Energy Saving Enhancement through 3 way Downstream Compensator Main Valve, a Viable Option for Efficient Off Highway Applications

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Abstract

In the last decade, the high percentage of energy wasted on the control of movements by pressure throttling on mobile machines has been the subject of increasingly detailed studies, which drove to the proposal of revolutionary hydraulic architectures as Valveless and Zonal Hydraulics. Despite of higher efficiency, those proposals found poor application by OEMs, due to their higher complexity and cost, and to mind-set change required to all people involved, from design to aftersales.
As an intermediate solution, Walvoil has developed the evolution of a LS Flow-Sharing Main Valve, introducing a 3-way compensator downstream of the spool meter-out notches, in which the third way opens an energy recovery line, with an enthalpy flow available for storage or conversion into electricity.
This work presents the results of the experimental comparison of the energy consumption measured on repeated cycles of lifting and lowering loads carried out with two different architectures (standard LS flow-sharing valve and evolved LS flow-sharing valve with downstream 3-way compensator) applied to a front-loader with double actuation on boom and tilt. Both configurations were tested simulating the application to a hybrid vehicle, where the main power is supplied by a test bench (in place of the prime mover), while a battery pack combined with an asynchronous motor equipped with a gear pump acts as both complementary energy supplier and energy recovery line and converter. In order to properly compare the efficiency, the same number of cycles is performed, computing the whole energy consumption given by both the prime mover and the battery (monitoring its "State of Charge").
Electrical, hydraulic and control parameters were monitored throughout the duration of the tests, to evaluate the efficiency of the sub-components too.

Keywords. Energy saving, Energy recovery, Downstream compensation, Hybrid vehicles.

1. INTRODUCTION

In recent years, the high fraction of energy wasted on the control of movements by pressure throttling on earthmoving and agricultural machines has attracted growing attention and has been the subject of increasingly detailed studies, in order to investigate the energy leakage from prime mover shaft to final users, throughout the whole hydraulic system.

If pumps account generally for 15 to 31% of losses (on excavators, e.g.), the directional flow control valve (DFCV) has been proven to be the main culprit of the energy waste, with percentages ranging from 40 to 62% of the total energy that the prime mover makes available at the shaft [1, 2, 3, 4, 5].

The global trend of reducing the consumption of fossil fuels and their conversion into harmful emissions has also forced the Fluid Power sector to devise innovative solutions to perform the same tasks requiring less energy to the prime mover ("energy saving"), and perhaps even succeeding to recover from inertial or overrunning loads, or through less dissipative controls of the movement ("energy recovery") [18, 32].

Several applications have been studied in order to evaluate where to recover energy on machines [6, 7, 8], how and where to store or use it [9, 10], and how to integrate the Energy Recovery System within the hydraulic circuit [8, 11, 12].

To avoid energy losses due to the compensation of the different resistant loads applied to users simultaneously activated and powered by the same source, the logically most promising method is to make the flow rate supplied to coincide, for each movement, with that one strictly necessary to reach the speed of required actuation, with the minimum pressure sufficient to overcome the load opposing the movement.

This finding led to the proposal of very efficient architectures (attributable to the concepts of "Valveless Hydraulics" [13, 14] and "Zonal Hydraulics" [15, 16, 17]), but quite more complicated and expensive than those currently consolidated on the market, thus finding poor application by OEMs to date.

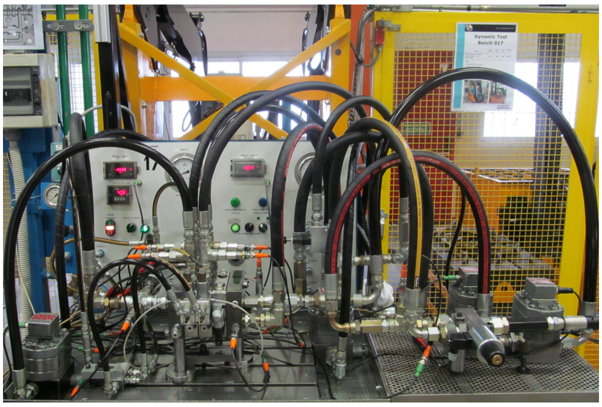
Despite of difficulties that these architectures implied, due to significant circuit changes at the level of design, production, after-sale service, and due to benefits to communicate to final users in order to justify the vehicle's price remarkable increase, both academic and industrial players have been further investigating an holistic approach to optimize the overall energy efficiency of the vehicle, including prime mover, transmission and users [4, 19, 20, 21].

