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Optimal design of a hybrid hydromechanical transmission for a reach stacker

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Abstract. The hydraulic hybridization of the hydromechanical transmission is an interesting solution to reduce fuel consumption in heavy duty machinery, thanks to the high power peaks recovered in the braking phase and the low cost technology involved. However, hybridization must be carefully considered, as there is no optimal configuration for all applications. For this reason, the design of a hydromechanical transmission must be developed on the basis of the specific data of the vehicle and must tend to optimization. Following this concept, the optimal layout of the hybrid Output Coupled configuration for a particular vehicle application, the reach stacker, was studied in this work. The study will be carried out in two steps: first the optimal layout will be sought based on the continuous formulation of the planetary gear. Subsequently, based on the design parameters obtained, the models of the non-hybrid and hybrid transmission will be simulated for a functional and energy comparison.

INTRODUCTION

In recent years, scientific research has been following some strategies for improving efficiency and reducing emissions in the agricultural and heavy duty machinery sectors: the electrical and hydraulic hybridization [1, 2, 3], the optimal management of the hydraulic powered tools [4, 5, 6], and the adoption of innovative drivelines, such as the hydromechanical transmissions [7, 8]. The latter has been subject to many studies concerning the design [9-11] and the optimization of both components and layout [12, 13].

A further step in increasing the efficiency of the hydromechanical transmission is hydraulic hybridization, which is achieved by adding an accumulator on the high pressure line of the transmission and the appropriate control systems. The accumulator acts as a second source of energy and is supplied by the energy recovered during the vehicle braking by means of the pump of the hydrostatic group. For this reason, hydraulic hybridization is suitable for heavy vehicles subjected to work cycles with frequent starts and stops.

The hydraulic hybridization of a hydromechanical transmission was patented by Ivantysynova [14] and theoretically and experimentally studied by Kumar [15], and Kumar and Ivantysynova [16, 17], who proposed a sizing method, focusing on the problem of transmission management.

Cheong [18], comparing Input Coupled (IC) and Output Coupled (OC) hybrid power-split architectures, showed a similar fuel economy for both transmissions, with a distinction: IC configuration is advantageous to wheel torque control, while OC benefits wheel speed control.

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Many other works dealt with hybrid hydromechanical transmission and all agreed on its potential to reduce consumption [19-21].

In the design of a hydraulic hybrid powertrain, the problem is the definition of the most suitable layout among the six possible ones within the chosen configuration (IC or OC), and the size of the components, in particular the size of the hydraulic energy source, i.e. the volume of the accumulator and its precharge pressure.

Basically it is a simultaneous optimization of layouts and components.

The concept of simultaneous optimization has already been addressed by the authors [13, 22]. In [22] the authors proposed a theoretical treatment that allows to describe the layout of an IC transmission by means of a continuous variable; in this way it is possible to insert the layout within an optimization procedure like any other variable and reach the solution that identifies the most efficient layout together with the most efficient components.

In this work the same concept will be exploited and applied to the OC configuration, aimed at identifying the best layout with the functional parameters of the components, which this time also include accumulators.

The aim of the work is to highlight the energy saving potential of hydraulic hybridization for a hydromechanical transmission. However, the comparison between OC and OC hybrid is not a simple task, because the latter requires a new management criterion. Consequently, the comparison would account for two variations at the same time.

The comparison, therefore, will proceed by steps in order to separate the effects due to the regulation criterion from those due to hybridization.

The reference vehicle for the comparison is a commercial 235 kW reach stacker, which operates with frequent start-stops.

THE OUTPUT COUPLED HYBRID TRANSMISSION

The base configuration adopted for the hybridization is the classic Output Coupled (fig.1), in which a planetary gear sends the engine power via two paths: the first consists of a variable speed hydrostatic group; the second is a purely mechanical path. Finally, the two paths rejoin in an ordinary gearing before the exit.

The variation of the output speed is obtained by acting on the displacements of the two hydraulic units so as to guarantee the predetermined speed of the vehicle. As is known [7, 8], this transmission operates in three modes: 1) additive mode: the powers coming from the two branches are added together in the final gear to move the wheels; 2) full mechanical point: when the planetary ring stops, all the power passes through the mechanical branch, thus achieving the maximum efficiency condition; 3) recirculating mode: to increase the output speed, the displacement of the unit 2 is carried on the negative side and the unit 2 itself becomes a pump that transmits power to the unit 1, which becomes a motor. The planetary sums this power to that one coming from the engine.

