Electric or Hydraulic Energy Recovery Systems in a Reach Truck– A Comparison

Tatiana Minav1,* - Henri Hänninen1 - Antti Sinkkonen1 - Lasse Laurila2 - Juha Pyrhönen2
1 Department of Engineering Design and Production
Aalto University, School of Engineering, Espoo, Finland
2 LUT Energy, Lappeenranta University of Technology,
Lappeenranta, Finland

In this paper, electric and hydraulic regeneration methods of recovering potential energy from an electro-hydraulic forklift truck are studied. Two similar forklift setups equipped with either electric or direct hydraulic energy storage are compared. In the first setup, the forklift lifting system is controlled directly with an electric servo motor drive. The servo motor is driving a hydraulic pump capable of operating also as a hydraulic motor during lowering motion. In the second setup, the hydraulically operated forklift is equipped with an energy recovery system consisting of pressure accumulators for storing energy and a hydraulic digital valve package for precise leakage free flow control. The paper describes the arrangements of the experimental setups, and the results for the proposed systems are compared from the energy efficiency point of view. Energy-saving ratios for electric and hydraulic test systems were calculated for different fork velocities and payloads.

Keywords: Digital Flow Control Unit, electric energy recovery, energy storage, forklift, hydraulic energy recovery, hydraulics, lead-acid battery, Induction Motor (IM), hydraulic accumulator, Permanent Magnet Synchronous Machine (PMSM), Reach truck, supercapacitor.

0 INTRODUCTION

Globally, energy efficiency and energy saving have become important practical research topics in Non-road mobile machinery [1, 2]. In [3, 4] energy saving lifting hydraulic systems and control techniques [5] have been already suggested. However, energy saving is still, in particular, very important in non–road mobile machine applications, for instance, in excavators [6, 7] and especially those operated purely by accumulator-stored electric energy [8, 9]. In order to reduce the energy consumption of a machine, either the efficiencies of the components are to be improved or energy, that is otherwise lost in the process, has to be utilized by regeneration. When considering the latter, this is accomplished in many cases most advantageously by re-using the kinetic or potential energy of the machine or its subsystem [10, 11]. Depending on the system and process in question, the utilization energy recovery can lead to significantly lower overall energy consumption and with mobile machines to longer operating times. [12, 13]

When considering regeneration of energy, the work cycles in which forklifts often operate includes bidirectional material or payload transfers, which enables opportunity for efficient recovery of potential energy. In this study reach trucks (sub type of forklifts) are modified to allow energy recovery from the payload of the masts lifting/lowering function. In the case of the other functions of the machine, there is no potential energy to be recovered, and kinetic energy levels are too low for any feasible recovery system.

There are several base technologies on which to build a recovery system: thermal, mechanical (i.e. fly-wheel or counter weight based recovery systems), electric or hydraulic. This study focuses on the last two types. The well-established method to recover energy in mobile working machines is an electric recovery system. This system type usually consists of an electric motor/generator, an inverter, possibly a DC/DC converter, a battery, and in some cases also an electric double-layer capacitor (EDLC) [14]. The advantages of this kind of a system are control flexibility, compactness, efficient control, and fairly high energy efficiency [15].

When concerning the mast operations of a hydraulic reach truck, another well-justified option is to use a direct or indirect hydraulic recovery system. In the indirect hydraulic storage system consisting of a hydraulic motor-pump, a
controllable hydraulic pump-motor, and a hydraulic accumulator, the flexibility of control is as good as in the case of the electric recovery. This system first converts the hydraulic energy into mechanical and then back to hydraulic, requiring, therefore, as many conversions as the electric storage system. In this paper, however, the direct hydraulic recovery system is compared with the electric recovery system. The direct hydraulic recovery system removes the need for energy conversions from the hydraulic to electric form in the recovering phase and vice versa in the regenerating phase.