Consistent efforts have also been done to develop innovative components, potentially useful for energy recovery and storage or direct drive of single user, as digital accumulators [22], hydraulic transformers [23, 24], special displacement controlled implements, close-circuit electro-hydraulic actuators [25, 26], or dedicated lines for energy recovery [33, 34].

Among these efforts, Walvoil (together with National Research Council of Italy and University of Naples Federico II) is developing a Load Sensing Flow Sharing (LSFS) DFCV with new circuit solutions: a 3-way compensator downstream of the spool meter-out notch, in which the third way opens an energy recovery line (references [27, 28, 29, 30, 31 and especially 37], that details evolution from existing LSFS DFCV).

On this line, an enthalpy flow is available for storage in a hydraulic accumulator or for conversion into electricity (useful in coming hybrid vehicles).

2. Aim of Present Work



Figures 2.1, 2.2. Test rig configuration and hydraulics *set-up*

In consideration of worldwide increasing diffusion of hybrid on-road vehicles, also off-road mobile machines OEMs are manufacturing prototypes of hybrid vehicles and engines ([35, 36], e.g. Caterpillar 336E excavator, Carraro SRX Hybrid tractor, Kohler K-HEM hybrid engine, etc.). This opens new chances on energy management of vehicles, due to the availability on the vehicle of devices for electricity conversion, management and storage (motor, inverter, batteries, etc.). By convoying the energy flow recovered from the DFCV "third line" to a hydraulic volumetric device (hydraulic pump/motor) connected to an electric motor, it is possible both to convert recovered oil enthalpy into electricity to be stored into batteries and to assist the motor that can compress oil via the volumetric device. This device can work in parallel to the main pump installed on the prime mover in order to generate the required flow, so creating a hybrid vehicle.

This work presents the results of the experimental comparison of the energy consumption measured on repeated cycles of lifting and lowering loads carried out with two different circuit solutions applied to a front loader, connected to an experimental test bench available at Walvoil Test Department.

The two hydraulic solutions are:

- standard load sensing flow sharing DFCV (here following called DPX);
- evolved load sensing flow sharing DFCV with downstream 3-way compensator (here following called EPX).

The application consists in the lifting, lowering and tilting of a load by double simultaneous actuation of front loader boom and bucket cylinders (with the first having differential areas with ratio of 1,39, while the second is a double rod cylinder).
The purpose is to physically simulate a hybrid vehicle with a prime mover giving about 85% of energy required to accomplish the task, while the remaining 15% is given by an electric booster, a motor coupled with a volumetric hydraulic device used to convert oil internal energy into electricity and vice versa.

3. SIMPLIFICATIONS INTRODUCED IN THE TEST AND METHODS OF COMPARISON AND ANALYSIS

3.1. Prime Mover

On the test rig it has been substituted by a 37 kW asynchronous motor, running at constant speed of 1.450 rpm and equipped with a 71 cm^3 variable displacement Load Sensing (LS) axial piston pump.

3.2. Electrical Hybrid Booster

Electrical power is supplied by a 48 V Li-MH 16 kWh battery pack, combined with a 12 kW asynchronous motor, coupled to a Group 2 11,0 cm^3 external gear pump. Its speed is controlled by a programmable inverter that communicates with an external ECU via CAN protocol (a Walvoil CED400X programmed with proprietary Integrated Development Environment “PHC Studio”). It can work as both “electrical motor + hydraulic pump” and “hydraulic motor + electrical generator”.

3.3. Load

The front loader has been loaded with a steel basket containing seven “200kg-weights”, for a total payload of 1.750 kg, plus the weight of the loader itself.