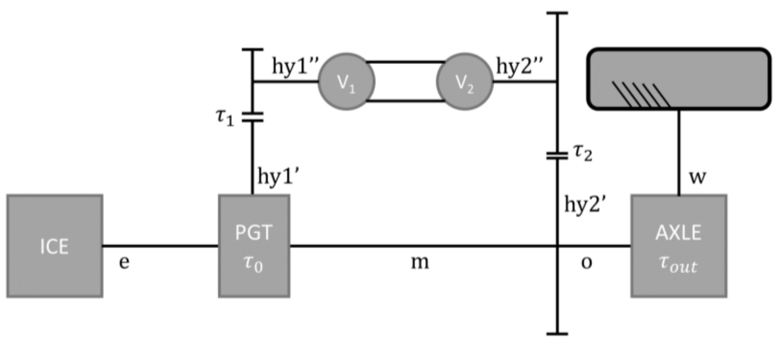

FIGURE 1. General layout of the hybrid Output Coupled hydromechanical transmission

The hybrid OC configuration is obtained from the base configuration by inserting a high pressure accumulator in the high pressure line. This accumulator receives the energy recovered during braking in this way: for any vehicle speed condition, the unit 2 reverses its displacement becoming in this way a pump that rotates in the same direction and sends oil to the line from which it was previously received. The pressure of this line will raise allowing a certain amount of oil to enter the accumulator. In the meantime, unit 1 has been set to zero displacement.

Continuous representation of a planetary gear

The velocity equation of a simple planetary gear states the velocity of one of the axes given the other two and the standing gear ratio τ_0 of the gear:

$$[-1; \quad (1-\tau_0); \quad \tau_0] \begin{Bmatrix} \omega_s \\ \omega_c \\ \omega_r \end{Bmatrix} = 0 \tag{1}$$

For example, considering the layout of a planetary gear in which the sun is connected to the engine, the carrier connected to the wheels and the ring connected to the hydraulic group, when ω_s is equal to 1, the other two speeds are provided by eq.1:

$$\omega_c = \frac{1}{1-\tau_0} \; con \; \omega_r = 0 \qquad \text{e} \qquad \omega_r = \frac{1}{\tau_0} \; con \; \omega_c = 0 \tag{2}$$

For the other five layouts of an OC transmission, which can be obtained by exchanging the connections with the engine, hydraulic group, and wheels, the expressions of the speeds are summarized in Table 1, which must be read according to the following notation

$$P_{abc} = [x; \quad y\,] = \left[\omega_b/\omega_a \;\Big|_{\omega_c=0}; \quad \omega_c/\omega_a \;\Big|_{\omega_b=0}\right] \tag{3}$$

Table 1. Coordinates of points representing six planetary gear different layouts.

	x	y
P_{scr}	[$\tau_0/(\tau_0-1)$;	τ_0]
P_{csr}	[$(\tau_0-1)/\tau_0$;	$1-\tau_0$]
P_{rsc}	[$1/\tau_0$;	$1/(1-\tau_0)$]
P_{src}	[τ_0;	$\tau_0/(\tau_0-1)$]
P_{crs}	[$1-\tau_0$;	$(\tau_0-1)/\tau_0$]
P_{rcs}	[$1/(1-\tau_0)$;	$1/\tau_0$]

As τ_0 varies between the limit values $-\infty$ and -1, the x and y coordinates of Table 1 describe a continuous curve C, which is a function of the parameter τ_0 and of the layout, as shown in fig. 2.a.

$$C \in \mathbb{R}^2, \; C = f(Layout, \tau_0), \; Layout \in [csr, \; crs, \dots, rsc], \tau_0 \in [-\infty, -1] \tag{4}$$

The curve C can be described by a polar representation, centered in the point [1, 1], through a single parameter, the angle θ (fig. 2.b):

$$C \in \mathbb{R}^2, \; C \coloneqq \{\theta; \; r(\theta), \; \theta \in [0, 2\pi]\} \tag{5}$$

With a proper inversion function f^{-1}, the standing gear ratio and the layout can be expressed as results of the angle θ:

$$[Layout, \tau_0] = f^{-1}\big(\theta; \; r(\theta)\big), \theta \in [0, 2\pi] \tag{6}$$

In this way, the layout of a transmission can be handled in the form of a continuous function of the real parameter θ, allowing to easily discuss and compare the performances of different layout and designs.

Mode details of this procedure can be found in [22].