The direct application of the hydraulic accumulator, however, contains more limiting factors than the indirect recovery system [16, 17]. The utilization of this type of system requires in the recovery phase two flow control edges in order to maintain controllability of mast velocity. This is achieved by utilization of a digital valve package (DFCU).

This study compares electric and direct hydraulic recovery systems with each other in terms of energy efficiency. The operational characteristics of both systems are also analyzed.

1 TEST SETUPS

This section describes the studied system setups, description of energy evaluation and work cycle. As two different setups were used, as electric recovery setup was located in Lappeenranta University of Technology, and hydraulic recovery setup – in Aalto University.

1.1 Electric recovery system setup

The original non-regenerative AC electric drive and the hydraulic system of the Humanic HS-16F5400 forklift were replaced with the schematics shown in Fig. 1. The electric motor servo drive directly controls the fixed displacement hydraulic pump speed and thereby the position of the hydraulic cylinder piston instead of a traditional proportional valve. The two-way normally closed poppet valve is used as a safety valve, which prevents the load from dropping in the case of a failure. For lifting, the hydraulic pump produces a flow depending on the rotational speed of the servomotor. During lowering a mass, the potential energy forces the hydraulic machine to rotate as a motor, and the electric machine acts as a frequency-converter-controlled generator [18]. The converter controls the generator torque and actively rectifies the generated electric energy to the DC link. Because of the relatively short lowering period (around 10 s), recharging of conventional lead acid batteries is considered inefficient [19]. For energy measuring purposes, a brake resistor was used as the “energy storage”. At the moment, super capacitors seem to be the most suitable solution for fast recharging. The measured super capacitor charge-discharge cycle efficiency of 99 % [4] will be used to estimate the cycle efficiency of the future system equipped also with electrical energy storage. In [4], the measured voltage and current signals of the forklift electric recovery setup were used for super capacitor efficiency measurements. In the forklift electrical energy recovery test setup, a control program was made to control both the electrical and hydraulic parts of the forklift system [20].

Fig. 1. Electric and hydraulic circuits of the main lift function with energy regeneration from potential energy. The experimental system consists of a) single-acting cylinder (I - free lift zone, II - second cylinder zone), two-way normally closed poppet valve, pressure relief valve, hydraulic pump/motor, oil tank, permanent magnet synchronous motor/generator, phase voltage and phase current probes, frequency converter and brake resistor $R_{brake}$, DC voltage and DC current probes.

The instrumentation of the system covers measurement devices for pressures, rotational speed, torque, load position, phase voltages and...
currents, and DC voltage and DC current. The energy consumption in this paper was calculated from the measured current in DC link. Measurements were carried out utilizing dSpace-measurement software. The Converter software was used to measure the rotating speed of the PMSM and to estimate the motor torque. Two S-10 pressure sensors manufactured by WIKA were installed to measure the pressures at pump outlet and between the 2/2-valve and cylinders. Yokogawa PZ4000 Power analysers with a sampling time of 10 μs were used to measure the phase voltages and currents. The speed and height of the fork were measured by a wire-actuated encoder SGW/SGI by SIKO. A HITEC Zero-Flux B 2000 current sensors were used. The accuracy of the sensors can be considered acceptable for these test purposes.

1.2 Hydraulic recovery system setup

The hydraulic recovery setup is based on the fairly similar truck model Humanic HX-16. The simplified hydraulic system of the forklift is shown in Fig. 2. The main components in the studied energy recovery system are the pressure accumulators and the digital flow control unit (DFCU). The DFCU consists of two individually adjustable control edges containing five poppet-type on/off-valves each, all paired with differently sized orifices. The individual adjustability of the control edges is needed for the dynamic division of volume flows between the accumulator package and the tank.

A hydraulic accumulator is a device that stores pressurized hydraulic fluid with an internal nitrogen gas volume enabling the energy storing. The accumulators, manufactured by Hydroll with nominal size of 4 liters, used in this study are of piston-type, which consist of oil and gas chambers separated from each other with a piston. The pre-load pressure level of the gas chamber determines the maximum energy content of the accumulator and affects the efficiency of the recovery. Thus, for efficient operation the pressure level must be adjustable. The altering of pre-load pressure to a higher value between tests, when needed, was done utilizing an external gas container.

