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IMPROVING ENERGY EFFICIENCY OF A REACH STACKER USING A POTENTIAL ENERGY RECOVERY SYSTEM

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ABSTRACT

This paper aims to compare three solutions of potential energy recuperation on a mobile harbour machine designed to carry and stack containers. The boom actuated by two hydraulic cylinders can lift and lower loads up to 45 tons. The current system dissipates the energy during boom lowering through a flow control valve.

The three presented systems use hydropneumatic accumulators to temporarily store the energy and then release it during a boom lifting or any power demand. The first system uses only a flow control valve to reach the pressure imposed by the hydropneumatic accumulator. The second system uses a transformer based configuration directly coupled to the internal combustion engine (ICE). This layout allows the stored energy to be easily released but also to be recovered regardless of the pressure difference between the hydraulic circuit and the hydropneumatic accumulator. The last presented system is also composed of a pressure transformer which is not attached to the ICE. This solution enables the recuperation devices to have a rotation speed independent of the ICE but also to avoid the engine braking.

The solutions have been modelled and simulated for different initial and final positions of the container in terms of height and depth. A duty cycle has been performed giving for each solution a global view of the fuel savings. The system without transformer showed an amount of energy recovered lower than 37% because of the limitation of the accumulator volume. The second solution directly coupled to the engine shaft demonstrated better recuperation performances. However between 15% and 20% of the recoverable energy is dissipated by the engine braking. The recuperation motor drives indeed the ICE during the boom lowering generating a parasitic negative torque. Besides, the relatively low speed imposed by the ICE induces the need for high displacement units to achieve the rod retraction speed specification. The last architecture showed fuel consumption economy of up to 16%. The great advantages of this solution is its independence of the ICE speed and the engine braking but also the possibility to recover energy even when the lifting actuator pressure is low, for instance when an empty container is lowered.

KEYWORDS: hydraulic hybrid, potential energy recovery, reach stacker, off-highway vehicle

1. INTRODUCTION

Since the end of the 1990' off-highway vehicles are subjected to more and more restrictive norms concerning pollutant emissions. We can mention for instance the US Tier IV Final standards or the EU Stage IV. Moreover owners of heavy machines are trying to reduce their operating costs to keep competitive. Since fuel costs related to the use of machines are an important part of the operational charges, the reduction of the consumption is a key topic. Lot of work has been done recently to find solutions to save fuel like improving the efficiency of already existing components or finding new architectures to transmit energy in a more efficient way [1] [2]. It is also sometime possible to recover the energy which is lost in traditional machines.

The case study focuses on an off-highway vehicle working in harbours or transport hubs whose task is to carry containers and stack them on the appropriate area. Lifted heavy containers represent a large amount of potential energy which is currently transformed into heat by the meter-out valve of the hydraulic circuit. Recovering the potential energy and use it later when the rest of the system has a power demand would result in a decrease of the required power of the internal combustion engine (ICE) and consequently fuel savings. In this paper, three different architectures of potential energy recovery systems are presented and compared by the mean of simulation. The three solutions are based on hydraulic components for different reasons. Firstly the high power density of hydraulics is suitable for recovering a important amount of energy during a short time [3]. Secondly working in the same energy domain as the rest of the system is a good way to limit energy transformations which induce automatically losses for each change of physical domain. Finally it is much simpler for the owner of the machine to maintain a system with similar components.

The part 2 of this paper describes the operating principle of the reference machine. The layouts and the modelling of the three solutions are presented in part 3 and the last part deals with simulation results and the comparison between each architecture. Future work will focus on the implementation of the selected solution on a real machine.

2. OPERATING PRINCIPLE OF THE MACHINE

2.1. Description of the reach stacker

Reach stackers are mobile machines capable of carrying containers up to 45 tons to a height of five standard containers and 35 tons in sixth height. Figure 1 shows a typical maximal load range of container stacking. The machine has an Internal Combustion Engine (ICE) as primary energy source providing the necessary power to several actuating lines permitting the machine to translate, lock and lift the container in order to move it to its next place. We can mention the powertrain composed of a torque converter and an automatic gearbox driving the energy to the front axle. The ICE also drives two hydraulic pumps both equipped with a Load Sensing (LS) system. The hydraulic fluid is supplied to the lifting and telescoping cylinders via two proportional valves controlling the flow rate. The two lifting cylinders and the telescoping cylinder are the main actuators as they require the highest flow rates. Table 1 presents the main features of the ICE and hydraulic circuit. Finally the spreader interfaces the machine with the container via different smaller actuators and a twist-lock system.

Auxiliary components like steering system, cooling circuit, braking circuit or spreader actuators are neglected insofar as their energy consumption is much lower than the other actuator needs. Thus are selected the powertrain, the lifting and telescoping system to be modelled. The powertrain is considered in order to simulate the functioning of the machine on a global duty cycle including translations and hydraulic movements. Finally a multi-body dynamic model is also established in order to represent accurately the behaviour of the machine.

Table 1. ICE and hydraulic features

Element	Feature	Value
Engine	Maximum power	235 kW
	Maximum torque	1580 N.m at 1260 rpm
Pump 1	Maximum displacement	145 cm ³
Pump 2	Maximum displacement	145 cm ³
Lifting cylinders	Maximum pressure	420 bar
	Length of stroke	2.95 m
Telescoping cylinder	Maximum pressure	350 bar
	Length of stroke	8.3 m