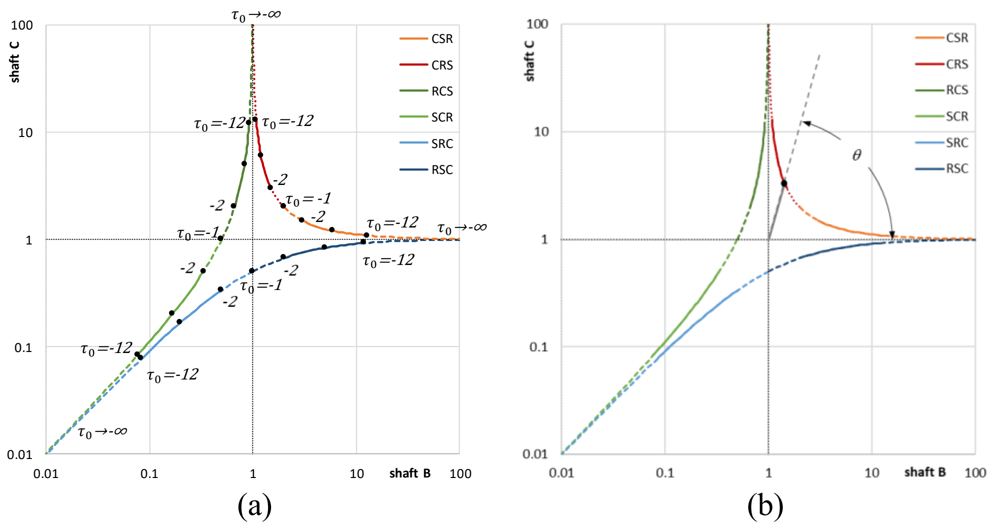

FIGURE 2. Continuous representation of planetary gear layouts: (a) functional representation for an OC configuration; (b) parametrization of curves in Fig. 2a.

Sizing procedure of the OC transmission

As the generalized layout approach will cycle all the possible orientation of the planetary gear train, there will be six different design procedures. In each one, the contribution of the planetary gear will be accounted by using the proper relation between the engine shaft, the mechanical shaft and the shaft connecting to the first hydraulic unit, i.e. the shafts *e*, *m* and *hy1'* of figure 1. Nevertheless, the same procedure will be used for all the considered layout, as follow.

First the maximum speed at the wheels' shaft is computed, assuming the maximum vehicle speed:

$$n_{w\ FMP} = \frac{v_{max}*60}{2*\pi*r_w*3.6} \tag{7}$$

Assuming the maximum vehicle speed corresponds to the full mechanical point, the mechanical shaft speed is computed using the planetary gear train speed equation knowing the speed of the other two shafts:

$$\text{Solve: } S_{PGT}(n_e, n_{hy1'}, n_m) = 0, \text{with } n_e = n_{ice} \text{ and } n_{hy1'} = 0 \tag{8}$$

The axle gear will then be sized according to the wheel and the mechanical shaft speeds ratio, as $n_m = n_o$:

$$\tau_{out} = \frac{n_{o\ FMP}}{n_{w\ FMP}} \tag{9}$$

The maximum torque at the wheels' shaft is a design constrain, that allows the calculation of the maximum torque at the output shaft:

$$T_{o\ MAX} = \frac{T_{w\ max}}{\tau_{out}} \tag{10}$$

On the other side, the maximum torque at the mechanical shaft can be obtained by the torque equilibrium of the planetary gear, given the engine torque:

$$\text{Solve: } T_{PGT}(T_e, T_m) = 0 \text{ given } T_e \tag{11}$$

The torque on the hydraulic shaft $T_{hy2'}$ can be computed given the torque on the output and on the mechanical shaft:

$$T_{hy2'} = T_{o\,MAX} - T_m \tag{12}$$

Given a first guess for the unit 2 displacement, the maximum torque can be obtained assuming the maximum pressure difference as follow:

$$T_{hy2''} = \frac{V_2\, p_{max}}{20*\pi} \tag{13}$$

The gear ratio τ_2 can be obtained as:

$$\tau_2 = \frac{T_{hy2'}}{T_{hy2''}} \tag{14}$$

Finally, the maximum allowable speed of the hydraulic unit should be verified:

$$n_{hy2''MAX} > n_{hy2'MAX}\tau_2 = n_{w\,FMP}\ \tau_{out}\ \tau_2 \tag{15}$$

If equation 9 is not verified, the procedure should be restarted from equation 9, using a larger unit.