3.4. Cycle

A general cycle has been chosen, in order to have combinations of simultaneous resistant and overrunning loads: a first “boom-up + bucket-down” phase was followed by a “boom-down + bucket-up” second phase.

3.5. Actuation and Control

The spools of both sections were automatically activated by the same ECU that communicates with the inverter in order to set the electrical booster motor angular speed. Both the positions of the spools and the angular speed were set-up in “open-loop” mode, with values chosen after careful tuning.

3.6. Repetition and Duration

In each configuration the cycle was automatically repeated 10 times under control of the same ECU, setting same flow rates, spool positions and motor's revolution per minute, and checking time and cylinder position. Repeatability in different configurations was within ±0,3 s out of the 368,6 s of time required to run the 10 cycles.

3.7. Hydraulic Main Valve

A provisional casting has been used, in order to be able to switch rapidly from DPX to EPX configuration and vice versa; the internal circuits are not optimized in neither DPX, nor EPX configurations, due to the different castings that would have been necessary. The two different circuits are realized by changing spools, compensators, notches, etc. Due to prototype internal constraints, it was chosen to do not include regeneration in DPX mode, meanwhile it was in EPX mode. Anyway, pressure levels re-entering from cylinders are lower in DPX than EPX, so implying higher benefits from regeneration for this last one configuration. Since this is due to test set-up and not to state-of-the-art limits, its energetic contribution will be carefully evaluated in comparison and conclusions.

3.8. Energy Consumption

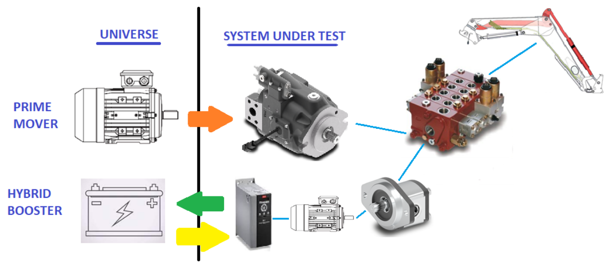


Figure 3.8.1. System border and energy exchange with environment

The "border" of the system (Figure 3.8.1) on which to evaluate the consumed energy was placed at the prime mover shaft and at the cables connecting the inverter to the battery pack of the electrical booster. The prime mover pump total efficiency $\eta_{T,p}$ was characterized on a dedicated test bench, while the delivered flow rate (QP, in liter per minute) and pressure (pP, in bar) were measured throughout the test. The power absorbed at the prime mover shaft (W_{PM}, in Watts) was hence calculated with the following formula (where the "0,6" coefficient at denominator is due to a mix of units of measure taken from both SI and the still widespread industrial practice):

$$W_{PM} = \frac{QP \cdot pP}{0{,}6 \cdot \eta_{T,p}} \qquad (3.8.1)$$

Similarly, the power exchanged with the batteries (W_{EL} when discharging and W_R when recharging, i.e. recovering energy), was calculated through the following formulas:

$$W_{EL} = V \cdot I_d \qquad (3.8.2)$$

$$W_R = V \cdot I_c \qquad (3.8.3)$$

where V is the battery voltage (in Volt), I_d and I_c are the discharge and charge currents (in Ampere) respectively.

3.9. Datalogging and Sensors

25 physical variables (including 10 pressures, 7 flow rates, 4 strokes, 2 currents, 1 voltage and 1 angular speed) were monitored during each test, and logged with a sampling interval of 9 ms during each whole test. Several virtual sensors where then implemented upon the signals of these real sensors.

The pressure sensors used have 60 MPa full scale (FS), except the Tank line and the Stand-by differential pressure, having 6 MPa and 5,2 MPa FS respectively, and total error of ±0,3% of FS. Gear flow counters used can detect a 2 cm^3 flow rate pulse and have an accuracy of ±0,3% of the read value, up to a FS of 120 lpm.

Cylinder strokes have been measured through optical laser sensors (600 mm FS and 0,25% FS total error), while compensator strokes relied on LVDT sensors, calibrated prior the test and compensated in post-processing to a residual error of less than 0,1 mm.

Voltage was measured with a precision of ±0,02 V.

