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Energy Procedia 126 (201709) 1131–1138

Energy Procedia

www.elsevier.com/locate/procedia

72nd Conference of the Italian Thermal Machines Engineering Association, ATI2017, 6–8 September 2017, Lecce, Italy

Study and Simulation of a Hydraulic Hybrid Powertrain

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Abstract

In the agricultural and work machine sectors, the hydro-mechanical transmission competes with traditional mechanical transmission; a tool to improve its competitiveness is the hybridization. The hydraulic branch of the transmission may be integrated with two accumulators which allow the storage of energy derived from braking; the presence of the two hydraulic units facilitates this solution: the hydraulic motor acts as a pump during the braking and as a motor during the starts of the vehicle. This solution appears to be interesting also from the costs point of view, because it does not require a high-level technology. For this reason, in the present work, the hybrid hydro-mechanical transmission is studied with the aim of pointing out its potential for the urban passenger transport sector. At first, a hydro-mechanical transmission for an urban bus is designed; then, the same transmission is modified adding the components for the hydraulic hybridization. After that, the two vehicles is modelled and simulated using the AMESim code, and, finally, compared to each other in terms of energy savings.

Peer-review under responsibility of the scientific committee of the 72nd Conference of the Italian Thermal Machines Engineering Association

Keywords: Hydraulic Hybrid Powertrain, Hydro-mechanical Transmission, Hybrid Hydro-mechanical Transmission.

1. Introduction

The reduction in fuel consumption and emissions imposed by the regulations also for agricultural and heavy-duty machinery has forced the manufacturers towards hybrid solutions, as for the road transport sector.

The literature shows several examples of hybrid solutions taking advantage of energy storage devices such as electric batteries, supercapacitors or hydraulic accumulators [1, 2].

During the 90's, the continuous hydro-mechanical transmission has been introduced in agricultural tractors for comfort and marketing reasons [3, 4]. This transmission, paying a small reduction in efficiency, allows the thermal engine to be operated under the minimum consumption conditions, thus ensuring to the powertrain a slightly higher efficiency than that of the traditional powertrains, equipped with torque converter and power shift transmission [5, 6].

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Peer-review under responsibility of the scientific committee of the 72nd Conference of the Italian Thermal Machines Engineering Association
10.1016/j.egypro.2017.08.279

Hydraulic hybridization has recently been proposed for hydro-mechanical transmissions by Ivantysynova [7]. This hybrid system was theoretically and experimentally studied by Kumar [8] and Kumar and Ivantysynova [9, 10], who proposed a sizing method, focusing the attention on the problem of transmission management.

In this paper the hybrid hydro-mechanical transmission is studied and simulated in order to evaluate the actual benefits of hydraulic hybridization. Then, the operations and performance of a hybrid hydro-mechanical transmission is compared with the one of a non-hybrid hydro-mechanical transmission.

A 12 m class bus is used as reference vehicle for both the transmissions.

2. The Hydro-mechanical Transmission and Its Hydraulic Hybrid Version

2.1. The Output Coupled Hydro-mechanical Transmission

The hydro-mechanical transmission (HT) has two configurations, the so called Input Coupled and Output Coupled (OC) [3, 4]; the latter best suits hybridization.

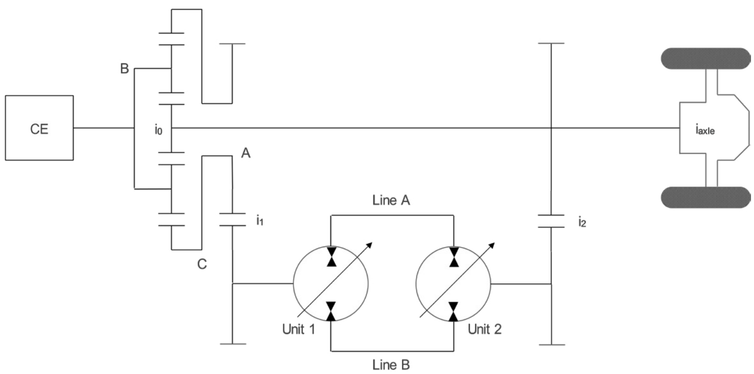

Fig. 1. Layout of the Output Coupled Hydromechanical Transmission.

