Downstream Compensator: Innovative Systems, Modeling Analysis and Energy Optimization

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Abstract. The Energy optimization is becoming fundamental in the Fluid Power world. University and industries are working hard to promote innovative and efficient ideas to optimize components that are the main cause of energy dissipation of ICE and recently Electric Off-Road vehicles. A new hydraulic layout based on the concept of “Downstream compensation” is introduced and then validated using real test data. Three architectures of this innovative Directional Control Valve are presented in this paper. The first idea of layout includes a compensator controlled by two pressure signals taken before and after the main spool of the hydraulic circuit. Thanks to its controlled stroke, this compensator diverts to a high-pressure accumulator part of flow that otherwise would be delivered to the tank. Moreover, two different layouts able to satisfy the Flow Sharing characteristic were developed. In any configuration, the compensator, thanks to its downstream position, allows to control the return flow, realizing a remarkable energy recovery from the overrunning loads and the simultaneous use of multiple actuators at different pressure levels.

For all the analyzed hydraulic circuit, lumped parameter models were realized, using a commercial software. These models, validated with experimental tests, have allowed to calculate the energy recovery achieved by the system. Moreover, an optimization of the most important system's parameters and components were realized to improve the system efficiency. In every tested configuration, this compensator ensures great advantages for both the energy recovery and the economic point of view. Finally, an outlook is drawn of the reuse of recovered flow through the application of an electro-hydraulic motor.

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Introduction

In the recent years, in the world of Off-Highway vehicles, different ideas have come out to improve efficiency of the hydraulic systems acting on the energy saving and recovery [1].

Particularly, as consequence of the limitations on the pollutant emissions fixed by the stringent regulations, the industries are focusing the attention on the electrification of machines driveline. However, to date, all energy saving opportunities have not been fully exploited. In fact, a lot of ideas, developed in these years, only imply the replacement of the ICE engine with the electric motor, leaving all the hydraulic circuits unchanged. Nevertheless, interesting ideas are coming from literature acting on the hydraulic system to improve the overall vehicles efficiency.

Naturally, it is necessary to understand that the optimization of a system passes through the improvement of all the components that are the main causes of flow dissipation. As an example, energy recovery is rarely applied in the traditional Directional Control Valves, in which about 60% of power losses of the hydraulic circuit are located [2-5]. The main causes of these losses are the centralized system, that provides the distribution of the flow from a single pump to multiple users, and the speed control of inertial loads. However, for the same reasons, the directional control valve is the hydraulic component with the higher energy recovery opportunities. In fact, an important energy recovery can be obtained through the control of overrunning loads and the compensation in simultaneous movements.

In this paper, a new solution of Directional Control Valve, based on the Downstream Compensation concept, is presented [6-8]. Thanks to its particular configuration, this innovative compensator is able to control the return flow of the circuit, deflecting it to an accumulator for the recovery. The evolution of three different layouts and the energy advantages obtained by each architecture are compared using numerical technique.

In conclusion, there is an outlook about the possibility to reuse this energy through an electro-hydraulic motor.

1 System architectures

The Three Way Downstream Compensation concept was born with the goal of realizing a single device able to make a hydraulic recovery from both the inertial loads and the local compensation.

The idea of a 3 ways compensator comes out from works already presented by the authors in recent studies [6]; where the system was limited to realize internal regenerations, making impossible the energy storage from the overrunning loads and local compensations. For this reason, a downstream compensation system was realized through the introduction of a new 3 ways compensator architecture [7]. The scheme, illustrated in Fig.1, shows a centralized system in which a Load Sensing pump (hereinafter called LS pump) supplies two sections. Every single section consists of a directional control valve that manages the flow Q_{IN} , directed to the actuators, and a 3 ways compensator that controls the flow Q_{OUT} on the return line for the energy recovery phase. In fact, each compensator is connected to the recovery line R directed to an accumulator.

The compensator is called “3 ways” as it has the inlet port P (connected to the T port of the main spool) and other two outlet ports that are the port R, directed to the recovery line, and port T, directed to the tank.

Moreover, the compensator is controlled by two pressure signals that act on the same area A_{PIL} : a signal on the spring side (p_{COMP}) and one on the opposite side (p_{OUT}), respectively taken after and before the main spool discharge orifice.

Fig. 1. First hydraulic layout (local compensation)

The pressure drop on the proportional meter out notch is regulated by the following equation:

$$p_{COMP} \cdot A_{PIL} + F_{SPRING} = p_{OUT} \cdot A_{PIL} \quad (1)$$

And the flow Q_{OUT} is expressed by relation below:

$$Q_{OUT} \propto A_{OUT} \cdot \sqrt{p_{OUT} - p_{COMP}} \quad (2)$$

Thanks to its position, the compensator realizes a constant pressure drop through the outlet notch of the main spool making sure that the flow (and then the velocity of the actuator) is independent from the load pressure and depends only on the opening outlet notch of the main spool.