2.2. Baseline model

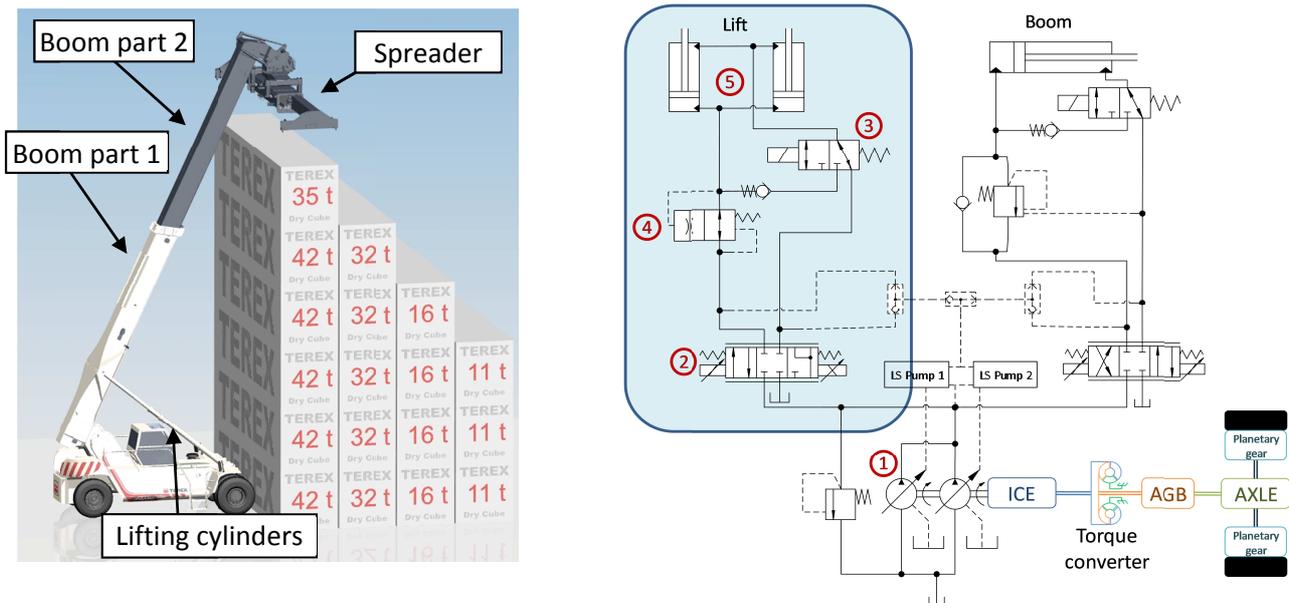


Figure 1. Baseline architecture of the reach stacker

In [4] a dynamic model of the baseline machine has been established and compared with measurements. A simplified outline of the machine is shown on Figure 1. This physical model which is divided into three connected submodels, namely the multibody, the hydraulic and the powertrain submodels has been used as a reference.

The dynamics of the ICE includes its inertia and shaft stiffness but the torque is deduced from the speed-torque characteristic curve given by the manufacturer. Concerning the powertrain the torque converter has been dynamically modelled and the gearbox behaviour like transmission shift points and internal clutches engagement has been taken from measurements. Those measurements also permitted to define the rolling resistance and the losses along the powertrain.

Pump inertias and fluid compressibility are considered in the model and pressure drops for each valves have been correlated with the reality. The low pressure circuit dedicated to spools actuation as well as the load

sensing system are neglected because of their low energy consumption. Both are replaced by non powered lines transmitting only information.

In this paper the powertrain and the telescoping circuit are not subjected to any modification of their components. The lifting circuit on its side is modified to recover the lost energy during boom-downs. The actual system is based on two LS pumps (Figure 1 (1)) providing the desired flow to the actuators (5) via a proportional valve (2). We can notice that both lifting and telescoping circuit have regeneration valves (3) which are only used when unloaded. Oil outgoing from the rod chamber is added to the pump flow resulting in a faster extension than in normal operating. In loaded phases the reached pressure in the piston chambers would be too high that is why the rod chamber is in this case connected to the tank. During boom down the flow limiter (4) added to the lifting circuit avoids lowering overspeeds especially in loaded case. The hydraulic pressure is dissipated in this valve to maintain a maximal predefined lowering speed. The rest of the hydraulic energy is lost inside the proportional valve.

2.3. Work cycle

Reach stackers are working mostly on container storage areas and their task which is repetitive consists of unstacking and stacking containers from one place to another like unloading a truck and stacking the container in the storage area or the contrary. The duty cycle employed to simulate the functioning of the machine is divided into two parts. The first part represents an unstacking stage followed by a truck loading and the second part is exactly the contrary. This configuration enables the machine to face all different situations of rolling, lifting and telescoping at loaded and unloaded cases.

The translation speed profile remains always the same whereas the hydraulic side is highly variable. The initial position of the container (row, height) changes the required length and angle of the boom to handle the container so that the energy distribution between lifting and telescoping will be different for each case. Figure 2 shows an example of a working cycle in terms of vehicle speed, boom angle, boom length and load for a given initial position of the container. A change of this initial position modifies the cycle except the vehicle speed profile meaning that there are as many different duty cycle as container positions. Finally the work cycle also includes an obligatory boom position for translation. Indeed, the telescope must be completely retracted and the boom angle must be approximately 45° to avoid any unbalance situation which could be dangerous during steering or braking phases. Thus if a container is picked up at the second height the driver will be obliged to lift the boom until 45° in order to provide sufficient visibility to drive. As the boom is necessarily elevated for each cycle the interest in a regeneration system on the lifting circuit increases.