The maximum torque for hydraulic unit 1 can be obtained from the equilibrium of the shafts e and $hy1'$:

$$T_{PGT}\left(T_e, T_{hy1'}\right) = 0 \text{ given } T_e \tag{16}$$

while the maximum speed $n_{hy1'MAX}$ is obtained from the speed correlation of the planetary gear when the vehicle is still:

$$\text{Solve } S_{PGT}\left(n_e, n_{hy1'}, n_m\right) = 0, \text{with } n_e = n_{ICE} \text{ and } n_m = 0 \tag{17}$$

The sizing of the hydraulic unit 1 follows than the same steps highlighted for unit 2 in equations 13, 14 and 15, using $T_{hy1"}$ and $n_{hy1"MAX}$ instead of $T_{hy2"}$ and $n_{hy2"MAX}$.

Once the displacement of unit 1 is defined, the continuity equation of the hydraulic path has to be checked, verifying that in every working condition the flow rate required by unit 2 can be processed by unit 1.

Sizing the accumulator

The accumulator has been sized following the method suggested by Kumar [23], whose goal is to calculate the minimum volume of the accumulator and its maximum and minimum pressures.

The accumulator is sized to recover a portion α of the vehicle's maximum kinetic energy:

$$E_{acc} = \left(\frac{1}{2} * m * v_{max}^2\right) * \alpha \tag{18}$$

given the friction of tires and bearings, α is less than one.

The accumulator volume that allows to recover E_{acc} through an isentropic compression between p_{min} and p_{max} is expressed by the following:

$$V_{acc} = \frac{(1-k)E_{acc}}{p_{max}\left(\left(\frac{p_{min}}{p_{max}}\right)^{1/k} - \frac{p_{min}}{p_{max}}\right)} \tag{19}$$

For a predetermined value of p_{max}, the previous equation shows that the volume depends only on p_{min}. With a simple search of the minimum it results:

$$\frac{p_{min}}{p_{max}} = k^{\frac{k}{1-k}} \tag{20}$$

For the case of the reachstacker, with data in Table 2, α = 0,8 and p_{max}= 400 bar, the minimum pressure and volume of the accumulator are respectively equal to p_{min}=123 bar e V_{acc}=190 l.

TEST VEHICLE: THE REACHSTEACKER

The reference vehicle for the study of the hybrid transmission is the Kalmar Cargotec DRG 450 reach stacker (Fig. 3), whose main data are summarized in Table 2

FIGURE 3. The test vehicle.

Vehicle Data		Unit:
Engine Model	TAD-1150-VE	-
Max power @ speed	235 @ 1450 - 2000	kW@rpm
Max torque @ speed	1581 @ 950-1450	Nm@rpm
Maximum Load	112400	kg
Tyre Radius	0.825	m
Maximum Torque at Wheel	158000	Nm
Max Vehicle Speed	26	km/h
Gradeability at rated load	17	%

Table 2: Main data of the test vehicle.

Optimal layout of the transmission

The optimal sizing of the hybrid transmission followed the path previously outlined: the whole field of existence of the C function was probed, varying the θ angle between 0 and 2π, and, for each of the points of the curve, the sizing was carried out and the performance evaluated.

The average efficiency of the transmission extended to the whole vehicle speed range and rated at maximum power was taken as a performance parameter.

$$\bar{\eta} = \frac{\int_0^{v_{max}} \eta \, dv}{v_{max}} \tag{21}$$

The optimal sizing of the classic OC transmission has provided the results summarized in the Fig. 4, in which the average efficiency of the transmissions obtained by varying the θ angle and the full mechanical point speed is represented. As can be seen from the coloring, the two most interesting layouts are the RSC and the CSR, with v_{FMP} between 0.8 and 1.

As already mentioned, the hybrid OC transmission cannot have speeds higher than v_{FMP}, so the results concerning it are those in Fig. 4 with v_{FMP} = 1, and which are shown in Fig. 5. As can be seen, the CRS layout has values slightly higher than RSC.