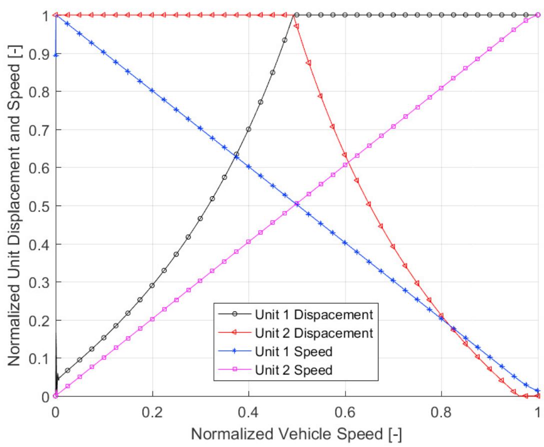

Fig. 2. Speeds and displacements of the units of an OC.

The OC configuration, see Fig. 1, consists of a planetary gear, which separates the thermal engine power into two branches: the mechanical branch and the hydraulic branch, the latter consisting of a traditional hydrostatic transmission. The continuous variation of the output speed is achieved by acting on the displacements of the two hydraulic machines. The transmission efficiency is the weighted average of the efficiencies of the two branches, in any case higher than that of pure hydrostatic transmission. The operation of the OC hydro-mechanical transmission is depicted in Fig. 2.

At start-up, the displacement of Unit 1, which operates as a pump, is null, so null is the speed of Unit 2, which operates as a motor. Now, Unit 1 runs at its maximum speed as the carrier of the epicyclical gear is stationary. As the displacement of Unit 1 increases, the flow sent to the motor grows and the speed of both the carrier and the vehicle increase. Consequently, the epicyclical gear decreases the ring velocity.

Once the maximum displacement of Unit 1 is reached, the further speed vehicle increase is achieved by decreasing the hydraulic motor displacement. The ring is slow down to the point where it is stop, also stopping the rotation of the pump. In this operational condition, called full mechanical point, all the engine power comes to the wheels only through the mechanical branch. This is the condition of highest efficiency for the transmission. Inverting the motor displacement, which now becomes a pump, the carrier still rises the speed but the ring inverts the direction of rotation. Now, part of the power flows from Unit 2 to Unit 1. This condition, called recirculating mode, further increases the vehicle speed at the expense of the efficiency.

2.2. *The Hybrid Output Coupled Transmission*

The hydraulic hybridization of the OC involves the insertion of two accumulators into the high and low pressure branches, according to the scheme proposed by Ivantysnova [7] and shown in Fig. 3.

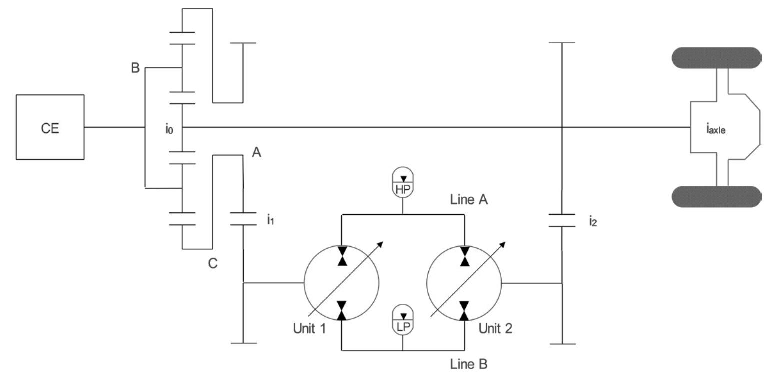

Fig. 3. Layout of the hybrid Output Coupled hydromechanical transmission.

The energy flows of the two sources can be handled sequentially or in parallel, i.e. the hydraulic source could be dedicated to the starts only, leaving the thermal engine in normal operation; or the hydraulic source could integrate the engine during operation as long as the accumulator pressure allows it. The charge of the hydraulic accumulator occurs during braking. It is clear that this transmission is suitable for all urban vehicles with frequent starts and stops, such as buses.

The operation of the transmission, therefore, is similar to that of a non-hybrid transmission, except for the regenerative braking phase, which takes place by putting the Unit 2 displacement in a negative position. The unit works now as a pump sending the oil into the HP accumulator.

3. Modelling of the Transmissions

The two transmissions have been sized for a 12 m urban bus, whose data are listed in Table 1. The sizing procedure starts by choosing the full mechanical point speed and axle gear ratio, as suggested by Blake et al. [11]. Results are summarized in Table 2.

The accumulators sizing has been designed by assuming of storing half of the maximum kinetic energy of the vehicle: this is necessary to take into account the energy dissipated in the brakes, the losses in the wheels and in the rear axle.

Table 1. Reference Vehicle data and maximum hydraulic parameters.

Parameter	Value
Engine Power	200 kW
Engine speed	1700 rpm
Maximum Vehicle speed	70 km/h
Wheel radius	0.5 m
Maximum wheel torque	10000 Nm
Max. speed for hydraulic units	3000 rpm
Max. pressure for hydraulic units	400 bar

Table 2. OC transmission parameters.

