# Hydraulic Energy Recovery System Utilizing a Thermally Regenerative Hydraulic Accumulator Implemented to a Reach Truck

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## Abstract

The implementation of an energy recovery system for retreiving otherways wasted energy is an effective method for reducing the overall energy consumption of a mobile machine. In a fork lift, there are two subsystems that can be effectively modified for recovering energy. These are the driveline and the lift/lower function of the mast. This study focuses on the latter by studying a recovery system whose main component is a hydraulic transformer consisting of a hydraulic motor, a variable displacement pump and an induction motor. Since the flow rate/pressure - ratio can be modified, the utilization of the hydraulic transformer enables downsizing of the accumulator volume. However, the decrease of the gas volume leads to an increase in the compression ratio of the accumulator, which in terms leads to higher gas temperatures after charging and consequently to higher thermal losses during holding phase. In order to reduce these losses, a thermally regenerative unit was implemented to the gas volume of an accumulator to reduce the temperature build up during charging. In this study, the effect of improving the thermal characteristics of the accumulator to the efficiency of the whole energy recovery system is investigated by means of measurements.

KEYWORDS: Reach truck, energy recovery, thermally regenerative hydraulic accumulator

## 1. Introduction

There are multiple technologies available for on to which a recovery system can be based on. Majority of the commercially available recovery systems are electric recovery systems found in hybrid and electric vehicles. Other commercially available recovery system types are rarer, however some examples can be found, such as Bosch hydraulic regenerative braking system for trucks /1/.

In previous studies related to this project a hydraulic recovery system has been proven to be an effective solution for harvesting the potential energy of the load while in lowering motion /2/. However, in these tests the used accumulator volume was relatively high resulting in low compression ratios in the accumulators when recovering the potential energy of the load. The low compression ratio in terms retains temperature increase in compression phase and thus reduces heat flux exiting the accumulator during subsequant holding phase. When downscaling the accumulator volume, the thermal efficiency of the accumulator becomes a more influencial factor in the overall efficiency of the recovery system.

The most widely used hydraulic accumulator type is the pressure accumulator with pre-charged nitrogen in the gas volume. Pressure accumulators can be divided into sub-categories, labelled by the element dividing the gas and liquid portions. These are piston-, diaphragm-, bladder- and bellow accumulators. These sub-types differ from each other by the maximum allowable compression ratio and by the response speed to changes in the liquid side pressure. The diaphragm- and bellow-type accumulators have the best response speeds and are therefore best suited for pulsation damping applications. The piston-type accumulator has a lower response speed but allows higher compression ratios which makes it more size effective for energy recovering purposes. In addition, the piston type construction allows the modifications for implementation of a linear position sensor to measure the piston position.

In majority of work cycles, thermal losses are the main contributor to the efficiency of pressure accumulators. The compression of the gas, usually nitrogen, leads to an increase in its temperature during charging, assuming non-isothermal process. When holding the charge, this leads to heat flux to the colder surrounding environment. The reduced gas temperature leads to a lower gas pressure, and thus to a reduction in the quantity of reusable energy. The thermal energy transfer occurs mainly through the shell of the accumulator to the surrounding atmosphere, but also to a lesser extent through the dividing element to the hydraulic fluid.

When excluding the changing of the gas type and external heat exchange the thermal efficiency of a pressure accumulator can be increased by improving the thermal isolation of the gas volume or by adding thermal inertia to the gas volume. The latter strategy, referred in this study as thermal regeneration, reduces the temperature buildup and therefore also the thermal flux exiting the device. Therefore, the gas pressure remains higher for expansion, i.e. energy reuse, phase and thus the devise's output energy is higher. In literature there are examples of utilization of both these strategies. For example, a study on automotive catalysator as the regenerative unit /3/ or the usage of lathing shavings for the same purpose /4/.

## 2. Measurement setup

This chapter describes the reach truck test setup, the used hydro-pneumatic accumulator and the measurements.

