioning EHLC (Fig. 1) mounted on a forwarder

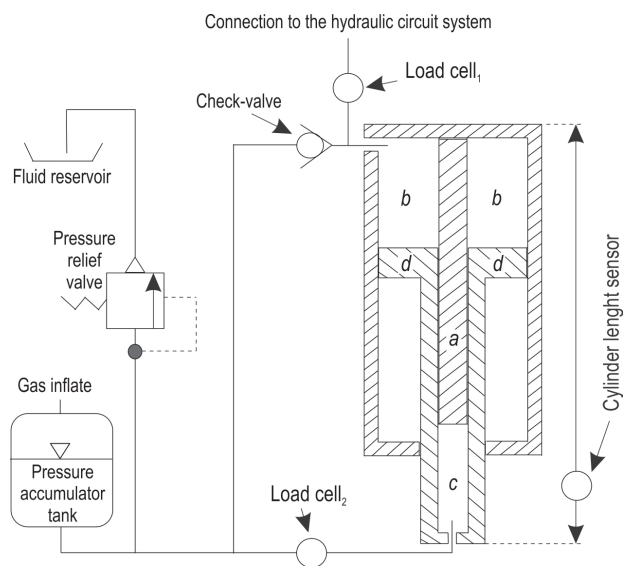


Fig. 1 Thordab AB's patented »Energy-efficient hydraulic lift cylinder« (EHLC) with a pressure accumulator tank (accumulator), a movably arranged secondary piston (a) that divides the cylinder system into primary (b) and secondary cylinders (c). The secondary cylinder is built inside the primary piston rod (d) and connected to the accumulator. Load cell₁ (p_1) measured the pressure in the standard cylinder and primary cylinder of the EHLC. Load cell₂ (p_2) measured the secondary cylinder pressure and accumulator pressure in the EHLC

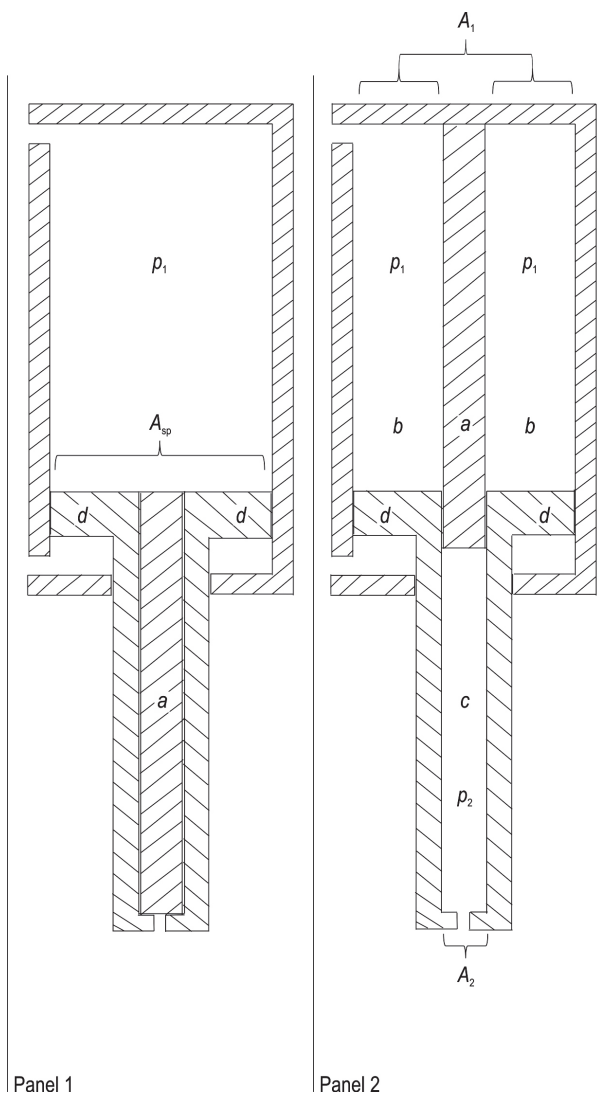


Fig. 2 Panel 1: a standard cylinder. Panel 2: EHLC. In both panels: *a* is a movably arranged secondary piston, *b* is the primary cylinder, *c* is the secondary cylinder and *d* is the primary piston rod. The standard piston area (A_{sp}) is the sum of primary piston area (A_1) and secondary piston area (A_2)

crane are described below based on the manufacturer’s information (cf. WIPO Patent WO/2011/075034).

In the starting position, the secondary piston (*a*) is inside the primary piston rod (*d*), basically forming a standard cylinder (Fig. 2: panel 1). Immediately after the machine has started and during the first boom lift the pressure in the hydraulic circuit of the accumulator will be the same as in the circuit of the standard cylinder. However, during boom lowering the accumulator is further pressurized, the check-valve prevents an out-flow and the standard cylinder turns into primary (*b*) and secondary (*c*) cylinders (Fig. 1 and Fig. 2: panel 2)

due to the pressure differences. As a result, the secondary piston will move in the direction of the lower pressure at any given time, since both of the secondary piston ends have the same area (Figs. 1–2). Hence, the following lift work will be conducted with assistance from the accumulator, provided that the pressure in the accumulator and secondary cylinder (p_2) is higher than in the primary cylinder (p_1). If not, the cylinder will function as a conventional cylinder.

Given that p_2 is higher than p_1 , the product of p_2 and the secondary piston area (A_2) create a secondary cylinder force (F_2) (see details in Figs. 1–2). As the secondary piston thrusts the primary cylinder head continuously with a F_2 that depends on p_2 , it creates an assisting force during boom lifts and a braking force during boom lowering. During boom lifting, the accumulator discharges and decreases p_2 , while a boom lowering charges the accumulator and p_2 increases (Fig. 3). Thus, F_2 decreases the EHLC’s need for external energy input.