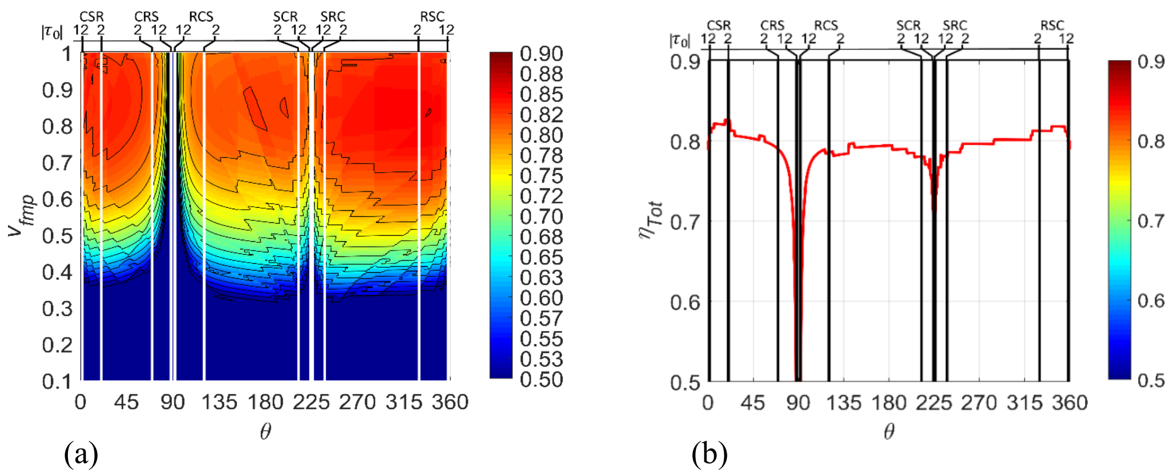

FIGURE 4. Average efficiency of the simple OC: a) whole map, b) map for v_{FMP}=1

Ultimately, the characteristics of the optimized transmission are those summarized in Table 3.

Table 3: Characteristic data of the hydraulic hybrid OC transmission

Transmission Data					
		Unit:			**Unit:**
τ_0	-2.08	-	$V_{\_2}$	400	cc
i_{out}	25,7	-	Δp_{max}	400	bar
$\tau_{\_2}$	1,99	-	$V_{\_acc}$	190	l
$\tau_{\_1}$	1,21	-	$p_{precharge}$	123*0.9	bar
$V_{\_1}$	107	cc	$m_{\_acc}$	1200	kg

MODELING AND SIMULATION

In order to evaluate the advantages of hydraulic hybridization, two transmissions have been modelled: the first one is the non-hybrid OC calculated according to the procedure illustrated; the second one is the hybrid transmission obtained from the previous one by inserting the accumulators. The model of the latter is shown in Fig.5.

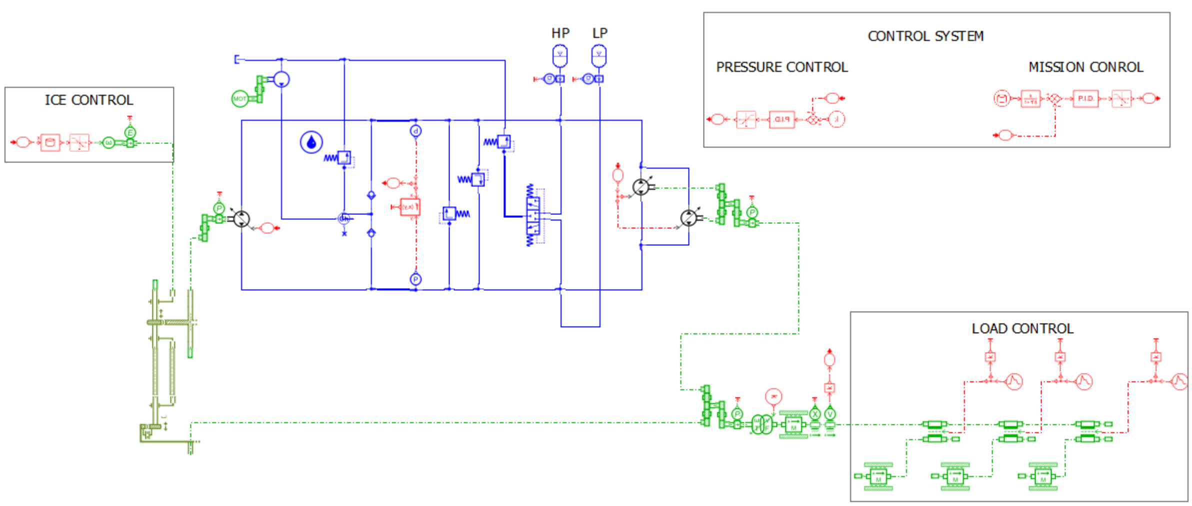

FIGURE 5. Scheme of the hybrid hydromechanical transmission

The two transmissions have different management criteria. The simple OC is, as mentioned, regulated on the speed of the vehicle through action on the cylinders of the two hydraulic units. The regulation criterion of the second one, on the other hand, requires that Unit 2 is adjusted to the speed of the vehicle, while Unit1 is adjusted to maintain the pressure of the high pressure line at a minimum level. At the current state of the work, the criterion has only been defined but not yet optimized, since in the literature and in the original patent [14] it did not appear. This evolution will concern future developments of the work.