Parameter	Value
Full mechanical point vehicle speed	50 km/h
Axle gear ratio	2:1
Standing gear ratio	-2.17
Gear ratio, ring gear-Unit 1	3.8:1
Gear ratio, sun gear-Unit 2	5.7:1
Displacement of Unit 1	130 cc/rev
Displacement of Unit 2	126 cc/rev

The high pressure accumulator parameters are: p_{min}=150 bar, p_{max}=400 bar and Volume=45 l while the pressure of the low pressure accumulator is assumed equal to 20 bar, as for the OC low pressure line.

The mechanic and hydraulic elements of the two vehicles, as well as the control system, have been modelled and simulated using LMS Imagin.Lab AMESim commercial software [12]. The scheme of the HOC is shown in Fig. 4 while the scheme of the OC is not reported here because it differs only for the accumulators and the selection valve.

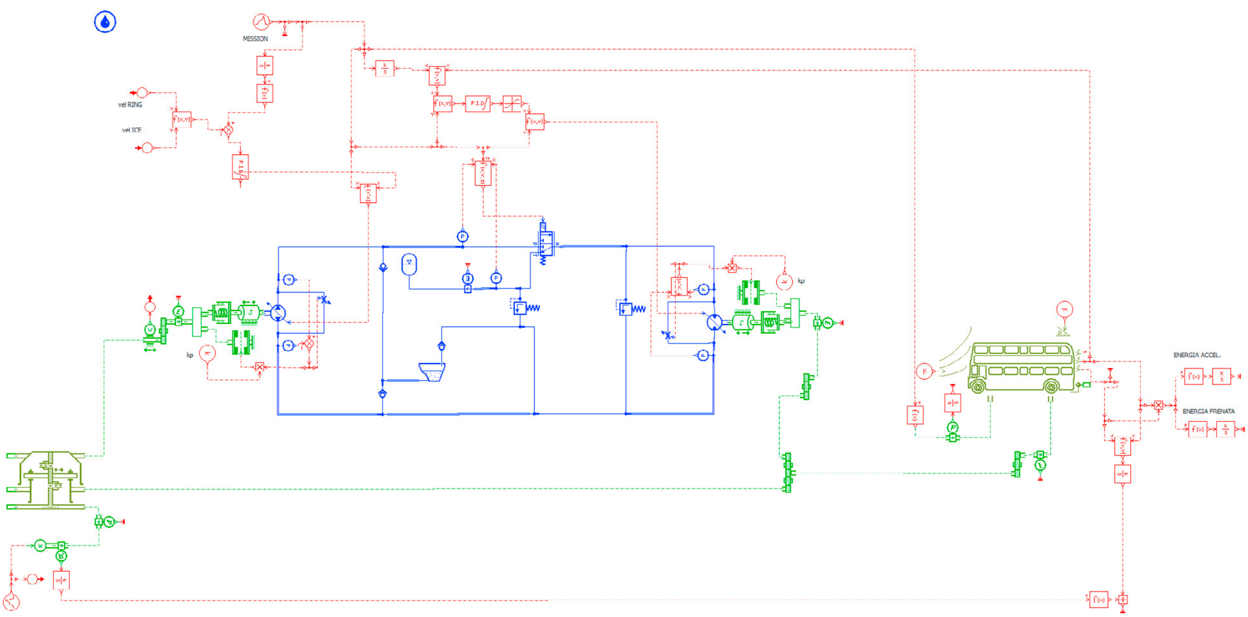

Fig. 4. Scheme of the reference vehicle with HOC transmission.

The hydraulic machines losses have been modelled based on the experimental data supplied by a manufacturer. The gear efficiency has been assumed equal to 97% for each gear pair; for each sun/planetary and ring/planetary contact and for the rear axle gear a 95% efficiency has been considered.

The relevant part of the scheme is the control of the vehicle speed, which is carried out by a PID controller. It receives the error between the desired speed and the vehicle speed to produce an output signal at first sent to the swash plate of the pump, and after to the swash plate of the motor.

3.1. Energy Sources Management

The two energy sources, the engine and the accumulator, are handled sequentially. The starting of the vehicle is always hydraulic, to the point where the accumulator pressure is able to maintain the required speed. After that, the control system increases the pump displacement and excludes the accumulator. During braking, the motor is put into a negative displacement, so, rotating in the same direction, turns into a pump, absorbing energy from the vehicle. The position of the valve is now inverted, allowing the flow to be directed to the accumulator.