### 2.1. Reach truck test platform

The studied reach truck test platform is an indoor forklift that uses a 48 volt lead acid battery bank as its energy source. The hydraulically operated functions, including the lift/lower function studied here, are powered by a 14 kW induction motor. The lifting motion uses a telescoping mast structure with three cylinders, one in the lower lift zone i.e. free lift zone and two in the upper lift zone. However, due to the limited oil chamber volume in the accumulator, all the tests in this study are carried out in the free lift zone.

The main component of the system, illustrated in **Figure 1**, is a hydraulic transformer consisting of two interconnected pump/motor units for changing the ratio between the volume flow and the pressure entering and exiting the transformer. The four normally closed 2/2-valves in the circuit are used for diverting the volume flow between the mast cylinder, the transformer and the hydraulic accumulator.

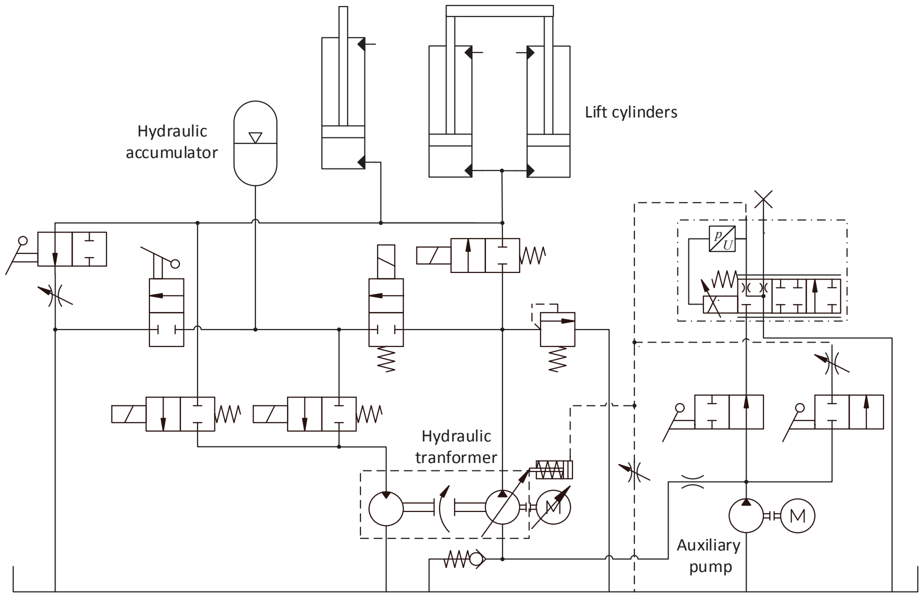


**Figure 1:** Hydraulic circuit

The two hydraulic machines in the transformer are connected with an overrun clutch. This enables the pump to operate without rotating the hydraulic motor when the accumulator is depleted. The system has an auxiliary pump for supplying the pressure control valve that controls the main pumps displacement. In addition, it is used to provide a small pressure to the pump of the transformer's pump to prevent cavitation.

The controlling of the system is realized with Simulink Real-Time software which is also used for data acquisition. The lowering velocity is controlled by creating torque equilibrium between the two hydraulic machines of the transformer. This is done by setting the pump displacement to a state in which the torque created by the pump against the pressure of accumulator is equal to the torque of the constant displacement motor created by the load induced pressure, equation 1.

$$T_{\mathrm{m}} = T_{\mathrm{p}} \rightarrow \frac{p_{\mathrm{cyl}} \cdot V_{\mathrm{m}}}{2 \cdot \pi} = \frac{p_{\mathrm{oil}} \cdot \varepsilon \cdot V_{\mathrm{p}}}{2 \cdot \pi} \rightarrow \varepsilon = \frac{p_{\mathrm{cyl}} \cdot V_{\mathrm{m}}}{p_{\mathrm{oil}} \cdot V_{\mathrm{p}}} \tag{1}$$

The machine efficiencies are ignored from the equation. The main component of the swash plate angle controller is a PI-controller. However, in order to achieve adequate response time and stability the controller was supplemented with a feed-forward coupled initial value calculator for adjusting the controller's command value close to the correct command value via utilizing equation 1 to solve ε. The controller subsystem also contains sub-blocks for additional functions, such as safety logics.