This last criterion could be usefully applied also to the simple OC. In fact, with this solution the supply of cylinders for handling the container could take place from the two branches of the hydrostatic group, instead of providing a special power outlet after the engine.

To separate the effects of the new control criterion from those induced by hybridization, the comparison was made by steps. In the first step (step A in Table 4) the OC was simulated with its traditional regulation criterion; in the second step (step B) the traditional criterion has been replaced with that provided for the hybrid OC; finally, in the third step (step C) the hybrid OC with its criterion was simulated.

A typical load cycle was applied to the vehicle model, which was derived from [24].

As shown in the table, the new regulation criterion causes a doubling of energy consumption, while hybridization produces a 21% reduction in energy consumption.

Table 4. Energy consumption for the three steps

STEP	ENERGY MJ/cycle
A (OC+rc1)	9.65
B (OC+rc2)	19.2
C (Hybrid OC+rc2)	15.1

It should not come as a surprise that the new control criterion produces more energy consumption. In fact, the new criterion constantly maintains the pressure of the high pressure line at a high value: this produces high friction losses and, especially, high leakage flow rates in the three hydraulic machines. In the first scheme, on the contrary, the pressure is variable and often takes very low values, especially during vehicle's stops. As already mentioned, the criterion has not yet been optimized. On the contrary, the effect of hybridization is clear and promises further reductions as a result of refinements to the management criterion.

Figures 6-10 show the main quantities of the transmission during the work cycle. Figure 7 shows the vehicle speed and the loads; Figures 7 and 8 show the displacements of the cylinders of the two machines in case A and in case C: the reversals of the pump displacement due to the reverse gear are observed. The movements of the displacements change radically in the case of C in which, in addition to the pump, the accumulator also collaborates with the flow produced by the engine. Figure 9 summarizes the dynamics of the accumulator.