Some losses in p_2 are likely to occur due, for instance, to oil leakage from the secondary to the primary cylinder. If p_1 exceeds p_2 the check-valve opens and the accumulator will be charged from the hydraulic circuit system (Fig. 1). Therefore, a higher pressure must be maintained in the accumulator and secondary cylinder than in the primary cylinder to get an assisting force during a boom lift. Occasional pressure spikes in the hydraulic circuit pass the check-valve and load the accumulator, to 25–30 MPa according to the patent. Thus, occasional pressure spikes that always appear in ordinary hydraulic systems (Manring 2005), and hence occasional short-term check-valve openings are essential contributions to the EHLC’s functionality because they maintain a higher p_2 than p_1 during boom lifts, and also provide leakage compensation (Fig. 3). In addition, the secondary cylinder work (W_2) increases with increasing p_2 , thereby decreasing the need for an external energy input. Thus, check-valve openings caused by occasional pressure spikes should not be confounded with »malfunctions« of the EHLC, i.e. regularly open check-valve.

The standard and secondary piston diameters of the studied EHLC cylinder were 124.5 mm and 32.0 mm, respectively (Roger Gustavsson, Thordab AB). This provides a standard piston area (A_{sp}) of 12,174 mm², A_2 of 804 mm², and a primary piston area (A_1) of 11,370 mm² (Fig. 2). The accumulator volume was 1.0 litre (Roger Gustavsson, Thordab AB).

Based on these specifications, the theoretical energy saving is 6.6–6.9%, given that $A_2/A_{sp} \approx 0.066$, and p_2 is in the range of 1.00 to 1.05 times p_1 during the entire boom lift.

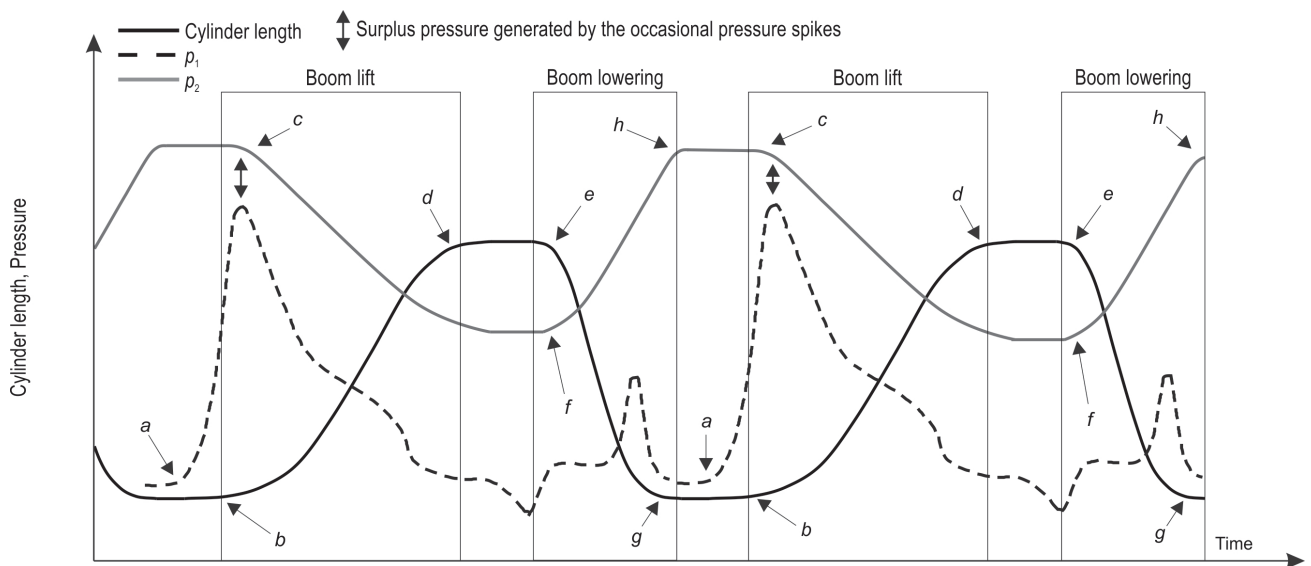


Fig. 3 Schematic visualization of flawless work by EHL. A lift starts by pressurizing the primary cylinder (a), and the boom starts to lift (b) as the primary cylinder pressure (p_1) reaches a required threshold. The secondary cylinder pressure (p_2) decreases (c) until the end of boom lift (d) as the increasing secondary cylinder volume discharges the accumulator (Fig. 1). During the boom lowering (e) p_2 increases (f) until the end of boom lowering (g) as the decreasing secondary cylinder volume recharges the accumulator (h). p_2 creates an assisting lift force, which is inversely proportional to the lift cylinder length. There are risks of p_2 and p_1 intersecting without a pressure surplus from the occasional pressure spikes. However, no occasional pressure spike is shown in the figure, only the pressure surplus

2.2 Test and measure procedure

An EHL was mounted on a standard Gremo 1050 F forwarder (engine power 120 kW, hydraulic working pressure 23.5 MPa) equipped with a Cranab FC80 crane (boom mass 890 kg), a standard rotator (56 kg), and a Cranab CR280 grapple (200 kg). The experiment took place between the 11th and 13th of July 2011 in Umeå, Northern Sweden.