For operating the DFCU, a cost-function-based controller was built to determine which valves of the DFCU (both control edges) are to be opened and which closed in order to simultaneously perform the charging of the accumulators and provide the required lowering velocity. The controller calculates the flow through the control edges using the data from the pressure transducers. For these calculations, the equation for turbulent flow through an orifice given by

\[ q_v = C_q A \sqrt{\frac{2 \Delta p}{\rho}} \]  

(1)

is modified to

\[ q_v = K \cdot \Delta p^z. \]  

(2)

The corresponding values of constants \( K \) and \( z \) are individually identified for each passage in the DFCU and adopted to the controller. The identification was done by manually finding the values for the said constants in the equation 2 to match the measured pressure difference – flow curve of each passage. This approach has been found to be a viable method for taking into account both the turbulent losses in the orifice and in the valve, in addition to partly laminar losses in the flow paths of each passage [21].

The recovered energy is utilized in the following lift phase by directing it to the pump inlet, thus reducing the pressure difference over the pump and thereby decreasing the power need of the electric motor driving the pump.

The instrumentation of the test bench covers measurement devices for pressures, flows, temperatures, rotational speed, torque, load forces, load position, battery voltage, and current. In this study, however, the flow sensors were bypassed to achieve the full recovery potential of the system. The current transducer used, was LEM DH 500 B420L B and the load position was measured with Waycon SX120-6000-420A-SA draw wire sensor. The energy consumption reduction results given in this paper were calculated from the measured battery voltage and current drawn from it. Measurements were carried out using Matlab/Simulink xPC Target software, which also included the controller for the DFCU.
1.3 Description of performed tests

The experimental setups were tested with payloads of 0, 500, and 1000 kg at different motor speeds. The velocities of the forks in both cases were set to 0.2, 0.3, and 0.4 m/s. The travel distance had to be limited to 1.6 m due to the maximum measurement time of 10 seconds with sampling time of 10 µs of the Yokogawa PZ4000 Power analyzer in electric recovery setup. The measurements were made in the free lift zone using the first cylinders of the telescopes, resulting in a low tare in this case. This was done in order to attain better correspondence of the pressure levels of the two systems, since the moving structural masses of the masts are fairly similar in free lift zone, but differ greatly in second cylinder zone.

For electric recovery setup lifting and lowering motions were measured separately due to limitations of used Power analyzer. For hydraulic recovery setup, measurements for each measurement point were performed in a single cycle consisting of continuous sequence of lifting, lowering and lifting phases. First lift phase in the cycle is executed without assistance from the accumulators and the latter with the assistance of energy recovered during lowering phase.

1.4 Detailed information on test platforms

Details on the main components of both test platforms are presented in Table 1.

Table 1. Platform main differences

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Test setup</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Electric recovery</td>
</tr>
<tr>
<td>Theoretical volumetric displacement of the pump, m³/rev</td>
<td>13.3·10⁻⁶, manufactured by Erker</td>
</tr>
<tr>
<td>Motor</td>
<td>10 kW CFM112M PMSM manufactured by Sew-eurodrives</td>
</tr>
<tr>
<td>Converter</td>
<td>ACSM1-04x×046A-4 by ABB</td>
</tr>
<tr>
<td>Piston cross-sectional area of the free lift cylinder, m²</td>
<td>0.0026</td>
</tr>
<tr>
<td>Maximum stroke of free lift cylinder, m</td>
<td>0.88</td>
</tr>
</tbody>
</table>

2 EVALUATION OF ENERGY UTILIZATION

Before investigating any energy-saving system, it is necessary to discuss how to evaluate the utilization of energy in an electro-hydraulic forklift. This section introduces the definitions used for the evaluation of energy utilization in the test setups:

Efficiency

Efficiency as a function of time \( \eta(t) \) is normally defined as a ratio between the output \( P_{out} \) and input \( P_{in} \) powers

\[
\eta(t) = \frac{P_{out}(t)}{P_{in}(t)}.
\]
Even though the efficiency $\eta(t)$ is a function of time, it is normally measured trying to keep $P_{\text{out}}$ and $P_{\text{in}}$ as constant as possible in a static situation to be able to obtain for example the rated efficiency of a motor without time dependence. Measuring of the rated efficiency of an electric motor normally takes several hours in order to reach the thermal equilibrium of the machine before defining the efficiency. In the case of a limited linear movement, however, it is very difficult to apply this definition of efficiency as there is no steady state in the operation. Even when abandoning the need to reach thermal equilibrium there is only a few seconds of constant speed operation at some “constant” efficiency, and thus, measuring of the efficiency becomes very difficult. In the case of a forklift, we are actually not very interested in the instantaneous efficiency of the system but the ratio of the total output and input energy. Therefore, the energy efficiency is defined.

**Energy efficiency**

The energy efficiency $\eta_{\text{energy}}(t)$ for a time interval $[t_1, t_2]$ is defined as

$$\eta_{\text{cyc}}(t) = \frac{\int_{t_1}^{t_2} P_{\text{out}}(t) dt}{\int_{t_1}^{t_2} P_{\text{in}}(t) dt} = \frac{E_{\text{out}}}{E_{\text{in}}}, \quad (4)$$

where $E_{\text{out}}$ is the total output energy and $E_{\text{in}}$ is the total input energy of the system during the time interval starting at $t_1$ and ending at $t_2$. Energy efficiency should normally be regarded as a comprehensive term taking the whole life cycle of the system into account. Therefore, the cycle efficiency is defined for each test setup. Calculation of energy efficiency is described in detail in [18] for the electric energy recovery setup and in [16] for the hydraulic recovery setup.

**Energy-saving ratio**

For comparison of the different test setup efficiencies, the energy-saving ratio $\Gamma_s$ is defined as:

$$\Gamma_s = \frac{E_{\text{old}} - E_{\text{new}}}{E_{\text{old}}}, \quad (5)$$

where $E_{\text{old}}$ is the energy consumption of the forklift without energy recovery and $E_{\text{new}}$ is the energy consumption of the forklift with the energy recovery. This ratio $\Gamma_s$ describes how much energy can be saved when the energy recovery is used.

The energy consumption $E_{\text{old}}$ of the electric drive forklift without energy recovery and $E_{\text{new}}$, energy consumption of the forklift with the energy recovery for an electric recovery test setup is defined as

$$E_{\text{cycle},\text{old}} = E_{\text{mot}}/(\eta_{\text{SC}} \cdot \eta_{\text{inv}}), \quad (6)$$

$$E_{\text{cycle},\text{new}} = E_{\text{old}} - E_{\text{brake}}/\eta_{\text{SC}}, \quad (7)$$

where $E_{\text{mot}}$ is the input energy from the electric motor, $\eta_{\text{inv}}$ is the inverter efficiency, $E_{\text{brake}}$ is the recovered energy, and $\eta_{\text{SC}}$ is the discharge efficiency of the supercapacitor, assuming that the charge efficiency is equal to the discharge efficiency. Therefore, the energy-saving ratio $\Gamma_s$ can be defined for an electric recovery test setup as:

$$\Gamma_s = \frac{E_{\text{cycle},\text{old}}}{E_{\text{cycle},\text{new}}} = \frac{E_{\text{brake}} \cdot \eta_{\text{SC}}}{E_{\text{mot}} \cdot \eta_{\text{inv}} \cdot \eta_{\text{CDSC}}}. \quad (8)$$