Current was measured with an accuracy of ±1% of 300 A FS.

Electronics of datalogger channels have an accuracy of 0,10% of FS.

Overall precision in datalogging is within ±1% for each physical variable, and within ±0,5% for most of them.

Hydraulic mineral oil used during the test is VG46 class; its temperature was regulated within a range of ±2 °C from the 40 °C temperature target.

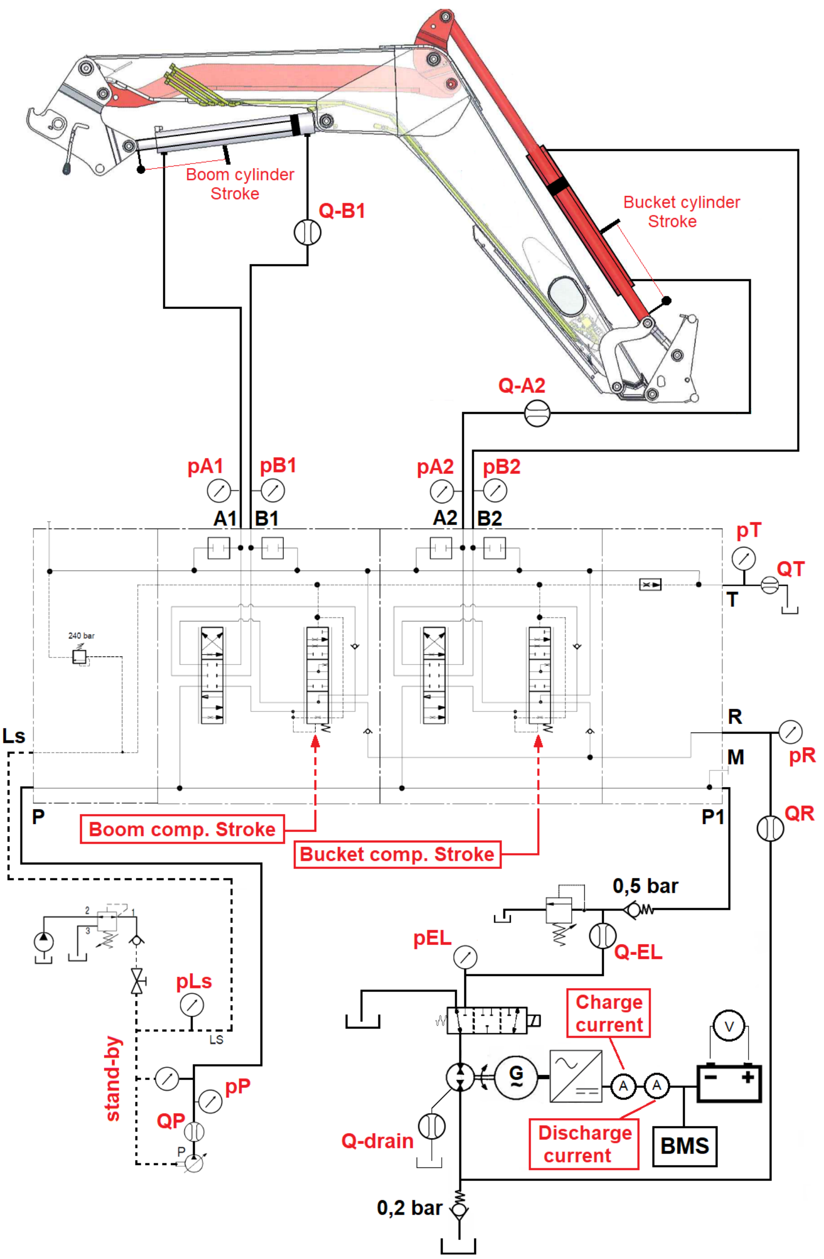


Figure 3.9.1. Hydraulic schematic and sensors position, EPX mode

4. CIRCUIT SET-UP AND TEST DEFINITION

During the set-up phase of the test, some difficulties emerged about the commutation of the hydraulic volumetric device of the electric booster from "pump" operation to "motor" operation and vice versa. To overcome these problems and other minor issues about stability of the system, the two phases in each configuration were simplified as follows.