The mid and extending booms were mechanically blocked in positions that gave a constant crane reach of 5.7 m. During the experiment, the boom was lifted and lowered by actuating the lift cylinder directional control valve using a joystick. Three settings for the valve response to the joystick actuations were used, resulting in different directional control valve opening speeds for the same joystick movement and hence different acceleration and boom speeds, designated slow, medium and fast (Table 1). A boom lift with »slow« valve setting was followed by a boom lowering with »slow« valve setting and so on. When actuated, the joystick was pushed to extremity with a fast movement. At end destinations of the boom (lifted and lowered), the joystick was released back to the neutral position for at least 5 seconds to ensure pressure stabilization in the hydraulic circuit.

The boom tip position varied from approximately 1.5–4.5 m above ground during a lift cycle. The operator (a 31 year-old male with no previous experience of work with heavy machinery) tried to keep the lift heights constant throughout the experiment. Initial and final cylinder lengths (Fig. 1) were documented (Table 1), and the possible effects of their variation on the results were reduced statistically (see chapter, »2.3 Statistical analysis«).

By closing and opening certain valves in the hydraulic system, the accumulator could be overridden. When it was overridden, the EHL functioned as a standard hydraulic lift cylinder (standard cylinder) providing a reference cylinder for comparisons of energy use (Fig. 2: panel 1).

Pressure observations (p_1 and p_2) were recorded by Bofors TDS-1 load cells (error ± 500 kPa) (Fig. 1). Both of the load cells used were calibrated with a Barnet Instruments dead weight tester once at the beginning of the experiment. The cylinder length sensor used (error ± 2.3 mm) was based on a 10 turn 1 k Ω potentiometer. Data from load cells and the cylinder length sensor were recorded 10^5 times per second using a DEWE 2520 datalogger with an integrated computer running the DEWEsoft 6.5 program.

Table 1 Initial and final lift cylinder lengths, stroke lengths, lift times and average piston velocities during boom lifts for each combination of lift cylinder model (cylinder model), payload and directional control valve setting (valve setting). Mean values, with standard deviations in parenthesis

Payload kg	Valve setting	Cylinder model	Observations n	Initial cylinder length mm	Final cylinder length mm	Stroke length mm	Lift time s	Piston velocity mm/s
0	Slow	Standard	25	1093 (1)	1213 (3)	121 (3)	4.6 (0.1)	26.1 (0.3)
		EHLC	27	1091 (1)	1211 (3)	120 (3)	4.6 (0.1)	26.1 (0.3)
	Medium	Standard	34	1093 (3)	1223 (5)	130 (4)	1.7 (0.1)	77.4 (2.5)
		EHLC	32	1091 (3)	1219 (8)	128 (8)	1.7 (0.1)	76.7 (2.7)
	Fast	Standard	57	1092 (6)	1223 (5)	131 (5)	1.5 (0.1)	85.5 (2.4)
		EHLC	34	1096 (8)	1224 (9)	128 (12)	1.5 (0.1)	84.1 (3.2)
264	Slow	Standard	21	1113 (1)	1228 (2)	116 (2)	7.7 (0.2)	15.0 (0.2)
		EHLC	22	1113 (1)	1229 (2)	117 (2)	7.9 (0.2)	14.8 (0.2)
	Medium	Standard	20	1111 (3)	1234 (10)	123 (12)	1.8 (0.1)	68.8 (3.0)
		EHLC	27	1114 (5)	1233 (6)	119 (9)	1.8 (0.1)	64.7 (1.5)
	Fast	Standard	26	1037 (12)	1173 (8)	136 (15)	1.9 (0.2)	71.3 (2.6)
		EHLC	27	1093 (9)	1236 (10)	144 (16)	2.0 (0.2)	72.6 (1.9)
513	Slow	Standard	15	1095 (1)	1213 (1)	118 (2)	20.6 (0.6)	5.7 (0.1)
		EHLC	18	1094 (3)	1213 (1)	119 (3)	19.3 (1.0)	6.2 (0.2)
	Medium	Standard	20	1060 (9)	1220 (8)	160 (13)	3.9 (0.4)	41.6 (2.6)
		EHLC	20	1066 (6)	1229 (9)	162 (9)	3.5 (0.2)	46.2 (1.8)
	Fast	Standard	26	1057 (18)	1216 (11)	159 (15)	3.7 (0.8)	44.4 (6.8)
		EHLC	22	1053 (12)	1233 (8)	179 (9)	3.7 (0.3)	49.0 (2.9)

A general model for determining lift work during a given time interval ($W_{t \rightarrow t+1}$) (Eq. 1) was used as a starting point for calculating the work conducted by the two cylinders.

$$W_{t \rightarrow t+1} = 0.5 \times (p_t + p_{t+1}) \times A_{\text{piston}} \times (L_{t+1} - L_t) \quad (1)$$

Where:

- p the cylinder pressure at time t or $t+1$,
- A_{piston} piston area,
- L is the cylinder length at time t or $t+1$.

The work conducted during a given boom lift was calculated by dividing the lift into 10^5 time intervals per second and summing the work for all time intervals.

The standard cylinder's lift work during time interval from t to $t+1$ ($W_{s, t \rightarrow t+1}$) was calculated based on Eq. 1, with p_1 used for pressure (i.e. the pressure observations retrieved from load cell₁, Fig. 1), and A_{sp} as piston area (Fig. 2: panel 1). Calculation of the functioning of EHLC's lift work in a technically perfect state ($W_{\text{EHLC}, t \rightarrow t+1}$) was based on p_1 and A_1 as piston area (Fig. 2: panel 2). However, the EHLC might not work perfectly; a possible condition being when p_1 might exceed p_2 during part of the lift.