Normally, the compensator spool rest position has both ways (A_R and A_T) open; when it begins to throttle, it firstly closes the tank line (T) and then the recovery line (R).

A second system layout was designed to provide the Flow Sharing feature [8]: the new scheme, represented in Fig.2, is similar to the original one, but some changes were applied on the compensator pilot system. The pressure drop on the meter out notch of the main spool [$p_{OUT} - p_{COMP}$] is not anymore defined by the spring force F_{SPRING} but rather by the differential force generated by the pilot pressures [$p_P - p_{LS}$].

The pressure drop on the proportional meter out notch is regulated by the following equation:

$$[p_{OUT} - p_{COMP}] \cdot A_{PIL} = [p_P - p_{LS}] \cdot A_{PIL} \quad (3)$$

The 4 pressure signals system, as in the previous version, was studied experimentally and numerically [8]. Moreover, if compared to the 2 signals compensator, this system is able to proportionally reduce the flow to all the actuators and then their velocity, satisfying all the users.

Fig. 2. Second hydraulic layout (local compensation)

The results guarantee energy recovery both from overrunning loads and local compensation. The latest innovation of this idea was the development of an architecture in which the compensator is controlled by the delivery line rather than the return line, as in the previous layouts.

Fig. 3. Third hydraulic layout (local compensation)

Fig. 3 shows the system with a double sections circuit for the compensation test. The operating principle of this system is similar to the previous ones: a centralized system in which the LS pump is connected to each spool valve. The two main spools are Load Sensing and through a shuttle valve, the LS signal pressure is connected to the pump. The outlet of every spool, as in the previous cases, is connected to the P port of corresponding compensator.

The compensator, depending on the work condition, can deliver the flow directly to the tank or to the accumulator while the T port of the compensator is closed.

The substantial difference between the analyzed layouts consists on the compensator pilot system. This aspect is clearly described in Fig.4 which an image referred to the numerical model built up using the commercial software Simcenter AMESim©.

In Fig.4 the same layout of Fig.3 is displayed but, in this case, a single section is tested in overrunning load configuration.

The previous architectures (Fig.1 and Fig.2), presented in recent studies [7,8] were a fundamental starting point to develop the last layout displayed in Fig.3 in local compensation and in Fig.4 in overrunning load configuration. On this last improved architecture, the experimental and model results of the paper are focused (chapter 2).

In Fig.4 (as in Fig.3) the compensator pilot system consists of two pressure signals taken from the delivery line: the local pressure LS of the corresponding main spool (p_{LS}), acting on the spring side and the LS pressure of the pump (p_{LS_PUMP}), acting on the opposite side.

In this evolution of the system, the pressure drop on the meter in notch is regulated by the compensator through the following equation:

$$p_{LS} \cdot A_{PIL} = p_{LS_PUMP} \cdot A_{PIL}$$

(4)

Every compensator works on the return line, aiming to copy the p_{LS_PUMP} pressure in the local p_{LS} line.

This configuration is able to recover energy both from the compensation and the overrunning loads, while guaranteeing a precise load sensing control with flow sharing function.

Fig. 4. Third numerical model layout (overrunning load)

2 Experimental tests and numerical model results

In this paragraph, experimental tests and numerical model results of the architecture in Fig.4 are described. The layouts were tested in the lab of the company Walvoil SpA using a double effect cylinder as an actuator (Fig.5(a)) while the overrunning load was generated by an additional opposite cylinder, pressurized by a secondary pump.

Table 1. Cylinder characteristics.

Piston diameter	125 mm
Rod diameter	70 mm
Stroke	530 mm
Max pressure	400 bar
Load pressure	65 bar

Fig. 5. (a) Experimental test bench for simulation activities; (b) Table 1

In Fig.6 (a) and (b) a first interesting comparison between the experimental data (continuous lines) and the simulation outputs (dotted lines) is shown; in Fig. 6 (a) the yellow line refers to the compensator displacement: when it begins to throttle, a part of the flow goes to the accumulator (green line). Consequently, the accumulator pressure (yellow line) increases from the settled precharge pressure (32 bar) to 60 bar. As the accumulator pressure reaches a value that is not anymore allowing the correct lowering speed, the compensator goes back to its initial position and the flow is diverted to the tank.

Fig. 6. System’s numerical results: (a) accumulator pressure (yellow line); flow directed to the accumulator (green line); compensator displacement (red line); (b) pressure at the cylinder’s outlet (blue line); flow directed to the compensator (green line); cylinder displacement (pink line)

The numerical models were used to perform an interesting energy analysis, in fact, introducing energy sensors, as in Fig. 4, the energy delivered by the pump during the lowering operation and the energy recovered by the three way compensator can be compared. The energy efficiency is calculated through the following equation:

$$\eta = \frac{En_{recov}}{En_{spent} + En_{pot}} \quad (5)$$

where En_{recov} refers to the recovered energy in the accumulator, En_{spent} is the energy delivered by the main pump during the lowering operation and En_{pot} refers to the potential energy of the load in its starting position.