4.1. DPX Configuration, "Boom-up + Bucket-down" Phase

Prime mover's pump delivers 32 lpm, while the electrical booster's one delivers 6 lpm in parallel. Both pumps have suction side at Tank pressure and P side at about 170 bar, due to pump margin of 20 bar and LS pressure given by boom cylinder pressure, base side, of 150 bar. Bucket compensator throttles inlet pressure to the required 20 bar to avoid cavitation, while the meter-out spool notch throttles the reentry pressure from 85 bar (due to overrunning load) to Tank pressure. Recovery line plugged.

4.2. DPX Configuration, "Boom-down + Bucket-up" Phase

Electrical booster switched off. Prime mover's pump delivers all the required 38 lpm at about 140 bar, necessary to push-up the bucket cylinder. Boom compensator throttles inlet pressure to the required 30 bar to avoid cavitation, while the meter-out spool notch throttles the reentry pressure from 120 bar (due to overrunning load) to Tank pressure. Recovery line plugged.

4.3. EPX Configuration, "Boom-up + Bucket-down" Phase

Recovery line opened to electrical booster pump suction side; regenerative line through bleed-off hole opened inside the bucket spool. The Recovery line takes energy from the bucket-down overrunning load, rising suction pressure side at 110 bar, so the current required from the electrical booster decreases from the 70 A necessary in DPX configuration to the present 25 A, in order to accomplish the same task of delivering 6 lpm at 170 bar. Due to regeneration inside bucket spool, prime mover's pump has to deliver just 26 lpm, instead of the 32 lpm of DPX configuration.

4.4. EPX Configuration, "Boom-down + Bucket-up" Phase

Recovery line opened to electrical booster pump suction side; regenerative line through bleed-off hole opened inside the boom spool. The Recovery line takes energy from the boom-down overrunning load, collecting 24 lpm flow rate at an average pressure of 140 bar during the movement; flow diverted to Tank after generating an average current of 70 A in the hybrid device. Due to regeneration inside boom spool, prime mover's pump has to deliver just 29 lpm, instead of the 38 lpm of DPX configuration.

5. Test Results

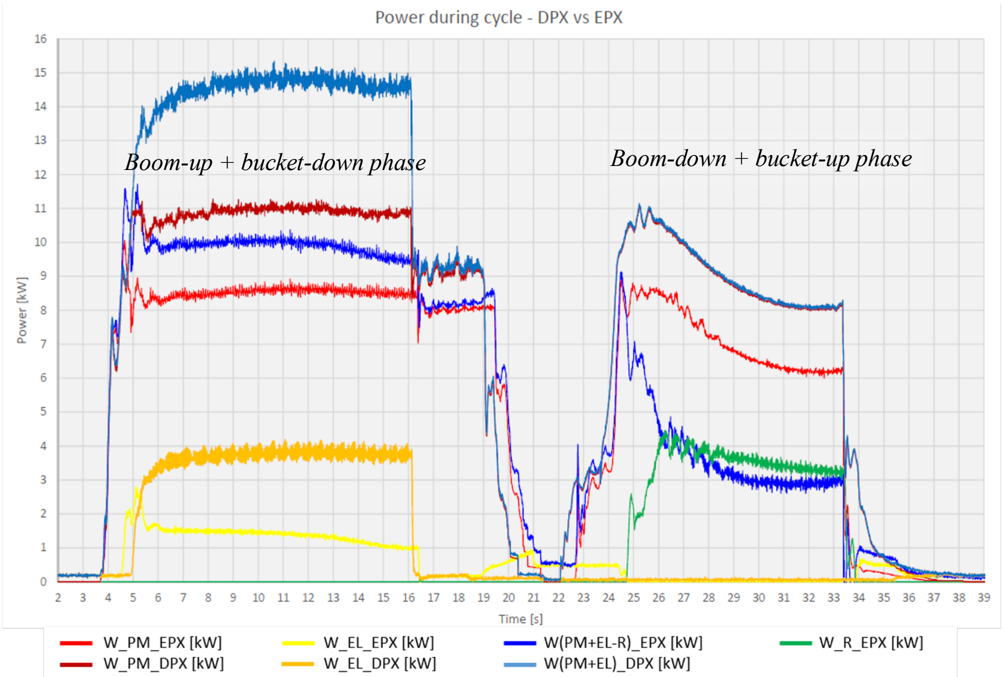


Figure 5.1. Power demanded and produced in both DPX and EPX configurations during a cycle