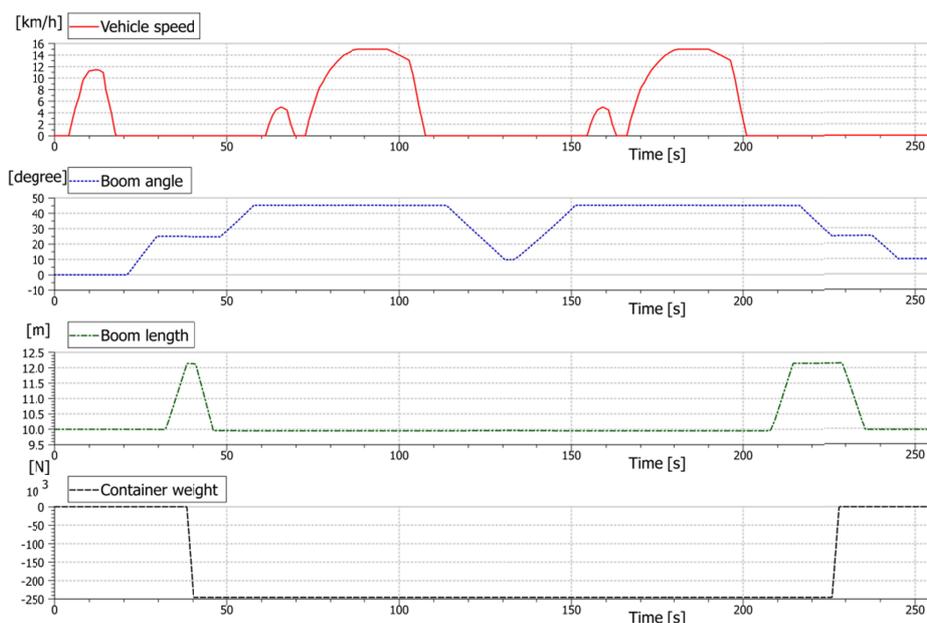


Figure 2. Duty cycle

3. STUDIED ARCHITECTURES

3.1. Presentation of the solutions

3.1.1. Solution 1: Meter-out control valve based system

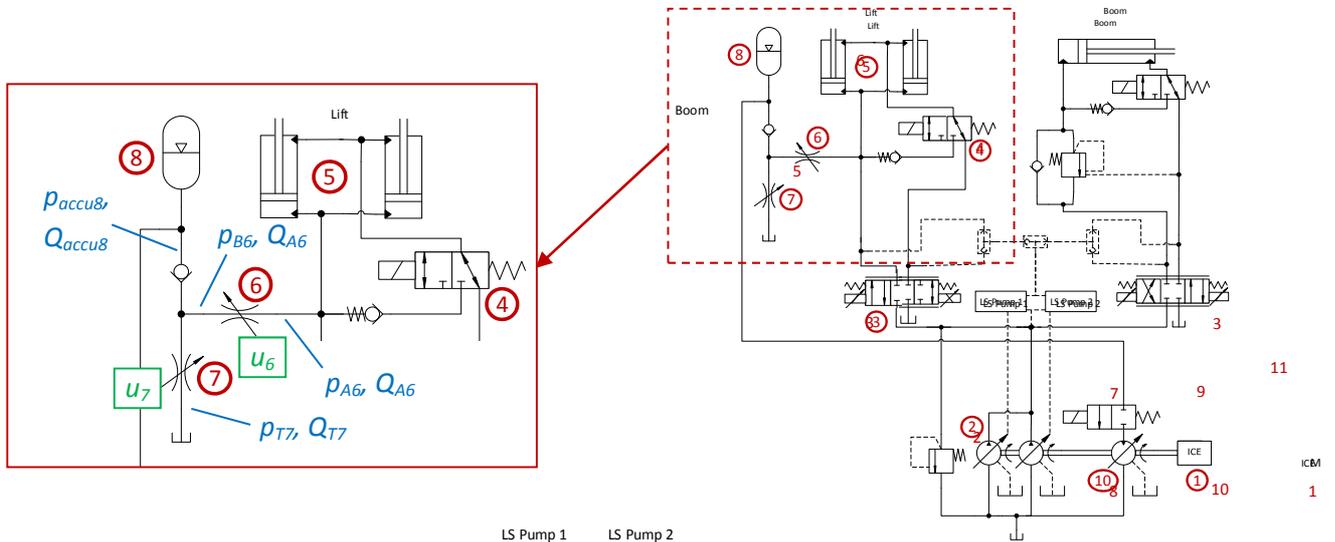


Figure 3. Hydraulic circuit with meter-out control valve based energy recovery system

The first solution illustrated in Figure 3 uses the inherent advantage of the lifting system. Indeed, when the boom angle decreases the necessary force to hold the load rises resulting in a higher pressure in the lifting cylinders. This intrinsic feature is suitable for the implementation of this solution because the pressure of the cylinders will rise together with the accumulator pressure, enabling the system to continue recovering a part of the energy unless the maximum pressure of the storage is reached.

A similar concept presented in [5] shows a regenerative system connected to the piston chamber of the lifting cylinders of an excavator. While boom-down a proportional valve and a pressure compensator adjusts the desired flow and transmits it either to an accumulator or to the suction line of the main pump so that its power demand for feeding other actuators is reduced. Thus the accumulator volume can be reduced because the major part of the regenerated energy is instantaneously reused by the pump. In our case the boom lowering is often carried out in a single operation. Moreover the volume of fluid to regenerate is much bigger than for an excavator. Those elements led to some modifications to adapt this system to our case. In [6] the authors present a similar architecture which demonstrated 10% fuel savings. However a digital valve package was used to achieve the energy recovery.

Here we use two piloted restrictors (6) and (7) to control the rod retraction speed. The flow goes from the lifting cylinders (5) to the accumulator (8) until its pressure is too high to keep the desired flow rate.

During a rod retraction, if the piston chamber pressure is higher than the accumulator pressure, the piloted flow restrictor (6) opens in such a way that it dissipates a part of the energy by creating a pressure difference in order to reach the accumulator pressure. The higher is the pressure difference between piston chamber and accumulator the smaller is the flow restrictor opening. While the valve (6) is not fully opened the restrictor (7) stay closed. A totally opened valve (6) means that the pressure difference is not high enough to hold the desired flow rate. At this time the valve (7) opens and dissipates the extra flow. As the cylinder pressure tends to rise during a rod retraction, the accumulator is still able to regenerate a part of the energy.

To reuse the previously stored energy, a variable displacement motor (10) is added to the engine shaft. The accumulator feeds the motor whose torque helps the ICE to drive the main pumps, the auxiliary components or the powertrain.