4. Simulation Results

In order to compare the two solutions and to evaluate the energy benefits introduced by the hybridization, the OC and HOC transmissions are simulated on the basis of the speed profile shown in Fig. 5, designed to stress the hybrid solution.

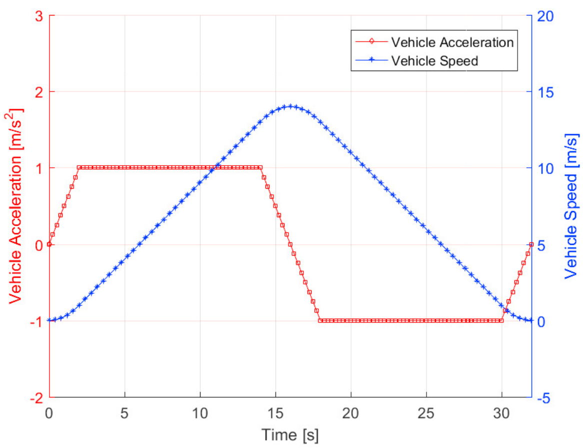

Fig. 5. Test speed profile.

4.1. Transmission Operation

4.1.1. The OC Transmission

In Fig. 6, the displacements of the two hydraulic machines are superimposed on the vehicle speed. At first the pump displacement increases, then, the motor displacement decreases; during braking the opposite occurs. The instability observed at the beginning of the braking phase is caused by the action of the mechanical brake.

4.1.2. The HOC Transmission

The vehicle acceleration is at first supported by the accumulator flow rate, then by the pump flow rate (Fig. 7). During braking, the pump displacement is set to zero, while the accumulator receives flow and stores the resulting energy. The charge and discharge phase is performed at an average efficiency of 85%.

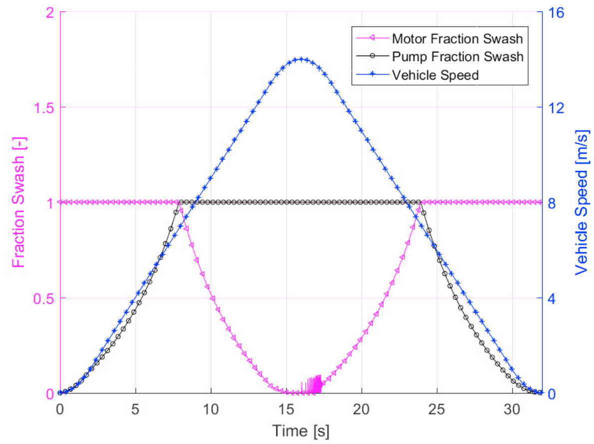

Fig. 6. OC transmission: displacements of hydraulic machines during operation.

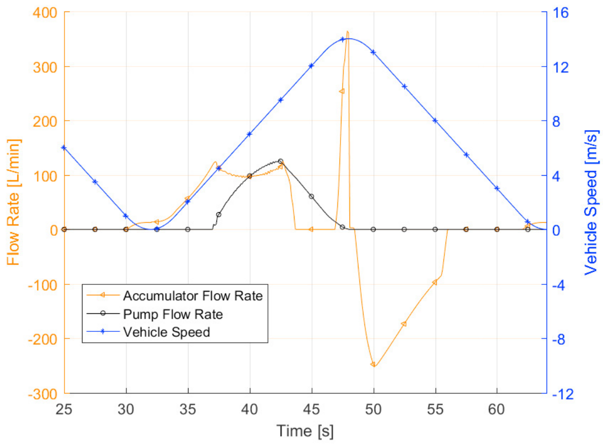

Fig. 7. HOC transmission: flow rates to the hydraulic motor during operation.

In Fig. 8, the engine power for the two transmission during two cycles is shown. It can be noticed that the HOC engine is called to operate 5 seconds after the OC one, saving in this way a not negligible quota of energy. However, the peak power remains the same for the two transmissions.

4.2. *Energy Comparisons*

Energy comparisons are based on a speed profile consisting of 10 elementary profiles as the one depicted in Fig. 5. Table 3 shows the comparison for the two transmissions.

The average OC efficiency is about 77%, in line with the literature values. HOC's efficiency rises to a value somewhat high, 95%, which is a consequence of neglecting the energy contribution of the accumulator. If the accumulator input is taken into account, the efficiency drops to 71%, below the OC value. This is caused by the higher pressure levels arising in the HOC, that produced higher losses in hydraulic machines.

Regarding the braking energy recovery, two observations can be claimed.

The accumulator charge and discharge phase efficiency is 85% on average. This value essentially depends on the resistances in the accumulator inlet.