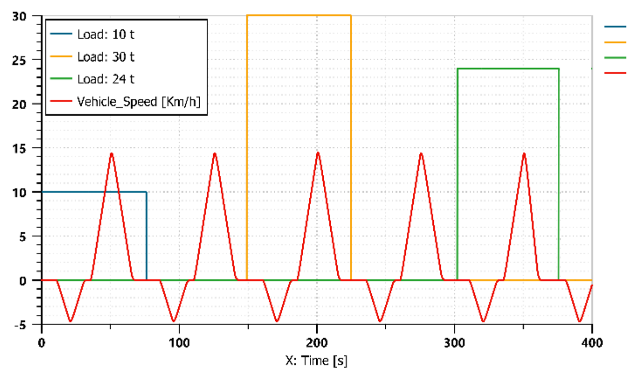

FIGURE 6. Working cycle: vehicle speed and loads

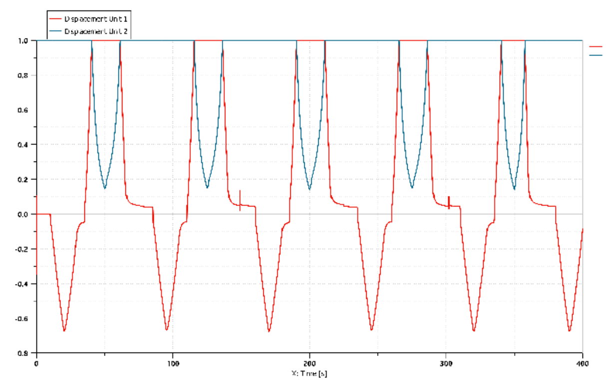

FIGURE 7. Displacement fraction of unit 1, 2 in case A

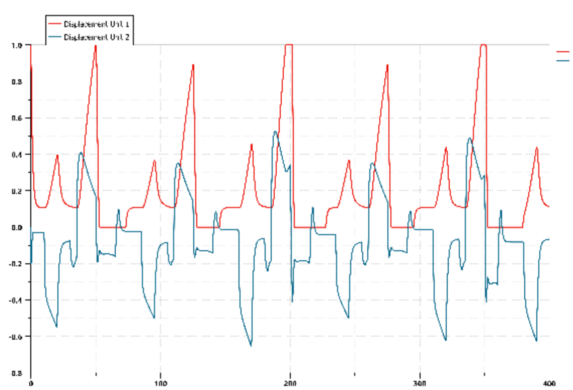

FIGURE 8. Displacement fraction of unit 1, 2 in case C

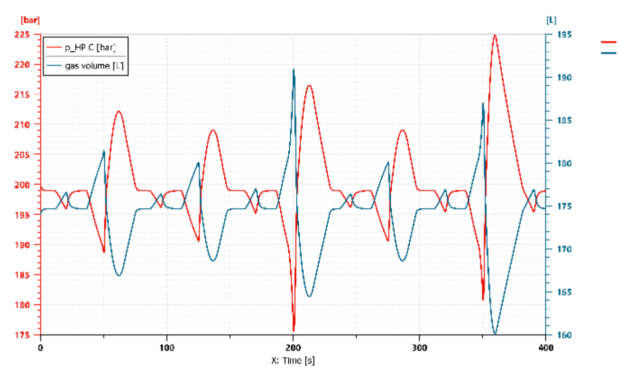

FIGURE 9. Pressure in high pressure line and gas volume in the accumulator in case C

CONCLUSIONS

In this work a procedure for the optimal sizing of an OC type hydromechanical transmission in hybrid or simple configuration has been presented. The sizing has been carried out through the continuous representation of the planetary gear, which allows to treat the transmission layout as a continuous variable and insert it within an optimization procedure, like any continuous variable. The procedure identifies the most efficient layout accompanied by the dimensions of the transmission components.

In the second part of the work the procedure was applied to the case of a reach stacker. The comparison between the hybrid transmission and its non-hybrid counterpart must be treated with caution since the two transmissions do not have the same regulation criterion. In this way the effect due to hybridization can be masked by the effects due to the action of the regulation criterion.

An attempt was made to separate, as a first approximation, the two effects. It emerged that the regulation criterion has deleterious effects on energy consumption; for this reason, it must be perfected. On the contrary, hybridization involves a reduction of about 20% of the energy required by the engine.

This point will be the object of future work in this topic.

Nomenclature:

m	Vehicle mass	[kg]
τ_0	Planetary Gear Ratio	[-]
τ	Gear Ratio	[-]
p	Pressure	[bar]
T	Torque	[Nm]
ω	Rotational Speed	[rad/s]
n	Rotational Speed	[rpm]
E	Energy	[J]
P	Power	[W]
V	Displacement	[m^3]
v	Vehicle speed	[m/s]
r_w	Wheel radius	[m]
S_{PGT}	Planetary speed equation	
T_{PGT}	Planetary torque equation	

Subscripts:

s	Sun
c	Carrier
r	Ring
ICE	Internal Combustion Engine
acc	Accumulator
FMP	Full mechanical point
hy1,2	Hydraulic Unit 1, 2
PGT	Planetary Gear Train
out	Rear axle

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REFERENCES

1. Wang J., Yang Z., Liu S., Zhang Q. and Han Y., “A comprehensive overview of hybrid construction machinery”. Advances in Mechanical Engineering 2016, Vol. 8(3) 1–15) 2016. DOI: 10.1177/1687814016636809
2. Lin T., WangQ., Hu B. , Gong W., “Development of hybrid powered hydraulic construction machinery”, Automation in Construction. Volume 19, Issue 1, January 2010, Pages 11-19.
3. Inderelst M., Losse S., Sgro S., Murrenhoff H., *Energy efficient system layout for work hydraulic of excavators*, The Twelfth Scandinavian International Conference on Fluid Power, 2011.