This system has the advantage of being very simple and the need of few extra components makes it a low cost solution. However the recovered energy might be low unless a huge accumulator is used.

3.1.2. Solution 2: ICE-dependant transformer based system

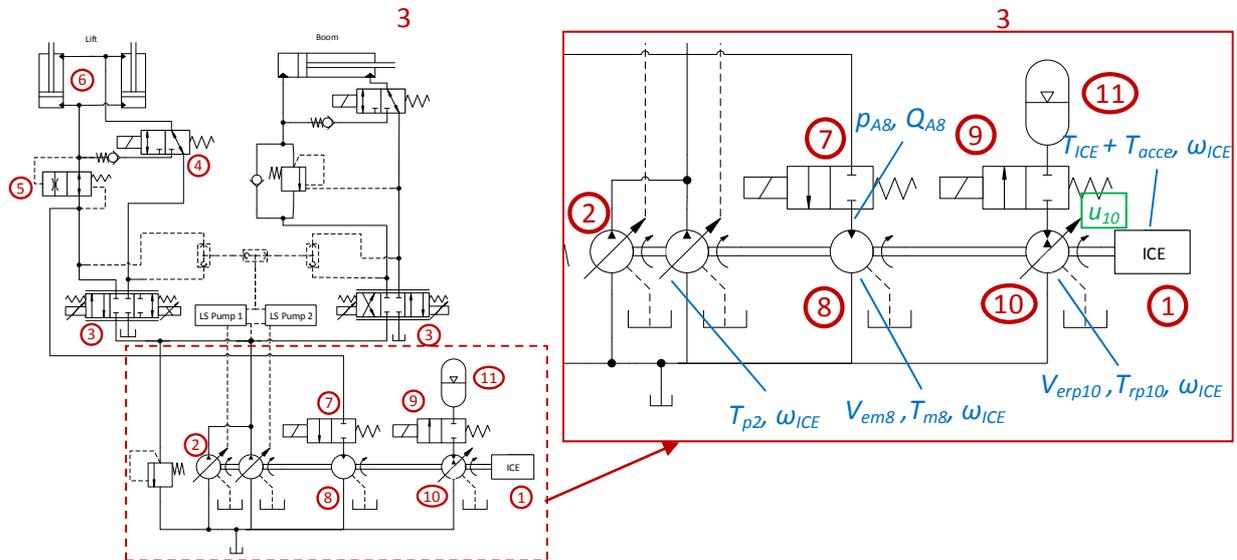


Figure 4. Hydraulic circuit with the common shaft transformer based energy recovery system

Similar regeneration architectures to the second solution shown in Figure 4 have been introduced in the literature, however we can find some differences. In [7] a common shaft for all units is employed, nevertheless the authors are using the same unit to drive the actuator and to recover the potential energy thanks to a displacement controlled (DC) system. In this machine such a configuration would require too many DC units, therefore an independent motor-pump system (Figure 4 (8),(10)) is used to recuperate the energy and reuse it.

In [8] the same motor is used to recover and to reuse the energy and the main pump also has a double function, since it can either feed the actuators during normal operation or store the energy into accumulators during energy recovery phases. On the contrary the proposed layout permits to combine energy recovery of the lifting system with other actuations. Indeed, some functions like steering must remain available even during short periods. Thus the main pumps (2) and the recovery system (7-11) are completely independent. Finally the layout presented in [9] has an extra pump-motor attached to the ICE which can either store energy during low load actuations and release it in peak power operation or recuperate energy during overruns. This solution appears to be inappropriate as our case study is characterized by long periods of high power requirement.

During a boom lowering the on-off valve (7) opens to connect the piston chamber of the lifting cylinders to the motor. The torque produced is transmitted to the pump (10) which stores the energy in the accumulator (11). The motor can also assist the ICE (1) to drive the main units (2) if there is a power demand at the same time. The case of a boom lowering without any other power demand is disadvantageous because an important part of the motor torque is dissipated into the engine braking and the friction torque of the pumps. At low speed those unwanted parasitic torques are reduced and the recuperated energy is higher. Nevertheless the displacement of the motor and the pump will have to increase to absorb the entire flow rate.

Compared to the solution presented in 3.1.1 it is here possible to adjust the displacement of the recovery unit (10) permitting more energy to be recuperated even when the pressure at the cylinder side is different from the accumulator side.

3.1.3. Solution 3: ICE-independent transformer based system

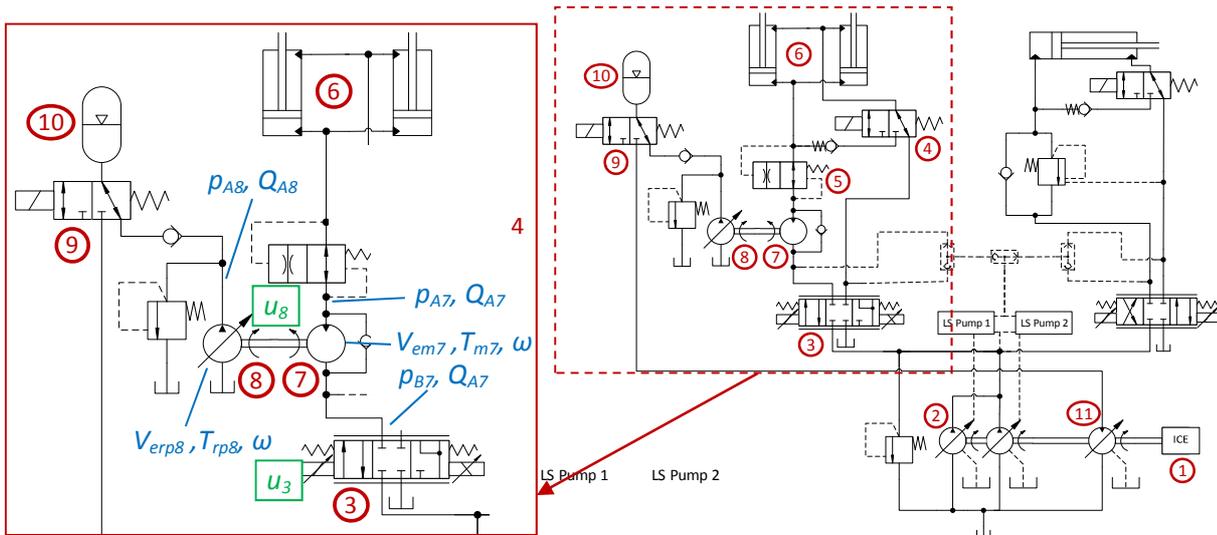


Figure 5. Hydraulic circuit with the independent shaft transformer based energy recovery system