### 2.2. The hydraulic thermally regenerative accumulator

The hydraulic accumulator used in this study is a custom designed piston type pressure accumulator with a gas volume of 9.6 liters. The accumulator is designed to facilitate the insertion of the thermal regeneration unit to the accumulator without limiting the limits of the piston motion. The oil chamber volume of the device is 3 liters. The accumulator is illustrated in **Figure 2**.

**Figure 2:** The accumulator

The regeneration unit used in this study is a structure created of steel meshes with a total weight of 5 kg. With the regenerator installed the maximum gas volume is reduced to 9 liters. The accumulator was instrumented with a linear position sensor for measuring the piston position, two pressure sensors for measuring both the gas and oil side volumes and a thermocouple installed to the gas volume.

### 2.3. Measurements

The test platform was instrumented with a range of transducers and sensors for measuring pressures, position, torque, temperatures as well as rotational velocity of the electric motor and the vertical fork position. The energy consumption of the machine is derived from the measured current drawn from the main energy source and of its voltage.

For this, the positive output lead of the battery was instrumented with a LEM DH500-420L B current transducer. The voltage of the battery was scaled down with a voltage divider to downscale the signal level to one acceptable by the measurement hardware. The electric energy consumption drawn from the battery pack is defined by integrating the momentary electric power output, equation 2.

$$E_{\mathrm{elec}} = \int_{t_0}^{t_1} P_{\mathrm{elec}} \mathrm{d}t = \int_{t_0}^{t_1} (U \cdot I) \mathrm{d}t \tag{2}$$

Since the measurements are conducted using a finite sampling frequency the calculation of the energy consumption is discretized to equation 3, where $\Delta t$ denotes the sample time of 1 ms that was used in the measurements.

$$E_{\mathrm{elec}} = \sum P_{\mathrm{elec}} \cdot \Delta t = \sum U \cdot I \cdot \Delta t \tag{3}$$

The accumulator efficiency is defined by using the *pV*-diagram of the charge/discharge cycle of the accumulator, as given in equation 4.

$$\eta_{\mathrm{accu}} = \frac{W_{\mathrm{discharge}}}{W_{\mathrm{charge}}} = \frac{\int_{t_2}^{t_3} P_{\mathrm{h,dischage}} \mathrm{d}t}{\int_{t_0}^{t_1} P_{\mathrm{h,charge}} \mathrm{d}t} = \frac{\int_{t_2}^{t_3} (p \cdot q_{\mathrm{V}}) \mathrm{d}t}{\int_{t_0}^{t_1} (p \cdot q_{\mathrm{V}}) \mathrm{d}t} \tag{4}$$

Every measurement point in these sets consists of a cycle which includes consecutive lift, lower and lift-operations. The first lift is done without assistance from the accumulators, i.e. with an empty accumulator, and the latter assisted by energy recovered during the lowering. The energy consumption reduction ratio is defined with equation 5.

$$\Gamma = \frac{E_{\mathrm{unassisted}} - E_{\mathrm{assisted}}}{E_{\mathrm{unassisted}}}, \tag{5}$$

The accumulator charge is held for a variable duration between the lowering motion and the second lift motion. The tests were conducted using two different payloads and for each, with four different holding times. Furthermore, the tests were carried out with and without the regenerative unit in the accumulator. The selected loads were 1000 kg and 1500 kg and the lifting and lowering velocity was set to 0.3 m/s and lift height to 2.5 m.