As depicted in Figure 5.1, the power demanded by the prime mover during "boom-up + bucket-down" phase (I) decreased from 11 kW to 8,5 kW passing from DPX to EPX mode (dark and bright red lines respectively). Similarly, the power demanded by the hybrid booster decreased from 4 to 1,5 kW (golden and yellow lines respectively).
The total power demanded was hence reduced from almost 15 kW in DPX configuration to about 10 kW with EPX (dark and bright blue lines respectively).
During the "boom-down + bucket-up" phase (II) the electrical hybrid unit was switched off in DPX mode, while it was used to generate current to be stored into batteries (without supporting the inlet flow to the main valve) in EPX mode. So in this phase prime mover power required by DPX coincides with total required power, at an average value of almost 9 kW (the value is not stable during this phase due to the kinematics of the bucket during cylinder extension). Meanwhile in EPX mode the prime mover delivers an average power of about 7 kW, but (after an initial transient) almost 4 kW are generated, thanks to the recovery line, and sent back to the battery (green line), so halving the net energy demand during this phase, to an average value of about 3,5 kW.

6. ENERGY CONSUMPTION COMPARISON

By integrating over time the power values previously calculated, it was possible to evaluate the total energy consumed to carry out the planned ten-cycle-test in both cases of DPX and EPX configurations.

As depicted in the pie charts of Figure 6.1 and in the histogram of Figure 6.2, the ten cycles carried out in DPX configuration needed a total amount of 2.988 kJ given by both the prime mover (2.535 kJ, 85%) and the hybrid electrical booster (453 kJ, 15%).

To accomplish the same task with EPX configuration just 2.277 kJ were required to prime mover (2.058 kJ, 90%) and hybrid electrical booster (219 kJ, 10%), but during phase II of each cycle the hybrid system had actually charged the battery through conversion into electricity of the fluid power flowing in the recovery line. A total amount of 302 kJ was stored back into the battery, bringing down to 1.975 kJ the net value of energy used, with battery receiving back in charging phase II 83 kJ of energy more than the amount supplied in discharging phase I.

By comparing DPX and EPX working modes and respective energy flows, it is possible to analyse where and how the energy consumption was reduced switching from DPX to EPX circuit.

About 478 kJ (16% of what required by DPX configuration) were saved by asking lower flow rate at the prime mover's pump, thanks to the bleed-off holes giving flow regeneration through bucket-down movement in phase I and through boom-down movement in phase II.

A further 233 kJ (8% of the total) were saved by using EPX R-line to overcharge the hybrid booster's pump suction side, thanks to bucket-down movement during phase I.

Moreover, there are the 302 kJ (10% of the DPX mode total) recovered and sent back to the battery, by using the EPX R-line to supply the hybrid unit in order to convert it into an electrical generator, thanks to boom-down movement during phase II.

With EPX mode was saved 24% of energy used in DPX mode, and further 10% was recovered.

Regarding the net value of energy used, so including the recovered energy within the balance, it's possible to evaluate in Figure 6.2 as EPX cycle ran using only 66% of energy used by DPX configuration.