The last solution studied (Figure 5) is also based on a motor-pump system (7-8) to recover the potential energy. In [10] the authors present a transformer based system and in [11], [12] a control strategy is developed but no energetic evaluation of the solution is proposed.

The idea is to reduce the displacement of the recuperation units by making them rotate faster. This is now possible thanks to the new location of the transformer which is directly integrated on the hydraulic line between the proportional valve (3) and the lifting cylinders (6). The recovery phases are independent from the engine braking and pump drag allowing more energy to be recuperated.

When the boom is lowered the pressure difference at each port of the motor creates a torque driving the variable displacement pump. If the accumulator pressure is not high enough to hold the load, then the pump displacement increases. The proportional valve (3) regulates the pressure difference at the motor side so that the load cannot drive the pump faster than desired. The stored energy is then released to the system by switching the valve (9). The accumulator (10) is discharged via an extra motor (11) whose variable displacement enables the assisting torque to be controlled.

3.1.4. Energy reuse strategy

For the solutions 1 and 3 an extra motor of 100 cm³ is attached to the ICE shaft in order to help the engine to drive the main pumps or the powertrain. This small unit has several advantages: the first one is its low price which is a good point for an additional system. Secondly it has a low friction torque at zero displacement meaning that the impact when it is not used is low. Finally a high displacement unit would provide an important torque making the ICE work at low load conditions corresponding to bad efficiencies working points. The second solution meanwhile uses the recovery pump as a motor to achieve the same task.

3.2. Modelling and control

3.2.1. Meter-out control valve based system

As explained in section 3.1.1 the opening signal u_6 of the valve (6)(Eqn.3) depends on the pressure at each port but also on the joystick position k_L establishing the cylinder speed instruction. The second valve (7) signal u_7 permits to keep the desired flow rate even if the accumulator pressure is too high (Eqn.8).

Valve (6) opening signal calculation:

$$k_L Q_{max} = C_{q6} A_6(u_6) \sqrt{\frac{2(p_{A6} - p_{B6})}{\rho}} \quad (1)$$

$$\text{with } A_6(u_6) = A_{max6} u_6 \text{ for } u_6 \in [0; 1] \quad (2)$$

$$u_6 = \frac{k_L Q_{max}}{C_{q6} A_{max6}} \sqrt{\frac{\rho}{2(p_{B6} - p_{A6})}} \quad (3)$$

Valve (7) opening signal calculation:

$$\text{If } u_6 < 1 \text{ then } u_7 = 0 \quad (4)$$

If $u_6 = 1$ then u_7 is calculated as follows

$$k_L Q_{max} = Q_{accu8} + C_{q7} A_7(u_7) \sqrt{\frac{2(p_{B6} - p_{T7})}{\rho}} \quad (5)$$

$$\text{with } A_7(u_7) = A_{max7} \cdot u_7 \text{ for } u_7 \in [0; 1] \quad (6)$$

$$\text{and } Q_{accu8} = C_{qCV} A_{CV} \sqrt{\frac{2(p_{B6} - p_{accu8})}{\rho}} \quad (7)$$

$$\left\{ \begin{array}{l} u_6 = 1 \\ u_7 = \frac{k_L Q_{max} - Q_{accu8}}{C_{q7} A_{max7}} \sqrt{\frac{\rho}{2(p_{B6} - p_{T7})}} \end{array} \right. \quad (8)$$

3.2.2. ICE-dependant transformer based system

The fixed displacement unit (8) makes its rotation speed target directly proportional to the rod retraction velocity. Equation (10) shows that the motor speed depends on different resistive torques due to the accessories, the engine braking and the main pumps. The displacement signal u_{10} of the pump (10) is evaluated (Eqn.12) to control this rotation speed. Thus if the torque provided by the motor is entirely used by the main pumps, the accessories and the engine braking, then the recovery pump displacement is set to zero. On the one hand the instantaneous reuse of the recovered energy by the accessories or the main pumps tends to downsize the storage capacity. But on the other hand the engine braking and the friction of the pumps at zero displacement induce parasitic torques which drastically reduce the energy savings.

$$T_{m8} = \frac{V_{em8}(p_{A8} - p_T)}{20\pi} \quad (9)$$

$$J_{ICE} \frac{d\omega_{ICE}}{dt} = T_{m8} - T_{rp10} - T_{ICE} - T_{p2} - T_{acce} \quad (10)$$

$$\text{At constant speed, } \frac{d\omega_{ICE}}{dt} = 0$$

$$T_{rp10} = T_{m8} - T_{ICE} - T_{p2} - T_{acce} \quad (11)$$

$$u_{10} = \frac{T_{m8} - T_{ICE} - T_{p2} - T_{acce}}{V_{erp10}(p_{accu11} - p_T)} \eta_{m,rp10} \cdot 20\pi \quad (12)$$

3.2.3. ICE-independent transformer based system

The evaluation of the needed pump displacement (Eqn.14) to ensure a sufficient resistive torque for the hydraulic motor is similar to the previous section. Nevertheless no additional torque is present since the shaft is independent of the ICE. Thus the pump displacement controls alone the boom lowering speed unless the resistive torque is not high enough. In this case the signal u_3 controls the proportional valve opening (Eqn.16) making the back pressure of the motor higher resulting in a lower torque transmitted to the pump.