The used holding times were 0, 30, 60 and 120 seconds. The test parameters are presented in **Table 1**.

| Payload [kg] | | | |
|---|---|---|---|
| 1000 | | 1500 | |
| Without regenerator | With regenerator | Without regenerator | With regenerator |
| $t_h$ = 0 s | $t_h$ = 0 s | $t_h$ = 0 s | $t_h$ = 0 s |
| $t_h$ = 30 s | $t_h$ = 30 s | $t_h$ = 30 s | $t_h$ = 30 s |
| $t_h$ = 60 s | $t_h$ = 60 s | $t_h$ = 60 s | $t_h$ = 60 s |
| $t_h$ = 120 s | $t_h$ = 120 s | $t_h$ = 120 s | $t_h$ = 120 s |

**Table 1:** Measurement parameters, $v$ = 0.3 m/s, $h$ = 2.5 m

Due to delays in the human-machine interface the target of 0 second holding times were not met and the realized holding times for these sets were measured of being on average 3 seconds.

## 3. Results

The calculated accumulator thermal efficiencies in the 16 measurements are presented in **Table 2**. As predicted, the accumulator efficiencies with the regeneration unit installed to the accumulator were higher than the ones without the regenerator in all measurement points.

| | Accumulator efficiencies | | | |
|---|---|---|---|---|
| | Payload [kg] | | | |
| | 1000 | | 1500 | |
| Holding time [s] | Without regenerator | With regenerator | Without regenerator | With regenerator |
| $t_h$ = 0 s | 97 % | 98 % | 91 % | 97 % |
| $t_h$ = 30 s | 92 % | 96 % | 86 % | 96 % |
| $t_h$ = 60 s | 89 % | 95 % | 85 % | 96 % |
| $t_h$ = 120 s | 89 % | 95 % | 83 % | 96 % |

**Table 2:** Accumulator thermal efficiencies

With higher payload the compression ratio of the accumulator also becomes higher, and thus the effect of the regenerator is more significant. The pressure-volume graphs for the 1500 kg payload with the shortest and longest holding times are shown in **Figure 3**, both with and without the regeneration unit. The area between the charging and discharging cycles depicts thermal losses. The scales on the volume axis are not uniform due to the reduced gas volume with the regenerator installed.

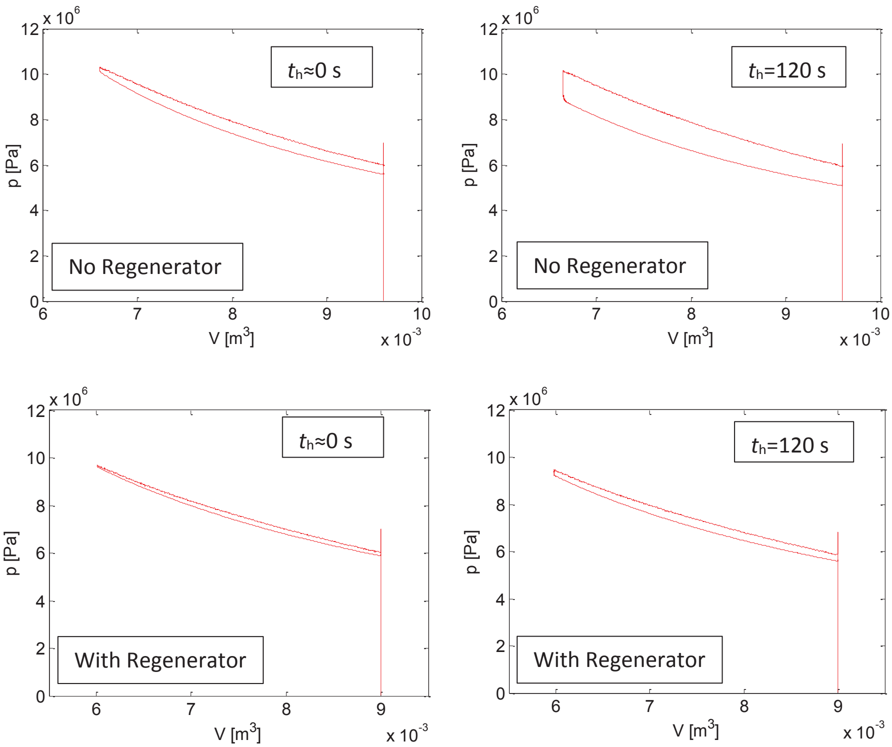


**Figure 3:** pV-graphs of accumulator: payload 1500 kg. Upper row: without regenerator. Lower row: with regenerator. Holding times: left 0 s, right 120 s

The effect of the thermal characteristics to the accumulator's gas pressure, with and without the regenerator, is depicted in **Figure 4**. They are taken from the measurements where the difference is at largest, i.e. with the 1500 kg payload and the holding time of 120 seconds.