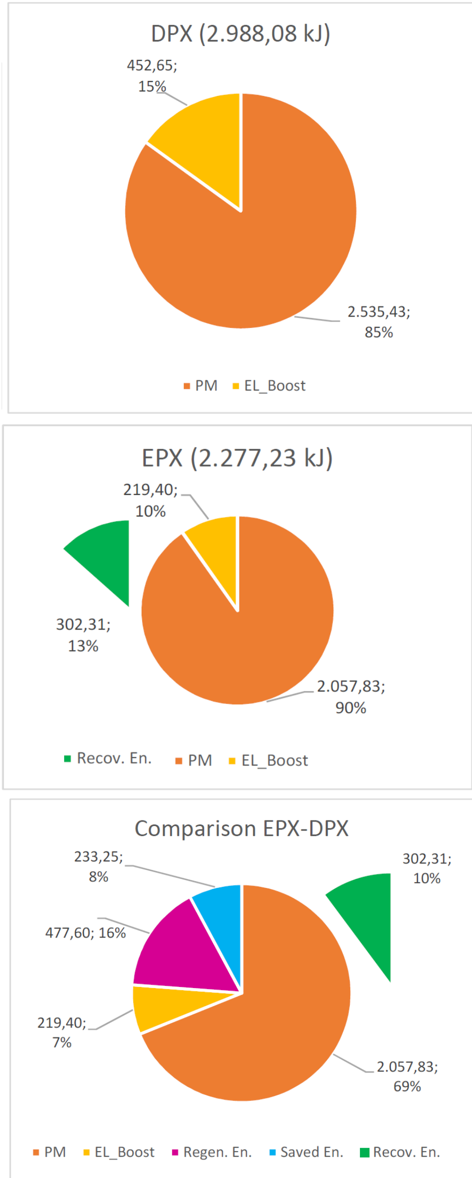


Figure 6.1. Total energy exchanged during the 10-cycle-test

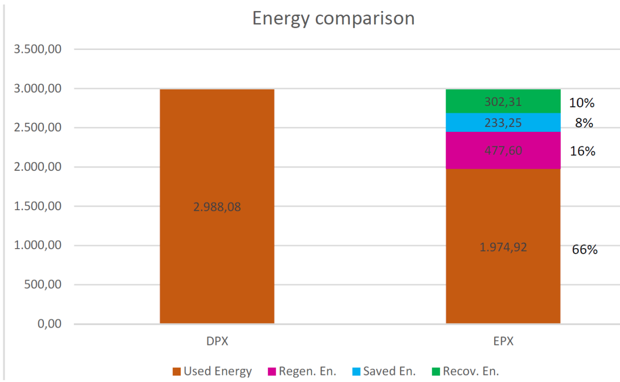


Figure 6.2. Comparison of net values of total energy used carrying-out the 10-cycle-test in DPX and EPX modes

7. Conclusions, Lessons Learnt, Next Steps

Overall savings of 34% went beyond expectations. Even if almost half of the total came from regeneration inside spools (that wasn't possible to use in DPX mode with this provisional casting), the result of 18% reached by savings and recovery through the R-line is quite good.

Introducing bleed-off and regeneration in DPX mode would similarly decrease the amount of energy required by this configuration, and this would bring the energy saved and recovered through R-line in EPX mode at a very satisfactory percentage value ranging around 20-21%.

Wider total savings are expected in different applications, eventually ranging up to 35% in forklift and cranes, while complex circuits as those of excavators and backhoe loaders have to be carefully investigated through simulation and testing, in order to fully exploit their potential.

It is correct to point out that even if this experiment benefited from presence of overrunning loads in each phase, EPX mode can generate a flow in R-line also exploiting the pressure compensations happening during multiple sections actuation in LS DFCV.

Beside satisfactory results regarding the amount of energy saved and recovered, several further goals were achieved carrying out the test, sketching the path for the following steps of the research activity.

The prototype external gear pump emerged to be critical in phases with rapid switch between the conditions of working as pump and as motor: with suction and delivery sides having similar pressure values, the drain flow rate can increase significantly, jeopardising the energy efficiency. Different volumetric device typologies are to be tested.

Further efficiency benefits can arrive from the introduction of closed loop control strategies: models with feed-back on compensator positions and recovery line pressure are under development.

Further benefits from EPX introduction can come from prime mover downsizing: after hybridization premises, with the 15 kW maximum demand split to 11 kW from prime mover and 4 kW from hybrid system, EPX reduced the required amount to 10 kW. Last but not least, lower use of oil throttling to control movements can drive to lower oil aging and downsizing of oil cooling system, with further indirect benefits in energy saving.

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