In this study, the part of a lift where $p_1 < p_2$ is referred to as the EHLC's »successful lift phase« (Fig. 4: $t_1 \rightarrow t_2$), because during this phase the EHLC is theoretically capable of contributing to the lift with recovery energy. Similarly, the »unsuccessful life phase« refers to the part of the lift after which $p_1 > p_2$ for the first time and the EHLC will not be able to contribute to reductions in energy use.

Lift work calculations for the EHLC's successful lift phase were identical to the calculations for flawless functioning of the EHLC. However, work calculations for EHLC's unsuccessful lift phase varied depending on whether p_1 or p_2 was highest. For time intervals with $p_1 > p_2$ the EHLC was assumed to function as a standard cylinder and work was determined correspondingly ($W_{s, t \rightarrow t+1}$). For time intervals with $p_1 < p_2$ during the unsuccessful lift phase, the EHLC's work was determined as the sums of $W_{\text{EHLC}, t \rightarrow t+1}$ and the secondary cylinder lift work ($W_{2, t \rightarrow t+1}$), given that regaining the EHLC's functionality was a result of accumulator being loaded from the hydraulic circuit during the lift (i.e. consuming energy), and not by using the recovered potential energy. $W_{2, t \rightarrow t+1}$ was calculated according to Eq. 1 with p_2 used as pressure and A_2 as the piston area (Fig. 2: panel 2).

The energy savings for the EHLC's successful lift phase were determined as the secondary cylinder proportion of the EHLC's total work during the successful lift phase, which in turn was determined as the sum of the primary and secondary cylinder work.

2.3 Statistical analysis

The statistical analysis included only boom lifts, as boom lowerings were excluded. Analysis of covariance (ANCOVA) was used to evaluate effects of three fixed factors (lift cylinder model, payload, and valve setting) on two dependent variables: total work per lift and the initial p_1 , retrieved from a load cell₁ (Fig. 1). The factor cylinder model had two levels – EHLC and standard cylinder, while the factor payload had three levels – objects with the mass 0, 264 and 513 kg – held in the boom tip. Finally, the factor valve setting had three levels: slow, medium and fast (Table 1). In total, this resulted in 18 treatments, which were each replicated several times ($15 \leq n \leq 57$, Table 1). The three-factorial model contained all possible interaction effects between factors.

Covariates were used if they significantly contributed to the model, and were considered logical and not risked to be confounded with treatment effects. In this experiment, the continuous variables – initial and final cylinder length, stroke length and lift time – were used as covariates for each boom lift. To avoid a rank

deficiency, the initial and final cylinder lengths were prioritized over the stroke length. The stroke length effect was tested only if the initial or final cylinder length had no effect.

In addition to the dependent variables mentioned above, the EHLC was also evaluated separately to address its functionality (i.e. without comparison between cylinder models). For such analyses all the cylinder model-related terms were removed from the three-way ANCOVA, resulting in a two-way ANCOVA with the fixed two-way interaction effect: payload \times valve setting. The dependent variables analyzed were related to pressure in the primary and secondary cylinders as well as work and time during the successful lift phase.

The order between treatments was randomized, but all replicates within a treatment were conducted sequentially. However, the first 10 boom lifts within each treatment were excluded from the analysis to ensure that the system had stabilized in terms of oil pressure and temperature during the data collection.

A general linear model (GLM) was used to analyze the analysis of variance (ANOVA) and ANCOVA models. During the GLM procedure, pair-wise differences were analyzed with Tukey's simultaneous test of means. The normality of residuals was evaluated by the Anderson-Darling test. Differences in initial cylinder pressures within the EHLC were tested for deviation from