$$(J_7 + J_8) \frac{d\omega}{dt} = T_{m7}(u_3) - T_{rp8}(u_8) \quad (13)$$

At constu spee , $\frac{d\omega_3}{dt} = 0$

$$u_8 = \frac{V_{em7}(p_{B7} - p_{A7})\eta_{m,m7}\eta_{m,rp7}}{V_{erp8}(p_{A8} - p_T)} \quad (14)$$

If $u_8 < 1$, then $u_3 = -1$ (15)

If $u_8 = 1$, then $u_3 = -\frac{k_L Q_{max}}{C_{q3} A_{max3}} \sqrt{\frac{\rho}{2(p_{B7} - p_T)}}$ (16)

with $p_{B7} = \frac{V_{erp8}(p_{A8} - p_T)}{V_{em7}\eta_{m,m7}\eta_{m,rp8}} + p_{A7}$ (17)

4. SIMULATION RESULTS

Table 2. Sizing of the hybrid systems

Element	Solution 1	Solution 2	Solution 3
Recovery motor disp. [cm ³]	-	220	160
Recovery pump disp. [cm ³]	-	145	190
Energy reuse motor disp. [cm ³]	100	-	100
Accumulator volume [L]	150	150	150

The sizing steps for each solution are not detailed but the Table 2 presents the displacements of the pumps and motors. The sizing of the accumulator volume depends on the amount of energy to be stored but also on the selected operating pressure range. Considering a 45t load lowered from the 5th height to the ground led to highly oversized storage capacities, that is why the average container weight and height (25t, 3rd height) were considered instead, resulting in a storage capacity of 150L. This volume corresponds also to the maximal available space for accumulators inside the machine. Finally the pressure operating points are adapted for each solution to optimize the amount of recovered energy.

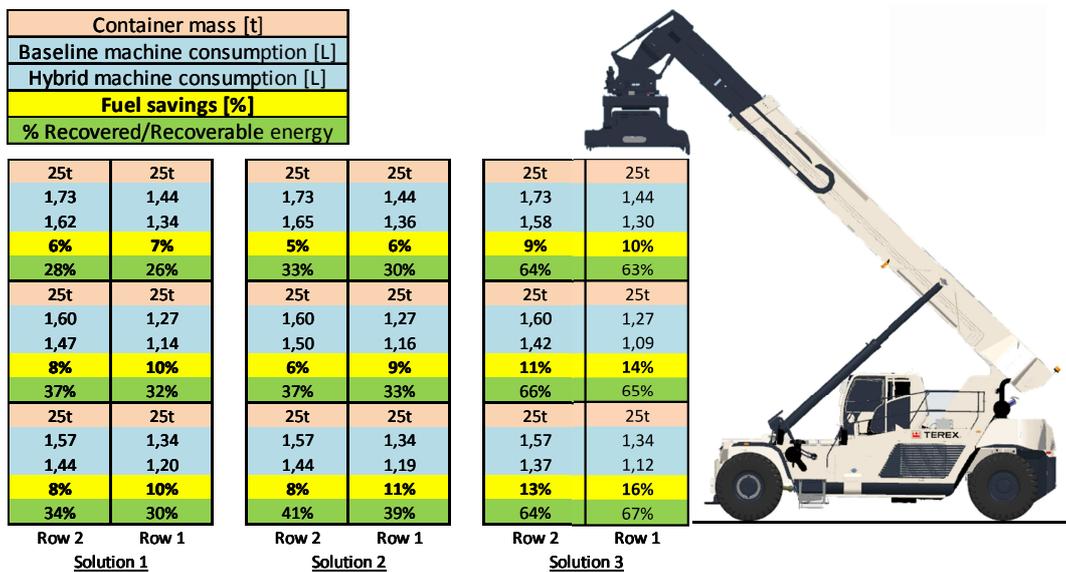


Figure 6. Simulation results

The three models has then been simulated considering the duty cycle previously presented. For each initial position of the container an energetic analysis was conducted but the focus has been made on the two first rows and the three first heights of container stacks since they represent most cases. Figure 6 shows fuel savings for each initial position of the container and each considered solution. The first and second solutions achieve between 5% and 11% fuel saving whereas the third solution is capable of consumption reductions up to 16%.

Solution 1 shows poor energy recovery possibilities due to its architecture without transformer. We can see on Figure 7 (a) that during a boom lowering the recovery can be split into two phases. Between the point A and B the pressure difference between the cylinder and the accumulator is high and almost all the flow is transferred to the accumulator (Figure 7 (b)). From point B to C the restriction connected to the tank starts opening wider inducing a flow rate to the accumulator much smaller. The remaining pressure difference corresponds to the pressures losses through the first restriction and the check valve. The system dynamics can theoretically be identical to the traditional solution, however we can see on Figure 7 (b) that during the transition between a full and a partial recovery (around point B) the flow rate tends to slow down. Here the control of the system should be improved to overcome this situation.