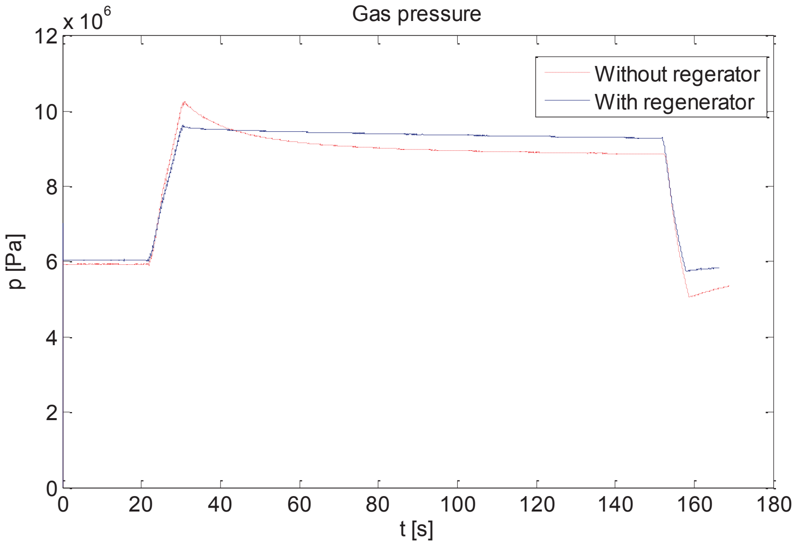


**Figure 4:** Accumulator gas pressure; payload 1500 kg, holding time 120 s

The measured energy consumption reduction ratios of the reach truck are presented in **Table 3**. Without the regenerator they decrease with the increase of holding time. With the regenerator, they remain relatively constant throughout the used holding times with the exception of the measurement with 1500 kg payload and 120 s holding time.

| | Energy consumption reduction ratios | | | |
|---|---|---|---|---|
| | Payload [kg] | | | |
| | 1000 | | 1500 | |
| Holding time [s] | Without regenerator | With regenerator | Without regenerator | With regenerator |
| $t_h = 0$ s | 24 % | 23 % | 31 % | 32 % |
| $t_h = 30$ s | 22 % | 23 % | 27 % | 31 % |
| $t_h = 60$ s | 19 % | 23 % | 27 % | 31 % |
| $t_h = 120$ s | 18 % | 24 % | 26 % | 29 % |

**Table 3:** Energy consumption reduction ratios

The measured electric energy consumption of the reach truck for the measurement with a payload of 1500 kg and a 0 second holding time with the regenerator installed is presented in **Figure 5**.

