Study of a dual layout Input-to-Output Coupled hydromechanical transmission

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Abstract.

Hydromechanical three-shaft transmissions, the well-known Input Coupled and Output Coupled, require additional mechanical ranges in order to cover the usual speed ranges of off-highway vehicles. This increases the costs and also the complexity of the transmission.

This work proposes a new configuration that does not require additional mechanical ranges. It consists of a single three-shaft hydromechanical unit, which is operated in dual mode through the action of some clutches: as an IC transmission at low speeds and as an OC transmission at high speeds. In this way the speed range the transmission can cover is doubled.
The performance comparison of the dual layout with that of the Input Couped and Output Coupled transmissions showed that the former maintains high efficiency values for wider vehicle speed ranges than the other two, while reaching the same efficiency peaks.

Keywords. Hydromechanical transmission, dual layout transmission, power split transmission, ICtoOC.

1. INTRODUCTION

The hydromechanical transmission, introduced in the agricultural sector in the 90s, is an interesting alternative to the power-shift transmission, since it offers greater driving comfort and the possibility of optimal engine management [1, 2]. The performance of a hydromechanical transmission, intended both as efficiency and as a functional capacity to meet speed and torque requirements, are strongly conditioned by its layout, which can take a simple or complex forms. The former are the well-known Input Coupled (IC) and Output Coupled (OC), having a single planetary gear; the latter are known as Dual Stage and Compound, having complex planetaries [3] and additional mechanical gears.

However, simple layouts cannot cover the usual vehicular speed ranges without operating at low efficiency conditions. Therefore, they too must adopt additional gears [2]. For example:

Vario from Fendt (OC + 3 gears), HVT R2 from Dana Rexroth (IC + 3 gears) [4], ZF cPower (OC + 2 gears) [5].

Simple hydromechanical transmissions have been the subject of numerous studies aimed at the design [3, 6, 7, 8] and at the optimization of the transmission [9], as well as at the optimal management of the powertrain enabled by the continuous speed variation [10, 11].

The multirange solutions were proposed in the 70's by Orshansky [12] who analysed the functioning of the hydromechanical transmissions with 3 planetary gears and three gear steps, and showed their advantages. Jarchow in his patent [13] proposed a compound scheme with *n* additional ratios. Petterson [14] will take up this pattern by transforming the design method for complex hydromechanical transmissions in an multiobjective optimization problem, whose objective functions are the energy efficiency and the installed displacements. The design method was applied to a heavy wheel loader.

Subsequently various multirange solutions were proposed, based on simple and complex layouts, applied to different off-highway vehicles.

Xia et al. [15] proposed a Compound in power recirculation showing that at the start the transmission provides a high torque value. The solution was applied to a tractor whose parameters were optimized using the multi-objective genetic algorithm [16].

Wan et al. [17] established the characteristic equations of the IC layout and applies them to the case of a four step gear transmission for a wheel loader.

Liu et al. [18] presented a new design method for the study of a multi-range hydro-mechanical transmission. It was based on dual stage IC layout and its gear ratio extension was obtained by means of another planetary gear.

Zu et al. [19] studied a composite powertrain for tractors consisting of a hydromechanical transmission with 4 planetaries and two energy sources: thermal and electrical energy. They defined an optimal energy management criterion and achieved significant consumption reductions compared to the power-shift and hydromechanical power-split tractor of the same power.

Apparently secondary aspects of hydromechanical transmissions, but essential in their functioning, have also been studied. For example, Samorodov et al. [20], studying the braking phase of an OC transmission for a mine locomotive, found some situations of excessive overpressure in the hydrostatic group during the braking phase.

From the above, the need to cover wide speed ranges implies the use of complex planetary gears and additional gears, with construction complications and cost increase.

In this work we propose a solution based on the idea that the same hydromechanical group, thanks to the action of some clutches, can switch from the IC driving mode to the OC driving mode. In this way the speed range can be practically doubled without excessive complications.

Since the shift between the two operating modes can only take place from IC to OC, the proposed layout has been called ICtoOC.

Even if this solution reduces the constructive complexity, it does not guarantee a priori better performances with respect to the two parent layouts. Therefore, in the paper the potential of this configuration will be studied through some performance comparisons with the IC and OC applied to the same vehicle.

The forklift is the reference vehicle, whose low speeds do not require additional gears to the IC and OC. In this way the forklift does not privilege one solution over another.

The work will develop in the following way. Section 2 will present the transmission and define its sizing criteria; in Section 3 the design and the models of the transmissions to be compared will be illustrated; in Section 4 comparisons will be made between the three configurations operating along a constant load cycle.

2. THE ICTOOC TRANSMISSION

The scheme of the ICtoOC transmission is shown in Figure 2.1. At zero vehicle speed, clutches A and B are closed, C and D open; the transmission is in IC driving mode and in power recirculation mode [2]. The solar and the ring rotate in the opposite direction and the carrier is stationary. By reducing the displacement of unit 1, D_1, the speed of the ring decreases and the speed of the carrier, i.e. the vehicle speed, increases. The power transferred through the hydrostatic transmission is reduced as the vehicle speed increases, thus reaching the full mechanical point, where all the power is transferred through the mechanical branch. Beyond this point the transmission still works in IC, but in additive mode, up to the change speed (V_{shift}). At this point, clutches C and D close and A and B open, and the transmission begins to work in OC driving mode and in power additive mode, until it reaches the second full mechanical point at maximum speed.