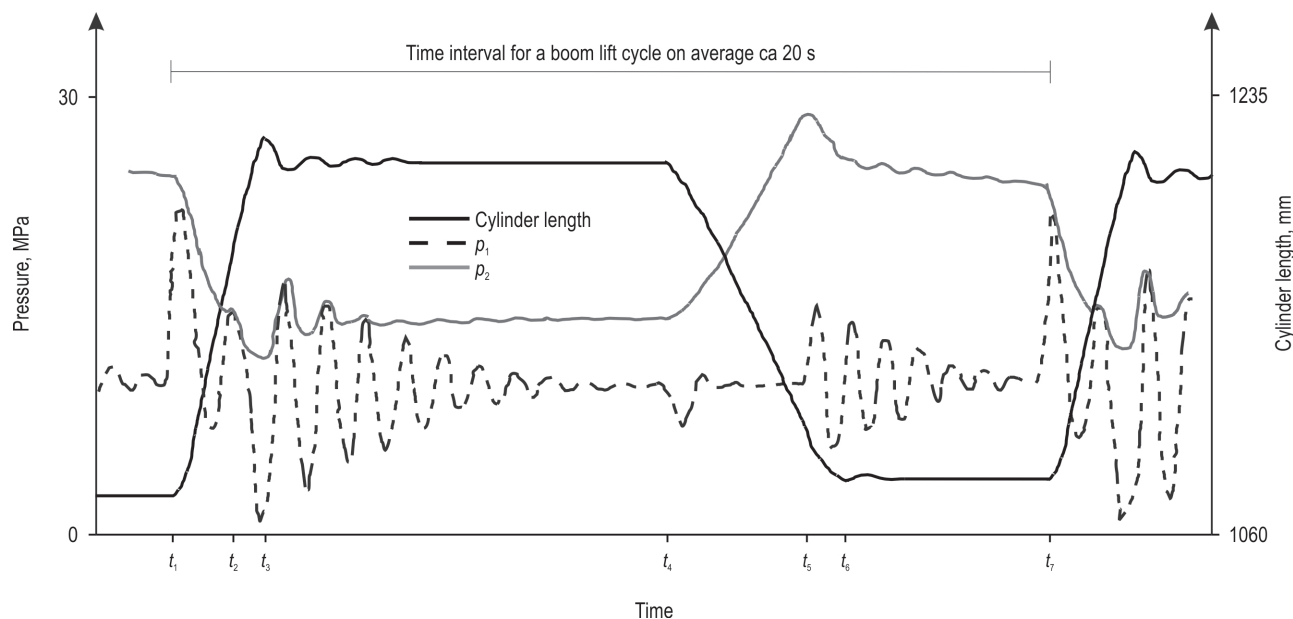


Fig. 4 Example of observed pressures and cylinder lengths as a function of time for one whole EHLC boom lift cycle ($t_1 \rightarrow t_7$) with the valve setting »medium« and payload of 0 kg. The boom lift, as well as the successful lift phase, starts at time t_1 . The successful lift phase ends at t_2 when p_2 exceeds p_1 for the first time. The boom lowering, as well as charging of the accumulator, starts at t_4 . At $t_5 \rightarrow t_6$, p_2 drops rapidly because the pressure relief valve opens and oil flows to the oil reservoir (Fig. 1). At t_7 , a new lift starts

Table 2 Levels of significance (p -values) and explained variance (R^2 adjusted values) obtained from the analysis of variance (ANOVA) of: effects on the dependent variables listed in the first column of the factors cylinder model (a), payload (b) and valve setting (c); their fixed interaction effects ($a \times b$, $a \times c$, $b \times c$ and $a \times b \times c$); and effects of the covariates initial lift cylinder length (d), final lift cylinder length (e), lift time (f) and stroke length (g)

Dependent variables	p-value											Adj. R^2 %	n	
	Factor							Covariate						
	a	b	c	a×b	a×c	b×c	a×b×c	d	e	f	g			
Energy consumption per whole lift	<0.001	<0.001	<0.001	<0.001	<0.001	<0.001	<0.001	<0.001	<0.001	<0.001	<0.001	–	99.7	473
Initial p_1 ¹⁾	<0.001	<0.001	<0.001	<0.001	<0.001	<0.001	<0.001	Excl.	–	–	–	–	83.8	473
Initial p_2 ¹⁾	–	<0.001	<0.001	–	–	<0.001	–	<0.001	–	–	–	–	99.6	229
Difference between initial p_2 and initial p_1	–	<0.001	<0.001	–	–	<0.001	–	Excl.	–	–	–	–	94.4	229
Number of p_2 and p_1 intersections per lift	–	<0.011	<0.001	–	–	<0.001	–	<0.001	Excl.	Excl.	Excl.	–	73.3	229
W_2 ¹⁾ during the successful lift phase	–	<0.001	<0.001	–	–	<0.001	–	Excl.	–	–	–	–	76.6	162
W_1 ¹⁾ during the successful lift phase	–	<0.001	<0.001	–	–	<0.001	–	Excl.	–	–	–	–	78.2	162
Relative energy saving for successful lift phase ²⁾	–	<0.001	0.068	–	–	<0.001	–	Excl.	–	–	–	–	43.7	162
Successful phase stroke length	–	<0.001	<0.001	–	–	<0.001	–	Excl.	–	–	–	–	76.0	162
Successful phase proportion of total stroke length	–	<0.001	<0.001	–	–	<0.001	–	Excl.	0.005	–	Excl.	–	76.5	162
Successful phase lift time	–	<0.001	<0.001	–	–	<0.001	–	0.011	–	–	–	–	88.6	162
Successful phase proportion of the total lift time	–	<0.001	<0.001	–	–	<0.001	–	Excl.	0.002	–	Excl.	–	78.3	162

$n \leq 229$ when only the EHLC treatments were included, and $n = 162$ when only the EHLC treatments with medium and fast directional control valve settings were included

– independent variable was not tested

Excl. covariate term was tested, but excluded from the model because it had no effect ($p > 0.05$) or it decreased adj. R^2 -value

¹⁾ p_1 primary or standard cylinder pressure (depending on cylinder type)

p_2 secondary cylinder pressure (only EHLC)

W_2 secondary cylinder work

W_1 primary cylinder work

²⁾ W_2 during the successful lift phase / (W_2 during the successful lift phase + W_1 during the successful lift phase)

zero by use of a one-sample t -test. The critical level of significance was set to 5%. Minitab 16 (Minitab Ltd.) was used for all analyses.

3. Results

The EHLC's functionality is dependent on the pressure in the secondary cylinder exceeding the pressure in the primary cylinder ($p_1 < p_2$, Figs. 1–2). However, this was found to never occur during a full lift, but only during various intervals of the first part of the boom lift, which varied from repetition to repetition (Fig. 4: $t_1 \rightarrow t_2$). Moreover, p_2 and p_1 were practically identical during the whole lift when using the valve setting »slow« (no data shown).

For valve settings »medium« and »fast«, the EHLC's p_2 level surpassed p_1 in the beginning of a lift (Fig. 4). During this »successful lift phase« (Fig. 4: $t_1 \rightarrow t_2$), the accumulator contributed recovered energy from the preceding lift. During the next boom lowering, the accumulator was loaded as $p_1 < p_2$ (Fig. 4: $t_4 \rightarrow t_5$). When p_2 exceeded approximately 30 MPa, the accumulator stopped charging, indicating that the pressure relief valve was released at that pressure (Fig. 4: $t_5 \rightarrow t_6$).