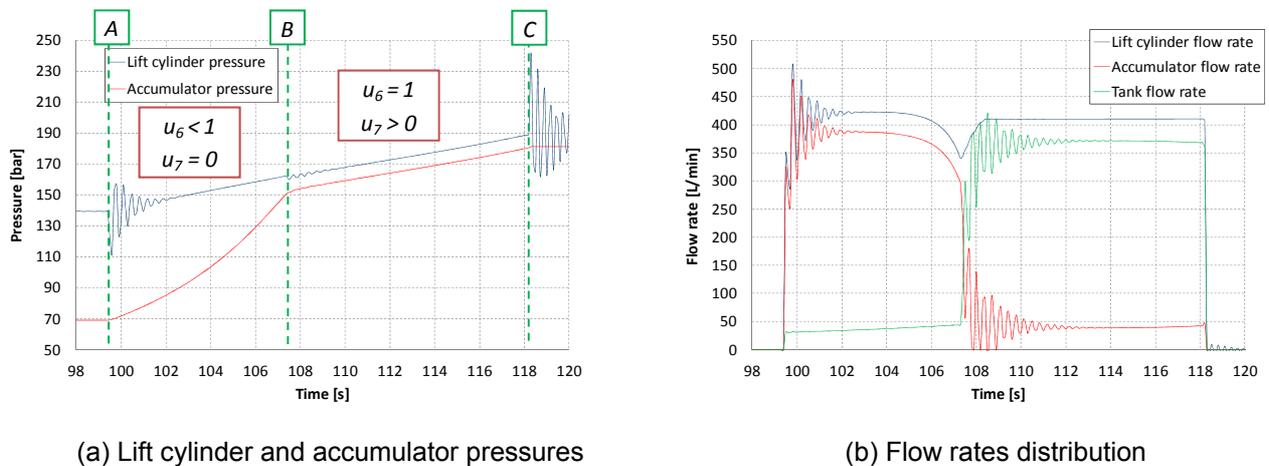


Figure 7. Solution 1: Pressures and flows during a recovery phase (boom angle from 45° to 10° and container of 25t)

The second solution achieves better recuperation performances but only in certain conditions. Indeed the system is able to recover between 40% and 50% of the potential energy when a heavy container is lowered. However the duty cycle also has boom-downs without container. In these conditions the lift cylinder pressure is too low to drive the hydraulic motor due to the numerous resistive torques. The potential energy cannot be recovered and a traditional throttling valve is used to dissipate this energy.

Figure 8 shows the torque distribution provided by the recuperation motor. From point D to E the desired lowering speed is not reached that is why the pump is not activated meaning that no energy is stored. Nevertheless about 30% of the torque is immediately reused by the accessories but the rest is dissipated through the pumps friction at zero displacement, the torque converter, the engine braking and the acceleration of the inertias. Furthermore a clearly identified drawback is the slow dynamics brought by uncontrolled resistive torques.

From point E to F the pump controls the lowering speed via its displacement and the energy storing begins. The recuperated energy on the Figure 6 is calculated by comparing the total recoverable potential energy with the stored energy plus the immediately reused energy. We can also see that the fuel savings are similar to the first solution despite of the better recuperation performances. In fact the friction losses at zero displacement generated by the additional motor and pump bring an extra torque to the ICE which consumes more fuel outside the recovery phases. This side effect decreases strongly the benefits brought by this system.

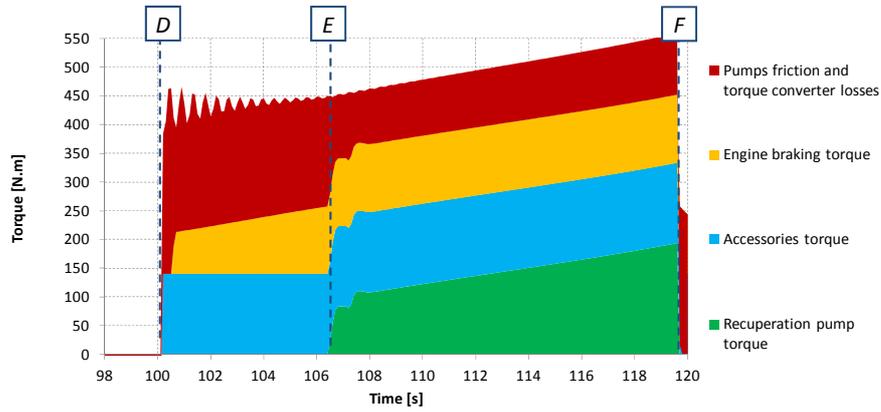


Figure 8. Solution 2: Torques distribution during a recovery phase (boom angle from 45° to 10° and container of 25t)

The last proposed architecture solves a part of the problem by using only a small unit attached to the ICE. The friction losses are then much smaller compared to the solution 2. We can observe on Figure 9 (d) that the motor consumes few energy when it is not used. Between the point G and H a container of 25t is lowered generating a torque to the pump which fills the accumulator. The pump starts the recovery phase at full displacement as the accumulator pressure is low and decreases gradually until 30% since the cylinder pressure increases slower than the accumulator pressure. In a second time the previously stored energy is released via the motor to help the ICE to drive the main pumps for a further boom lifting. From point I to J the boom is lowered from the driving position (boom angle at about 45°) to the container release position and between the point J and K the boom is lowered without any container. We can see that it is still possible to recover energy thanks to the variable displacement pump. This feature together with the independence from the engine braking and the other parasitic torques lead to better performances. We can see on Figure 6 that the rate of recovered energy is almost twice compared to the two first solutions.