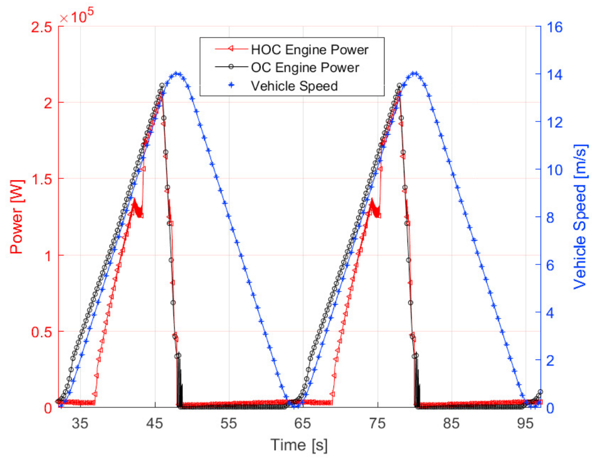

Fig. 8. Engine power for the two transmissions during operation.

Table 3. Energy comparison between OC and HOC transmissions.

Parameter	OC	HOC
Transmission efficiency	0.777	0.715 (0.955)
Accumulator efficiency	/	0.857
Recovery efficiency	/	0.398
Energy supplied by the engine	17.04 MJ	13.78 MJ

The recovery efficiency, defined as the ratio between the energy supplied to the vehicle by the accumulator and the kinetic energy available at the braking, is 40%. This value could be raised by carefully selecting the accumulator volume and working pressure levels.

Finally, the energy consumption of the HOC transmission is about 19% lower than that of the OC.

This result can be improved by optimizing the accumulator. To meet this need, an optimization procedure has been performed by varying the volume and the pre-charge pressure of the accumulator between 20 l and 200 l and between 50 bar and 300 bar, respectively. The mass of the vehicle has been increased by a quantity proportional to the volume of the accumulator. The maximum pressure of the pressure relieve valves has been kept constant at 450 bar.

The energy consumption of the engine, referred to the energy consumption of the OC power train under the same conditions, is depicted in Fig. 9.

Small volumes produce, as is obvious, the least energy recovery. On the other hand, during braking recovery, they quickly lead to high pressure levels that put pressure relief valves and mechanical brakes in action.

Two minimum consumption areas can be observed. A first area is located in the high volume zone (> 100 l) and in the high pressures (250-300 bar) zone: this is a predictable result because the accumulated energy directly depends on the volume-pressure product. A second area is located in an intermediate volume (50-100 l) zone and low pressure zone.

It is worth noting that the energy savings generated by hybridization can reach 35%. This is an encouraging result, but it must not be forgotten that it has been obtained with a theoretical cycle at a constant engine speed. More significant values can be obtained by considering the standard road cycles and a suitable model of the thermal engine. This is the future development of the research.

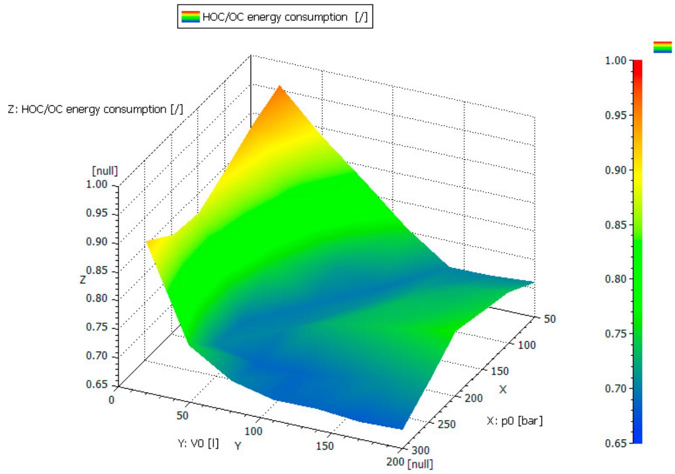

Fig. 9. Energy consumption of the HOC transmission referred to the energy consumption of the OC transmission.

5. Conclusions

The hydro-mechanical transmission is well suited to hydraulic hybridization, as the same hydraulic machines in the transmission can be used for this transformation; moreover, mature technology components such as accumulators, can be used.

The study presented in this paper focuses on the performance of a hybrid hydro-mechanical transmission of the OC type. Comparison with non-hybrid counterparts showed promising energy savings ranging from 20% to 35%. More reliable savings can be obtained by adding to the transmission model the one of the thermal engine, which has been considered here for simplicity like a fixed-speed engine.

Finally, it should be noted that the potentialities of this transmission cannot be fully highlighted without an optimal management of both the transmission and the engine.

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