3.1 Comparison of cylinder models for whole lifts

The three main factors – the cylinder model, payload, and valve setting – significantly affected both of the two dependent variables initial p_1 and the energy

consumption per full lift (three-way ANCOVA, $p < 0.001$, Table 2: rows 1–2). As expected, most of the discrepancies between the two dependent variables were explained by the payload, while the valve setting and cylinder model only had minor effects (data not shown). The initial p_1 was not affected by the only tested covariate – the initial cylinder length (three-way ANCOVA, $p = 0.774$, Table 2: row 2, complete data not shown). On the other hand, the energy consumption per lift was affected by all the recorded covariates: the initial and final cylinder length, and lift time (three-way ANCOVA, $p < 0.001$, Table 2). Overall, the total energy consumption model was improved most by inclusion of the initial cylinder length, followed by the final cylinder length. The lift time had least effect (data not shown).

The EHLC's initial p_1 increased significantly ($p < 0.001$) with increasing payload across all the three valve settings (Table 3). However, the standard cylinder's initial p_1 increased significantly ($p < 0.001$) with increasing payload only across the valve setting »fast«. In addition, there was a lack of statistically significant differences between the treatments when the payload effects were compared over the valve settings »slow« and »fast«. This resulted in significant ($p < 0.001$) two- and three-way interaction effects (Tables 2–3).

As expected, the energy consumption per lift for both cylinder models increased significantly ($p < 0.001$) with increasing payload across all three valve settings (Table 3). The valve settings significantly ($p < 0.05$) affected EHLC's energy consumption with payloads of 0 and 513 kg, but had no significant effect with the 264 kg

Table 3 Pressure and energy consumption for a whole lift with EHLC and standard cylinder for each combination of payload, valve setting and cylinder model. Mean values, with standard deviations in parenthesis

Payload kg	Valve setting	Cylinder model	Initial p_1 ¹⁾ kPa	Initial p_2 kPa	Difference between initial p_2 and initial p_1 ²⁾ , kPa	Number of p_2 and p_1 intersections during the whole lift, n	Energy consumption per whole lift, J	Observations n
0	Slow	Standard	14,182 ^{DE} (304)	–	–	–	18,992 ^{EF} (346)	25
		EHLC	13,068 ^E (98)	12,591 ^G (129)	–479 ^F (81)	47.5 ^A (18.9)	19,330 ^F (410)	27
	Medium	Standard	13,264 ^F (2617)	–	–	–	18,257 ^F (551)	34
		EHLC	10,464 ^F (1670)	24,374 ^{ABC} (465)	13,909 ^A (1912)	5.1 ^C (2.3)	17,669 ^G (1340)	32
	Fast	Standard	13,105 ^E (2698)	–	–	–	18,400 ^F (712)	57
		EHLC	9791 ^F (777)	21,637 ^D (400)	11,767 ^B (762)	11.7 ^{BC} (4.4)	18,488 ^{EF} (1573)	34
264	Slow	Standard	18,328 ^{BC} (524)	–	–	–	28,514 ^C (388)	21
		EHLC	17,588 ^C (208)	16,863 ^F (195)	–1053 ^F (93)	11.8 ^{BC} (3.1)	28,186 ^C (465)	22
	Medium	Standard	21,647 ^A (2163)	–	–	–	27,390 ^C (2423)	20
		EHLC	14,985 ^D (1154)	24,516 ^{AB} (180)	9178 ^C (1208)	23.4 ^B (3.4)	27,185 ^C (1663)	27
	Fast	Standard	15,167 ^D (936)	–	–	–	23,064 ^D (3010)	26
		EHLC	15,262 ^D (576)	24,152 ^C (211)	8870 ^C (506)	14.5 ^{BC} (6.0)	27,172 ^C (3294)	27
513	Slow	Standard	19,859 ^{AB} (145)	–	–	–	36,164 ^{AB} (475)	15
		EHLC	20,043 ^{AB} (115)	19,868 ^E (121)	–227 ^F (33)	18.4 ^B (7.9)	35,642 ^{AB} (841)	18
	Medium	Standard	21,142 ^A (1725)	–	–	–	32,773 ^A (3214)	20
		EHLC	19,928 ^{AB} (2417)	24,655 ^A (282)	5119 ^D (2081)	24.4 ^B (11.0)	32,097 ^B (2107)	20
	Fast	Standard	20,518 ^A (2047)	–	–	–	32,763 ^A (4298)	26
		EHLC	21,662 ^A (1896)	23,974 ^{BC} (244)	2909 ^E (1373)	58.3 ^A (21.9)	32,720 ^A (2269)	22

Within columns, different superscript letters indicate significant differences ($p < 0.05$). Statistical models are described in Table 2

¹⁾ p_1 , pressure recorded in load cell₁ (Fig. 1), i.e. in the primary EHLC cylinder and standard cylinder

p_2 pressure recorded by load cell₂, i.e. in the accumulator circuit and the secondary cylinder

²⁾ Each of the nine EHLC mean values in the column were statistically significantly different from zero (one-sample t-test, $p < 0.001$)

Table 4 EHLC's energy savings (mean values with standard deviations in parenthesis) for the successful lift phase