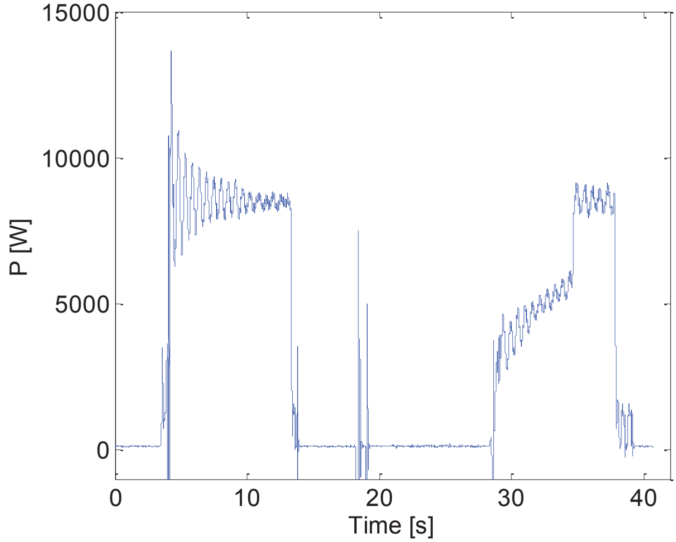


**Figure 5:** Measured energy consumption of the reach truck: payload 1500 kg, holding time ~0 s

## 4. Discussion

With higher loads, the energy stored in the accumulator also becomes higher which in terms leads to higher thermal losses in the holding phase and thus emphasizes the effect of the thermal regeneration unit. Therefore, the relatively high loads of 1000 and 1500 kg were selected for the measurements in this study.

The measurements indicated that the accumulator thermal efficiencies ranged from 97 to 89 % with a 1000 kg payload and from 91 to 83 % with a 1500 kg payload when operating without the thermal regeneration unit. With the regenerator the corresponding efficiency ranges were from 98 to 95 % and from 97 to 96 % with payloads of 1000 and 1500 kg respectively. The accumulator's thermal efficiency with the regenerator was measured of being higher in all of the measurements than its efficiency without the regenerator in corresponding measurement. Especially, when using longer holding times the efficiency with the regenerator was much higher. The largest uncertainties in these values are in the measurements with the 0 second holding times. As described in chapter 2.3., the actual values of these holding times can be as long as 4 seconds due to the command interface. In the measurements without the regenerator these few second can create a large effect due to the rapid initial fall of the pressure, as presented in Figure 4. For example, if the holding time would be precisely zero, the 91 % efficiency gained with the 1500 kg payload and without the regenerator should be roughly equal the 98 % efficiency measured with the regenerator.

Without the regenerator the energy consumption reduction ratios declined with increasing holding times from 24 to 18 % and from 31 to 26 % with the payloads of 1000 and 1500 kg respectively. With the regenerator, these ratios remained relatively constant near 23 % and 31 % when using the payloads of 1000 and 1500 kg respectively, with the exception of the 29 % reduction for the measurement with a holding time of 120 s and a payload of 1500 kg. This result was verified with multiple repetitions, however the cause of the anomaly was not found within the time restraints of this study and therefore requires further examination.

The auxiliary pump in the system was installed to produce the pressure required by the valve controlling the pump’s swash plate angle. In addition, it was used for producing a slight pressure to the pump inlet port. The hydraulic assistance power was measured of being roughly 60 W which is two orders of magnitude smaller than the lifting powers. Furthermore, in an actual production device the arrangement should be replaced by a pressurized tank. For these reasons, the power consumption of the auxiliary circuit was excluded from the results.

## 5. Conclusions

The regenerator was found to be a very effective method for improving the thermal efficiency of the used piston typed hydraulic accumulator. The efficiency remained at or above 95 % in all of the measurements, while without the regenerator it was at lowest 83 %. The usage of the regenerator also prevented the energy consumption reduction ratios to decline in a noteworthy manner with increasing holding time, with the one previously discussed exception. However, even in this measurement point the reduction ratio was well above the one measured without the regenerator.

With higher compression ratios the effect of the regenerator would have been greater, however the accumulator’s maximum oil volume of three liters limited the usage of higher compression ratios.

## 6. References

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## 7. Nomenclature

| | | |
|---|---|---|
| $E_{elec}$ | Energy – electric form | J |
| $E_{assisted}$ | Energy – with hydraulic assistance | J |
| $E_{unassisted}$ | Energy – without hydraulic assistance | J |
| $I$ | Current | A |
| $p_{oil}$ | Accumulator oil pressure | Pa |
| $p_{cyl}$ | Cylinder pressure | Pa |
| $p_{gas}$ | Accumulator gas pressure | Pa |
| $P$ | Power | W |
| $t_h$ | Holding time | s |
| $\Delta t$ | Sample time | ms |
| $U$ | Voltage | V |
| $\varepsilon$ | Setting of displacement (0…1) | - |
| $\Gamma$ | Consumption reduction ratio | - |
| $\eta_{accu}$ | Accumulator thermal efficiency | - |