4. Pintore F, Borghi M, Morselli R, Benevelli A, Zardin B, Belluzzi F. *Modelling and Simulation of the Hydraulic Circuit of an Agricultural Tractor*. ASME. Fluid Power Systems Technology, 8th FPNI Ph.D Symposium on Fluid Power():V001T04A004. doi:10.1115/FPNI2014-7848.
5. Zardin, B., Borghi, M., Gherardini, F., Zanasi, N., "Modelling and simulation of a hydrostatic steering system for agricultural tractors", Energies Volume 11, Issue 1, January 2018, Article number 230
6. Casoli, A. Gambarotta, N. Pompini, L. Riccò (2014). "Development and application of co-simulation and control-oriented modeling in the improvement of performance and energy saving of mobile machinery", Energy Procedia, Volume 45, 2014, Pages 849–858. Elsevier. http://dx.doi.org/10.1016/j.egypro.2014.01.090. Codice Scopus: 2-s2.0-84893640233. Codice isi: 000345384000089
7. Kress J.H., "Hydrostatic power-splitting transmissions for wheeled vehicles – Classification and theory of operation", SAE Tech. Pap. 680549 (1968). doi:10.4271/680549
8. Renius K.T., Resch R., "Continuously variable tractor transmissions", ASAE Disting. Lect. 29 (2005).
9. Casoli P., Vacca A., G.,Berta, Meleti S., Vescovini M. (2007) *A numerical model for the simulation of Diesel/CVT power split transmission* SAE paper n. 2007-24-137, SAE-NA ICE2007, 17-20 Settembre, 2007 Capri-Napoli. ISBN: 978-88-900399-3-0. DOI: 10.4271/2007-24-0137
10. Blake C, Ivantysynova M, Williams K. "Comparison of Operational Characteristics in Power Split Continuously Variable Transmissions". SAE 2006 Commercial Vehicle Engineering Congress & Exhibition, October 2006, *SAE Techical Paper series 2006-01-3468*
11. Linares P, Méndez V, Catalán H. "Design parameters for continuously variable power-split transmissions using planetaries with 3 active shafts". J Terramechanics 2010;47(5):323–35.
12. Macor A., Rossetti A. "Optimization of hydro-mechanical power split transmissions", Mechanism and Machine Theory, Volume 46, Issue 12, December 2011, Pages 1901-1919. DOI: 10.1016/j.mechmachtheory.2011.07.007 Codice Scopus: 2-s2.0-80052967974.
13. Rossetti A., Macor A., Scamperle M. "Optimization of components and layouts of hydromechanical transmissions". International Journal of Fluid Power Volume 18, Issue 2, 4 May 2017, Pages 123-134. DOI: 10.1080/14399776.2017.1296746. Codice Scopus:2-s2.0-85015076287.
14. Ivantysynova, M.M., Carl, B.A., Williams, K.R.. Power split transmission with energy recovery. 2008. US Patent 20080081724 A1.
15. Kumar, R. A power management strategy for hybrid output coupled power-split transmission to minimize fuel consumption. Ph.D. thesis; Purdue University; 2010.
16. Kumar, R., Ivantysynova, M. *Investigation of various power management strategies for a class of hydraulic hybrid powertrains: Theory and experiments*. Proc. of the 6th FPNI PhD Symposium. 2010, p. 87–99.
17. Kumar, R., Ivantysynova, M. *The hydraulic hybrid alternative for toyota prius - a power management strategy for improved fuel economy*. In: 7th International Fluid Power Conference (7. IFK); vol. 2. 2010, p. 329–341.
18. Cheong K. L., Y. L. Perry, Sedler S., Chase T. R. *Comparison between Input Coupled and Output Coupled Power-split Configurations in Hybrid Vehicles*. Proceedings of the 52nd National Conference on Fluid Power. Milwaukee, WI, USA, 2011, pp. 243–252.
19. Van de Ven J. D., Olson M. W., and Li P. Y., *Development of a Hydro-Mechanical Hydraulic Hybrid Drive Train with Independent Wheel Torque Control for an Urban Passenger Vehicle*. Proc. of International Fluid Power Exhibition, Las Vegas, Nevada, United States, 2008.
20. Ramdan M. I., Stelson K. A. "Optimal design of a power-split hybrid hydraulic bus", Proc IMechE Part D: J Automobile Engineering 2016, Vol. 230(12) 1699–1718. DOI: 10.1177/0954407015621817
21. Macor A., Benato A., Rossetti A., Bettio Z., "Study and Simulation of a Hydraulic Hybrid Powertrain". Energy Procedia 126 (201709) 1131–1138. DOI: 10.1016/j.egypro.2017.08.279. Scopus 2-s2.0-85030667198. Isi code: 000426884100143
22. Rossetti A., Macor A. "Continuous formulation of the layout of a hydromechanical transmission". Mechanism and Machine Theory. Volume 133, March 2019, Pages 545-558. DOI: 10.1016/j.mechmachtheory.2018.12.006. Scopus 2-s2.0-85058939485. Isi code: 000455041300031
23. Kumar R. Sizing of the hydraulic version of Toyota Pruis. Report No: MAHA11-2010-in. Purdue University – MAHA Fluid Power Center - Lafayette-Indiana 30.04.2010.
24. Mercati S., Panizzolo F., Profumo G. *Power split Hydro-mechanical Variable Transmission (HVT) for off-highway application*, 10th International Fluid Power Conference (10. IFK) March 8 - 10, 2016, Vol. 3, pp. 69-80