Payload kg	Valve setting	Cylinder work ¹⁾		Energy saving ²⁾ %	Successful lift phase stroke length mm	Successful lift phase proportion of the total stroke length, %	Successful lift phase lift time s	Successful lift phase proportion of the total lift time, %	Observations <i>n</i>
		W_2 , J	W_1 , J						
0	Medium	1033 ^A (423)	9422 ^A (4191)	9.4 ^A (1.0)	69.6 ^A (31.4)	52.3 ^A (22.0)	0.84 ^A (0.32)	51.9 ^A (17.5)	32
	Fast	34 ^C (15)	396 ^C (192)	8.4 ^B (1.3)	2.0 ^C (0.9)	0.3 ^C (0.11)	0.19 ^B (0.03)	9.3 ^C (2.2)	34
264	Medium	26 ^C (14)	318 ^C (140)	8.2 ^B (0.9)	1.4 ^C (0.6)	2.6 ^C (1.1)	0.25 ^B (0.06)	10.0 ^C (3.4)	27
	Fast	17 ^C (9)	191 ^C (87)	8.5 ^B (0.9)	0.8 ^C (0.3)	3.1 ^C (0.5)	0.17 ^B (0.03)	10.2 ^C (1.5)	27
513	Medium	407 ^B (30)	5326 ^B (365)	7.1 ^C (0.1)	22.1 ^B (1.6)	13.7 ^B (1.3)	0.98 ^A (0.07)	33.7 ^B (1.3)	20
	Fast	340 ^B (29)	4496 ^B (386)	7.0 ^C (0.0)	29.1 ^B (6.6)	17.7 ^B (4.2)	0.98 ^A (0.12)	31.9 ^B (1.7)	22

Within columns, different superscript letters indicate significant differences ($p < 0.05$). Statistical models are described in Table 2

¹⁾ W_2 secondary cylinder work, and W_1 = primary cylinder work

²⁾ W_2 during the successful lift phase / (W_2 during the successful lift phase + W_1 during the successful lift phase) $\times 100$

payload. In contrast, the standard cylinder's lift energy consumption was significantly affected by the valve settings with the 264 kg payload, but not with the 0 and 513 kg payloads. This resulted in significant ($p < 0.001$) two- and three-way interaction effects (Tables 2–3).

3.2 Evaluation of the EHLC's successful and unsuccessful lift phases

In addition to the initial p_1 and energy consumption per full lift, the EHLC's successful lift phase was further analyzed with supplemental dependent variables (Table 2: number of observations in the range of 162–229). The two main factors, payload and valve setting, significantly ($p < 0.001$) affected all the supplemental dependent variables, when data for all three valve settings were included in the analyses (Table 2). The initial p_2 , and number of intersections of p_2 and p_1 per lift, were also significantly affected by the covariate initial cylinder length ($p < 0.001$) (Table 2). The significant ($p < 0.001$) interaction effects showed that the factors effects varied between the compared treatments, and occasionally, the interaction effect was the result of a lack of differences between the compared treatments (Tables 2–3).

With the valve setting »slow«, the initial p_1 was significantly higher than p_2 (Table 3), thus the possibilities of energy recovery were eliminated even before the boom was lifted. Consequently, only data obtained with the valve settings »medium« and »fast« were further analyzed. With these settings, the initial p_2 was substantially higher than the initial p_1 , which enabled energy recovery (Table 3). The payload and valve set-

ting significantly ($p < 0.05$) affected all except one dependent variable in EHLC's successful lift phase (Table 4). The exception, which fell just outside the set level for significance, was that the relative energy savings for EHLC's successful lift phase was not affected by the valve setting ($p = 0.068$) (Table 2). In addition, the valve settings had a significant ($p < 0.05$) effect on the dependent variables with the 0 kg payload, but not with the 264 or 513 kg payloads (Table 4). This resulted in significant two-way interaction effects ($p < 0.001$, Table 2).

The EHLC's successful lift phase corresponded to 9.3–10.2% of the total lift time when lifting a payload of 0 kg with the valve setting »fast«, or when lifting a payload of 264 kg with either »fast« or »medium« valve settings. However, given that the successful lift phase covered only an acceleration phase, the successful lift phase proportion of the total stroke length was only 0.3–3.1%. Thus, W_2 was only 17–34 J (Table 4). The EHLC's successful lift phase constituted 31.9–33.7% of the total lift time when lifting a payload of 513 kg with either valve settings, corresponding to 13.7–17.7% of the total stroke length and W_2 of 340–407 J (Table 4). When using the valve setting »medium« and lifting a payload of 0 kg, the successful lift phase was 51.9% of the total lift time. This corresponded to 52.3% of the total stroke length and resulted in a W_2 of 1033 J (Table 4).

3.3 Energy savings

When evaluating a full boom lift, the EHLC functioned best for payloads of 513 kg and 0 kg with the valve setting »medium«. Under these conditions lifting with the EHLC consumed significantly less energy

than lifting with the standard cylinder (2.1–3.2%, $p < 0.001$) (Table 3). However, when lifting 264 kg with the valve setting »fast« the EHLC did not function flawlessly at all, and consumed significantly more energy for a full lift than the standard cylinder (17.8%, $p < 0.001$) (Table 3).

During the successful lift phase, the EHLC saved approximately 7.0–9.4% of the energy consumed ($p < 0.001$). The relative energy savings were also significantly higher with a lower payload (Table 4).

4. Discussion

4.1 Comparison with previous studies

During the successful lift phase EHLC saved up to 9.4% of the energy consumed. However, on average the successful lift phase accounted for only a small proportion of the whole boom lift, and sometimes during the following unsuccessful lift phase the EHLC even increased energy requirements. In addition, the savings during the successful lifting phase were in the lower end of ranges of savings (ca. 5–65%) reported in previous evaluations of other solutions for reducing energy requirements of mobile lifting devices (e.g. Liang and Virvalo 2001a, Sun and Virvalo 2003, Rydberg 2005, Sun and Virvalo 2005, Virvalo and Sun 2005, Lin et al. 2010, Lin and Wang 2012, Minav et al. 2012, Noréus et al. 2013, Wang et al. 2013). Furthermore, the EHLC did not give any energy recovery at all with the valve setting »slow«, presumably because of internal oil leakage between the primary and secondary cylinders. The boom lifting and lowering times were at least twice as long with the »slow« valve setting than with the settings »medium« or »fast«, and inevitably internal leakage and thus the pressure decrease will be larger if the duration is longer (assuming all other variables remain constant). An example of leakage from the secondary cylinder is shown in Fig. 4, where p_2 is decreasing during the time interval $t_6 \rightarrow t_7$ and the only possible reason for the decrease (after the pressure relief valve has closed) is leakage. The finding that degrees of leakage are correlated with its duration is consistent with previous reports (Wang et al. 2013).