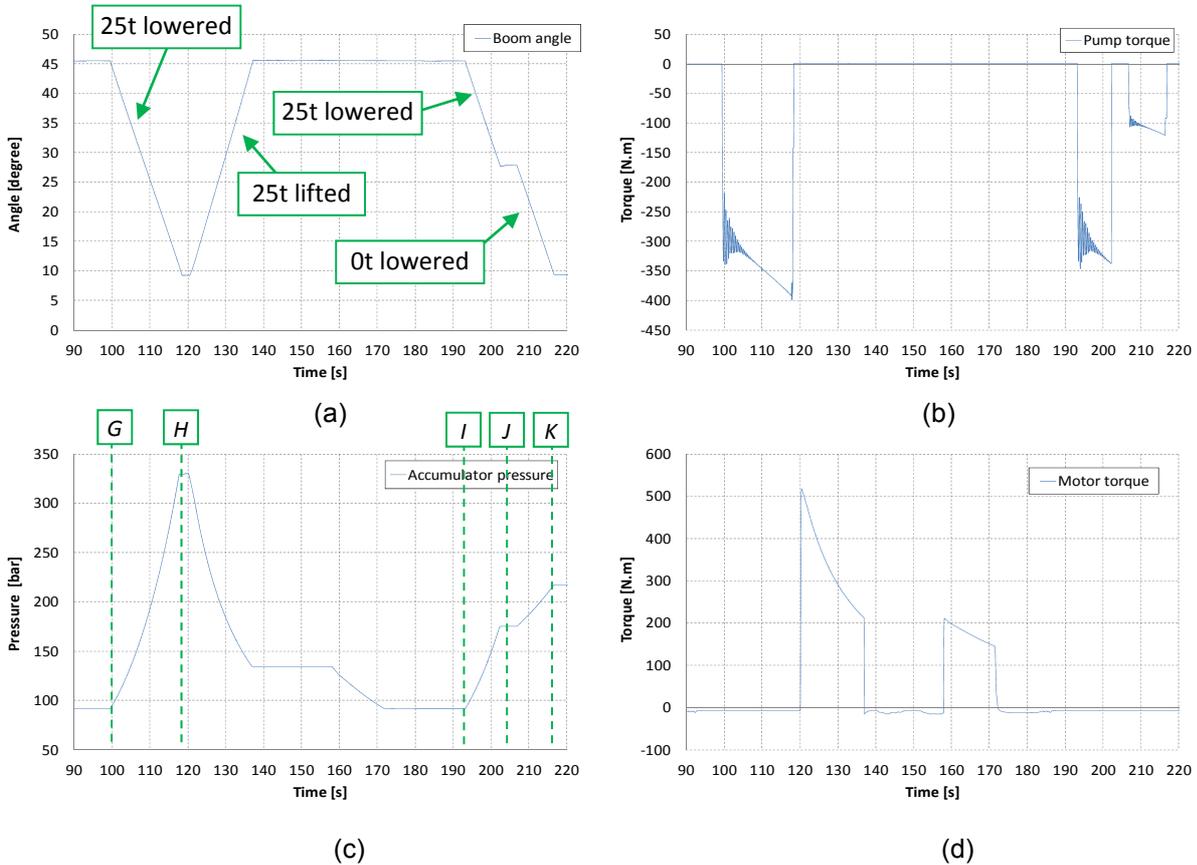


Figure 9. Solution 3: (a) Boom angle, (b) Regeneration pump torque, (c) Accumulator pressure, (d) Helping motor torque

5. CONCLUSION AND FUTURE WORK

In this paper three systems dedicated to potential energy recovery have been modelled and simulated. Each solution was implemented on the physical model of a container handling machine whose high potential for energy recovery is suitable for hybridization.

The first architecture based on a meter-out valve demonstrated on average 30% of energy recuperated compared to the total recoverable energy and 8% fuel savings. In spite of poor energy performances, this solution requires few additional components and the control system remains relatively simple. The solution with a pressure converter coupled to the ICE showed better performances in terms of energy recuperation but the fuel savings are similar to the previous architecture. This is mainly due to the friction losses generated by the additional motor and pump continuously driven by the engine. Moreover the extra pump for energy storing induces a more complex system and a higher cost of the system whereas the fuel savings are low. The last studied solution is based on the decoupling of the regeneration system from the ICE. This architecture achieves on average 65% of recovered energy and the consumption reduction can reach 16%. The great advantage is the possibility to recover energy regardless of the container weight meaning that the fuel savings are more independent from the duty cycle.

In the near future strategies concerning power management and more particularly the energy reuse will be studied deeper to improve the entire system. Moreover the most promising solution with the independent transformer based recuperation system will be implemented on a machine in order to validate the simulation results. The dynamics of a boom lowering with energy recuperation will also be compared with the traditional system to confirm the promising performances of the hybrid machine.

NOMENCLATURE

A_i	Flow area of the valve (i)	[m ²]
$A_{max\ i}$	Maximum flow area of the valve (i)	[m ²]
C_{qCV}	Check valve discharge coefficient	[-]
C_{qi}	Discharge coefficient of the valve (i)	[-]
J_{ICE}	Total inertia on the engine shaft	[kg/m ²]
J_7	Inertia of the recuperation motor (7)	[kg/m ²]
J_8	Inertia of the recuperation pump (8)	[kg/m ²]
k_L	Joystick position	[-]
$\eta_{m,rp10}$	Mechanical efficiency recuperation pump (10) system 2	[-]
$\eta_{m,rp8}$	Mechanical efficiency recuperation pump (8) system 3	[-]
$\eta_{m,m8}$	Mechanical efficiency recuperation motor (8) system 2	[-]
p_{Ai}	Pressure at port A of the component (i)	[bar]
p_{Bi}	Pressure at port B of the component (i)	[bar]
p_T	Tank pressure	[bar]
ρ	Hydraulic fluid density	[kg/m ³]
$p_{accu\ i}$	Pressure of the accumulator (i)	[bar]
$Q_{accu\ i}$	Flow rate at the inlet port of the accumulator (i)	[L/min]
Q_{max}	Maximum lifting cylinders output flow rate	[L/min]
T_{acce}	Accessories torque	[N. m]
T_{ICE}	Engine torque	[N. m]
T_{mi}	Torque of the motor (i)	[N. m]
T_{pi}	Torque of the pump(s) (i)	[N. m]
T_{rpi}	Torque of the recovery pump (i)	[N. m]
u_3	Valve (3) input signal (system 3)	[-]
u_6	Valve (6) input signal (system 1)	[-]
u_7	Valve (7) input signal (system 1)	[-]
u_8	Percentage of pump (10) displacement system 2	[-]

u_{10}	Percentage of motor (8) displacement system 3	[-]
V_{emi}	Displacement of the motor (i)	[cm ³ /rev]
V_{erpi}	Displacement of the recovery pump (i)	[cm ³ /rev]
ω_{ICE}	Engine shaft rotation speed	[rad/s]
ω	Shaft rotation speed of system 3	[rad/s]

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