The $\eta_{\text{CDSC}}$ is the charge-discharge efficiency of the supercapacitor [4]. The inverter efficiency is assumed to be constant and equal to 95 %. The calculation of $E_{\text{brake}}$ and $E_{\text{motor}}$ is described in detail in [19] for the electric energy recovery setup:

$$E_{\text{brake}} = \int_{t_1}^{t_2} (i_{\text{brake}} \cdot R_{\text{brake}}) dt \quad (9)$$

$$E_{\text{motor}} = \int_{t_1}^{t_2} (i_a u_a + i_b u_b + i_c u_c) dt \quad (10)$$

For the hydraulic recovery test setup, $\Gamma_s$ is defined as

$$\Gamma_s = \frac{E_{\text{unassisted}} - E_{\text{assisted}}}{E_{\text{unassisted}}}, \quad (11)$$

where $E_{\text{assisted}}$ is the energy consumption (calculated from the measured electric power drawn from the battery pack) with the hydraulic assistance on and $E_{\text{unassisted}}$ is the energy consumption with the assistance off. The energy consumptions for both the assisted and unassisted cases are defined as
\[ E = \int_{t_0}^{t_1} P \, dt = \int_{t_0}^{t_1} (U \cdot I) \, dt, \quad (12) \]

where \( U \) and \( I \) is voltage and current, respectively.

Since the energies are calculated from discrete measurements, (12) is discretized to

\[ \Delta E = \sum P \cdot \Delta t = \sum U \cdot I \cdot \Delta t, \quad (13) \]

where \( \Delta t \) is the sampling interval.

3 RESULTS

This section reports the results obtained from the measurements described in the previous section. Tables 2 and 3 show the energy saving ratios for different fork’s velocities and payloads for the electric and hydraulic recovery test setups, respectively. Fig. 3 below represents a graphical comparison of the energy saving ratios of the electric and hydraulic setup (Tables 2 and 3 (single pre-load pressure results)).

Table 2. Energy saving ratios for different speeds and payloads for the electric recovery setup

<table>
<thead>
<tr>
<th>Fork’s velocity, [m/s]</th>
<th>Load, [kg]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>500</td>
</tr>
<tr>
<td>0.4</td>
<td>0 % *)</td>
</tr>
<tr>
<td>0.3</td>
<td>0 % *)</td>
</tr>
<tr>
<td>0.2</td>
<td>0 % **)</td>
</tr>
</tbody>
</table>

*) Indicates motoring mode  
**) Indicates generating mode; because of its small value, the energy saving ratio was rounded to zero.

In Fig. 3, the hydraulic accumulator was optimized for the 500 kg payload and 1.6 m height. The direct hydraulic energy is storing results in a good energy saving ratio in such a case. However, when the same settings are used for the 1000 kg mass, the electric energy storage system outperforms the hydraulic by 5 to 14 % with corresponding velocities of 0.4 to 0.2 m/s (compare Tables 2 and 3).

Table 3. Energy saving ratios for different speeds and payloads for the hydraulic recovery setup

<table>
<thead>
<tr>
<th>Load, [kg]</th>
<th>500</th>
<th>1000</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.4</td>
<td>0 %</td>
<td>26 %</td>
</tr>
<tr>
<td>0.3</td>
<td>0 %</td>
<td>30 %</td>
</tr>
<tr>
<td>0.2</td>
<td>0 %</td>
<td>31 %</td>
</tr>
</tbody>
</table>

Single pre-load pressure (optimized for 500 kg)

<table>
<thead>
<tr>
<th>0.4</th>
<th>0 %</th>
<th>26 %</th>
<th>20 %</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.3</td>
<td>0 %</td>
<td>30 %</td>
<td>20 %</td>
</tr>
<tr>
<td>0.2</td>
<td>0 %</td>
<td>31 %</td>
<td>22 %</td>
</tr>
</tbody>
</table>

Fig. 3. Comparison of the energy-saving ratios in % for the electric and hydraulic tests systems, for hydraulic; results of single pre-load pressure setup shown, where \( V \) is fork’s velocity.
In Table 3, the optimized pre-load pressure for the hydraulic accumulator gave good results for the 1000 kg case for the hydraulic storage system. However, if this setting is used, there will be no recovery in the case of the 500 kg mass, which shows the vulnerability of the direct hydraulic storage system in the case of varying loads and heights.