An apparent problem with the tested EHLC is that the accumulator gas volume seems to be too small in relation to the secondary cylinder oil flow volume during the boom liftings and lowerings. This issue can be seen in Fig. 4, where the EHLC's pressure curve decreases sharply as a function of cylinder length ($t_1 \rightarrow t_2$), then increases rapidly until the pressure relief valve opens ($t_4 \rightarrow t_5$). According to Boyle's law, increasing the accumulator gas volume could solve this

problem as it would stabilize p_2 or reduce its peak-to-peak pressure amplitude. The main problem here is not too low maximum p_2 , which is already regulated by the pressure relief valve, as described, but that p_2 decreases too rapidly.

4.2 Strengths and weaknesses of the study

The study was conducted in an experimental setting, with standardized work procedures and high data recording frequencies, facilitating isolation of the effects of cylinder type under various work conditions, and potentially, the identification of useful general procedures for assessing forwarder cranes energy requirements, forces and functional parameters during lift work. Considering the EHLC as an early prototype, we have focussed mainly on work phases where the EHLC made a successful contribution when analyzing the data.

The forwarder used was a standard forwarder equipped with a load sensing system in the hydraulic system, so the hydraulic oil flow and pressure delivered from the hydraulic pump depended on the power required at the moment (see e.g. Scherer et al. 2013). The load sensing system was not overridden during the experiment. Thus, effects from the load sensing system could have been confounded with the tested factor effects, which may have affected the ANOVA results. However, to our knowledge any effects of the load sensing system should be negligible for the technical evaluation of the EHLC.

When determining the energy consumption of the EHLC for a whole boom lift, it was assumed that the secondary piston thrusts either the primary or the secondary cylinder head. However, this assumption was not entirely valid because the secondary piston could also move in the direction of lower pressure. Nevertheless, this error sources only applies when the successful lift phase is complete and should not therefore impact the analysis of the EHLC's successful lift phase.

4.3 Improvements and future studies

The results of this study indicate several possible improvements for the EHLC. First, the internal leakage problem has to be resolved. Second, in addition to enlarging the accumulator, its pressurization should be optimized so that the EHLC works flawlessly throughout the whole lifting phase. This suggestion is prompted by the observation that the generated pressure spikes do not appear to be sufficient to maintain pressurizations long enough to reduce energy needs within the current accumulator system. Consequently, it is essential to redesign the system to provide a notably higher initial accumulator pressure. During the

»unsuccessful lift phase« (after the first p_2 and p_1 intersection), the hydraulic pump pressurized the primary cylinder as well as the accumulator and secondary cylinder (Fig. 4: $t_2 \rightarrow t_3$). However, flawless functioning of the EHLC depends on the actual boom length and load, since if p_2 is too high and the boom too short energy will actually be needed to lower the boom. This reflects a general trade-off for all weight-balancing systems, and requires optimization according to the given crane dimensions and the loads lifted and lowered. Third, A_2 should be enlarged. Overall, the experiment should be replicated with a larger accumulator, a refined accumulator pressurization system, and possibly a larger A_2 .

4.4 Potential energy savings for all forwarder work

The current EHLC can save up to 3.2% of energy for a full boom lift under optimal conditions. However, if its technical weaknesses can be resolved, the savings should be at least 6.6%, as explained in the Introduction. Moreover, if some additional technical improvements are made, additional energy savings could be achieved. For instance, if the accumulator pressurization system is redesigned to ensure that p_2 is consistently at least 20% higher than p_1 , the energy savings for a whole boom lift should theoretically be at least 7.9%. In addition, increasing A_2 would increase the energy recycling potential of the EHLC. For example, increasing secondary piston diameter by 20% would increase the possible energy savings from 6.6% to 9.5%. Therefore, the EHLC's energy recycling potential could be enhanced by increasing both secondary piston diameter and p_2 by 20%. With these two improvements, the EHLC could theoretically recycle at least 11.4% of potential energy, ignoring possible leakage.

However, energy-efficient lifting devices have not yet acquired any of the market shares for forwarder cranes. So far, the EHLC's market consists primarily of hydraulic lift devices, where the full engine power is used for lift work, which is not the case for forwarders. Currently minor energy savings can be gained from energy-efficient lift cylinders for a standard forwarder, as driving the machine requires substantially more force and power than the boom lifting (cf. Löfgren 1999, Edlund et al. 2013, Table 3). Thus, the research priority should be placed on decreasing power requirements during the driving phase, i.e. improving powertrain efficiency, as in a few recent studies (e.g. Edlund et al. 2013, Swedish Energy Agency 2014).

Overall, combining energy-efficient lifting devices with a hybrid powertrain could be interesting for fu-

ture studies. With an energy-efficient lifting device reducing the energy needed for the crane work, more power from the combustion engine could be directed to loading the battery throughout the crane work. This electric energy, stored during the crane work, would then be available for use during driving, when the power input requirement is highest. Thus, an energy-efficient lift device could improve the battery loading efficiency of a hybrid system during crane work. Such technology could enable the use of less powerful, i.e. less fuel consuming, combustion engines in forwarders.

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