The energy results of the first two configurations were already shown in the papers [7,8] while in this work the focus is on the energy recovered through the last presented layout.

Fig.7 shows the results from the energy point of view; the red line refers to the sum of energy supplied by the main pump to the system and the potential energy in the lowering cylinder while the green line refers to the recovered energy into the accumulator. The system can achieve an energy recovery of 21% as shown in the diagram in Fig. 7 (b).

Fig. 7. (a) En. available vs En. recovered; (b) percentage results (%)

These results exclusively refer to this specific test with definite parameters as the volume and the precharge pressure of the accumulator and the load condition, expressed in Fig. 6 (a) and (b). The variation of these parameters could change the system's energy efficiency.

For example, it is important to consider that the test bench layout is not optimized, therefore significant pressure drops due to the several hoses and fittings effect on the result.

For this reason, the system layout was optimized removing all the main throttles of the system in Fig.4: The recovered energy showed an increase of 5 %, reaching a final value and a consequent overall improvement of almost 26 % (yellow line).

Analyzing the yellow line in Fig.7(a) it is clear that in correspondence of 20 s the accumulator was completely full (in fact the recovered energy takes on a constant trend after this point) and the still recoverable flow is wasted to the tank. This behavior is also clear analyzing Fig.8 in which all the flow coming out from the compensator goes to the accumulator (red line) and the accumulator pressure (yellow line) grows from the precharge value (32 bar) to 64 bar

and when the accumulator is full (in correspondence of 20 s) its pressure becomes constant and the flow is entirely diverted to the tank.

Fig. 8. Optimized system’s numerical results (6L accumulator): flow directed to the accumulator (red line); flow directed to the tank (blue line); accumulator pressure (yellow line)

This trend of the flow is influenced by the Δp between the load pressure of the test (65 bar) and the precharge pressure of the accumulator (32 bar). With a volume of 6L, the accumulator pressure reaches 64 bar in 8.5 s and from this point on, all the flow that failed to come in the accumulator, goes to the tank. With a bigger volume accumulator, the time employed from its internal pressure to increase the value from 32 bar to 64 bar is bigger than a 6L one and then more flow manages to enter in the accumulator.

For this reason, in a secondary analysis, a further optimization of the system was realized through the simulation software. In the optimized layout (without throttles) was replaced the existing 6L accumulator firstly with a 8L one and secondary with a 10L one. Results are shown in Fig.9; Fig. 9 (a) is referred to the first case and the flow keeps moving to the accumulator until 25 s with a small amount of flow lost to the tank.

Fig. 9. Optimized system’s numerical results (a) (8L accumulator); (b) (10L accumulator)

In Fig.9 (b), the trends are referred to the case with a 10 L accumulator; the flow coming out from the compensator goes entirely to the accumulator that has not completely filled until the end of the test with a consequent zero flow to the tank. Therefore, a system with a bigger and correctly sized accumulator can increase the energy recovery.

Accordingly, comparing the energy trends of these different cases (Fig. 10) emerges that the greatest energy recovery is obtained through the 10 L accumulator system (pink line). The graph in Fig. 10 (b) shows the percentage of recovered energy that thanks to this last optimized system reaches the 42 % of the available energy (spent energy + potential energy).

Fig. 10. (a) Energy comparison between different accumulator volumes; (b) percentage results (%)

Different techniques are developing to reuse the recovered potential energy. In the paper [9] a hydraulic lifting application was studied, both experimentally and numerically, that through an inverter-controlled electric motor connected with a reversible gear pump/motor is able to recover potential energy during the lowering operation. The use of this type of electric motor in one of the previous layouts for the energy recovery is one of the next objectives of this work.

Conclusions and future developments

This paper is focused on the comparison between three types of alternative architectures, all containing a three-way compensator controlled by different pressure signals. In particular, the objective of this work is the energy analysis of the third layout and the study of different solutions to optimize this architecture reducing energy losses and growing the energy efficiency.

This system archives an increase of the machine energy efficiency of 42%. Next developments will consist in the realization of a system in which combining the three way directional valve layout and the electric motor application higher energy efficiency will be achieved from the hydraulic system.

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Nomenclature

A_{OUT} Main spool metering in area

A_{PIL}	Compensator piloting area
A_R	Compensator way to recovery line
A_T	Compensator way to tank line
En_{pot}	Potential energy from the overrunning load
En_{recov}	Recovered energy
En_{recov_optim}	Recovered energy in the optimized system without the main throttles
En_{spent}	Delivered energy by the main pump
F_{SPRING}	Compensator spring force
p_{LS}	Local Load Sensing pressure
p_{LS_PUMP}	Load Sensing pump pressure
p_P	Pump pressure
p_{COMP}	Pressure between main spool and compensator
p_{OUT}	Pressure on actuator meter out line
Q_{IN}	Flow to the user (meter in flow)
Q_{OUT}	Flow from the user (meter out flow)
x_{COMP}	Compensator stroke

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