4 DISCUSSION

In this study, the target was to produce similar operation conditions for these two slightly different trucks. Considering the restraints and similarities of the two systems, the simplest solution was to run tests in the free lift zone, and in addition limit the travel distance to 1.6 m. The relatively short travel distance favored the hydraulic setup by few % - units in terms of saving ratios, since the hydraulic accumulator capacity in these test is limited to 16 liters. The selection of free lift zone instead of the second cylinder zone affected both of the systems by diminishing the achievable saving rations.

4.1 The results of the electric energy storage system

The results shown in Table 2 for the energy saving ratio of the electric recovery test setup seem low. The energy-saving ratio is slightly increasing when increasing the load and decreasing the speed. With the 0 kg load there was no recovery observed, and during lowering the electric machine was even working in the motoring mode instead of generating. The maximum energy saving ratio in the free lift zone is 36 %. It is significantly less compared with the earlier measurements achieved when operating in the second lifting zone of the forklift where the tare of the system is high and the relative hydromechanical losses of system is lower than in this case [20]. According to [20], the maximum energy-saving ratio reached with this same Humanic forklift was 53 % when operating in the second lifting zone of the telescope with a 920 kg load. The mass of the moving parts of the mast is larger, and thereby enables the electric drive components to operate closer to their nominal values. There is significantly more potential energy to recover in the second cylinder zone than in the free lift zone.

4.2 The results of the hydraulic storage system

The energy saving ratios of the hydraulic recovery circuit were measured to range from 0 to 45 %. With no load, the system pressure levels remain too low for energy recovery similarly as in the previous case. This is due to the fact that the total flow losses of the system are in the range of cylinder pressure. By introducing a load, the recovery system becomes effective. Using load-optimized pre-load pressures in the gas chambers of the accumulators, the energy saving ratios were measured to range from 26 to 31 % and from 41 to 45 % for loads of 500 kg and 1000 kg, respectively. When concerning the applicability of such system with load optimized pre-load pressure, it is evident that the loads should remain relatively constant for sufficient durations. Such cases are numerous, for example warehouses and material handling tasks in industry.

In usage situations in mixed goods (variable loads) warehouses, it would not be advisable to alter the pre-load pressure between each lifting/lowering phase (because of the additional energy consumption in form of pressurized gas), and, therefore, in most operating cases the pre-load pressure setting would not be optimal. The measurements indicate that when using a single pre-load pressure, optimized for a 500 kg load, the saving ratios with a 1000 kg load would drop significantly (to approximately 20 %) for the whole velocity range when comparing with the ones with the optimal pre-load pressure. The effect of pre-load pressure setting to the effectiveness of the energy recovery system is analyzed with greater detail in [22]. In order to devise a hydraulic recovery system that performs better in mixed load situations the research group in Aalto University will design and construct an alternative recovery circuit based on a hydraulic transformer. Even though this new system is expected to have a lower peak efficiency (than optimized direct system), simulations [13] suggests that it has better overall efficiency.

4.3 Differences in the two previous systems

It was observed that in the free lifting zone, the hydraulic recovery setup with its optimized pressure settings showed better results compared with the electric recovery setup. This fact of less efficient behavior of the electric setup can be partly explained by the short travel
distance (1.6 m), short time (max 5 s) during which generator can recover the electric energy from the potential energy in the electric recovery setup and less tare, which does not enable the electric drive components to operate closer to their nominal and efficient values.

4.4 Operational characteristics

The hydraulic recovery system was designed to leave the operational characteristics unchanged, and this was also achieved. The system can be used adopting one of two different operation strategies in terms of pre-load pressure optimization. Firstly, one could optimize the pressure level permanently for one load. This would be easier from the operator point of view, but the efficiency would not be optimal in most cases. The second strategy can be implemented if the parameters (load and height) for lifting and lowering are known in advance, they remain relatively constant, and their quantity is sufficient. In this strategy, the pre-load pressure is optimized to match the known upcoming cycle. This would allow the recovery system to operate at optimal efficiency, but as a drawback it would add an additional work phase consuming energy.

The electric recovery system showed very good controllability of the hydraulics side. There are no limitations in the electric recovery setup at least if the electric energy storage is selected large enough to receive the largest possible amount of recovered energy during a single lowering action. With increasing electric machine torque (proportional to the sum of the payload and tare), the system efficiency increases because the electric drive components operate closer to their nominal and most efficient values [23, 24].

4.5 Other observations

To modify a conventional forklift to recover potential energy, the following actions are required:

For electric recovery: the control valve has to be replaced with a two-way normally closed valve; the traditional single-acting hydraulic pump has to be replaced with a hydraulic machine working in both directions; an energy storage, e.g. a supercapacitor bank, has to be added for storing the recovered energy, and the control software of the electric motor has to be updated. Lead-acid batteries can also be used, but supercapacitors have a higher charging efficiency.

Current electric recovery setup is operated with a high voltage up to 900 V. It is considered dangerous voltage level for mobile working machines. In future detail comparison with a low voltage 48 V safe setup will be studied. However, we anticipate that it might have similar results from energy-saving point of view as the presented electric energy recovery system evaluated in this paper.

For direct hydraulic recovery: the hydraulic circuit has to be enhanced with an additional (leakage free) flow control unit and regeneration valve; a hydraulic accumulator(s) has to be added, pump has to be altered to type allowing pressurization of inlet, and the software has to be updated to control the directions of oil flow.

An indirect hydraulic energy recovery system consisting of two controllable hydraulic machines and a hydraulic accumulator could also be implemented, but its behavior was not studied here. However, we anticipate that it might have similar capabilities as the electric energy control system evaluated in this paper.

5 CONCLUSIONS

The presented work concentrated on analysing the opportunities of using energy regeneration in electro-hydraulic forklift systems. It was shown by measurements that energy recovery from potential energy is possible in both hydraulic and electric energy storage applications. According to the results, the maximum energy-saving ratio for the free lift zone with optimized hydraulic accumulator parameters was 45 % with the direct hydraulic recovery setup. In practice, however, the direct system suffers from the need to control the pre-load pressure of the hydraulic accumulator or the requirement to select a fixed value for the pre-load pressure, and as a result, significantly lower average values may be obtained. In this test, the best energy-saving ratio of the electric recovery setup was 36 %. This result is disappointingly low compared with the previous results, obtained when operating in the second cylinder zone of the same truck [19]. It can be concluded that the test arrangement favoured the direct hydraulic recovery system, but it still shows that the electric drive system has
numerous advantages. It does not require any pre-load settings and tunings of the energy storage for a specific load or lifting height. Therefore, it makes the direct hydraulic recovery approach impractical in cases where the lifting and lowering range and the mass vary. In such cases, an electric or indirect hydraulic energy recovery system should be considered instead.

ACKNOWLEDGEMENTS

Hydraulic recovery studies: This study is related to the MIDE/HybLab project, funded by Aalto University. The cooperation of Jyri Juhala, M.Sc. at the Department of Engineering Design and Production of the School of Engineering of Aalto University is highly appreciated.

Electric recovery studies: The research was enabled by the financial support of Tekes, the Finnish Funding Agency for Technology and Innovation and FIMA (Forum for Intelligent machines), the European Union, the European Regional Development Fund, and the Regional Council of South Karelia. The research was carried out at the Institute of Energy Technology, at the Department of Electrical Engineering at Lappeenranta University of Technology, Lappeenranta.

REFERENCES