The transmission is managed by adjusting only the displacement of unit 1.

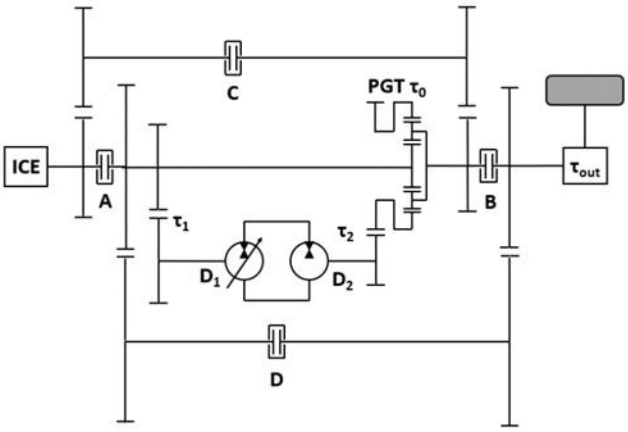


Figure 2.1. Scheme of the ICtoOC transmission.

2.1. Sizing the ICtoOC transmission

The ICtoOC power split transmission can be sized accordingly to the standard procedure used to design IC and OC powertrain, except that the shifting point has to be accounted for and each element has to be verified both in the IC and in the OC driving mode.

Given the constitutive velocity equation of the planetary gear of (1), the full mechanical point is obtained when the ring shaft speed is null reducing (1) to (2).

$$\tau_0\ \omega_s + (1-\tau_o)\omega_c + (-1)\omega_r = 0 \tag{1}$$

$$\tau_0\ \omega_s + (1-\tau_0)\omega_c = 0 \tag{2}$$

where ω is angular velocity, r, c, s stand for ring, carrier, and solar of the planetary, respectively.

When the IC driving mode is selected, the sun and the ICE are rigidly connected to each other and the carrier velocity is proportional to the vehicle speed:

$$\omega_s = \omega_{ice} \tag{3}$$

$$\omega_c = \frac{\omega_{out}}{\tau_{out}} = \frac{1}{\tau_{out}} \frac{V}{2\pi\ R_{tire}} \tag{4}$$

On the opposite, when the OC driving mode is selected, the engine is connected to the carrier shaft while the sun drives the output shaft:

$$\omega_c = \omega_{ice} \tag{5}$$

$$\omega_s = \frac{\omega_{out}}{\tau_{out}} = \frac{1}{\tau_{out}} \frac{V}{2\pi\ R_{tire}} \tag{6}$$

Combining the couples of equations (3) and (4) and (5) and (1) with (2), which express the full mechanical point condition, the two full mechanical point velocities V_{fmp}^{IC} and V_{fmp}^{OC} can be computed:

$$V_{fmp}^{IC} = \frac{\tau_0}{(\tau_0-1)} \omega_{ice} \tau_{out}\ 2\pi\ R_{tire} \tag{7}$$

$$V_{fmp}^{OC} = \frac{\tau_0-1}{\tau_0} \omega_{ice} \tau_{out}\ 2\pi\ R_{tire} \tag{8}$$

Equations 7 and 8 can then be used to define the standing gear ratio τ_0 of the planetary gear and the axle gear ratio τ_{out} once the two full mechanical points of the transmission are chosen. The full mechanical point velocity of the OC configuration V_{fmp}^{OC} can be assumed equal to the maximum vehicle speed as usually done in OC layout. The V_{fmp}^{IC} can be set to optimize the average efficiency of the IC mode or to favour a specific velocity. As a general rules the IC full mechanical point is designed at one third of the maximum speed as this has proved to lead to the best efficiency for this layout [9]. This design rule can be applied considering, instead of the maximum speed, one third of the shifting speed, which is the maximum speed realized using the IC driving mode (see (10)).

The request for a synchronous shift between the IC and the OC modes leads to the identity between the solar and the carrier shaft $\omega_s = \omega_c$. This condition, along with (1), defines the shift condition as the one where all the planetary gear shafts move synchronously as in the power split node:

$$\omega_s = \omega_c = \omega_r \tag{9}$$

As at the shifting point all the planetary shafts are synchronous, this rotational speed has to match the ICE speed, no matter if the IC (3) or the OC configuration (5) is selected, leading to identify the shifting speed, using (4) or (6):

$$V_{shift} = \omega_{ice}\tau_{out}\ 2\pi\ R_{tire} \tag{10}$$

Is interesting to notice here that the shifting speed V_{shift} corresponds to the geometric average of the IC and OC mechanical points

The gear ratios τ_1 and τ_2 are designed to limit the speed of the hydraulic units below the design value $\omega_{Hy\,Max}$. For the hydraulic unit connected to the power split node, the maximum of the ICE speed (connected during IC mode) and the speed at maximum velocity V_{max} has to be verified:

$$\tau_1 = \frac{\omega_{Hy\,Max}}{\max\left(\omega_{ICE}, \frac{1}{\tau_{out}} \frac{V_{max}}{2\pi\, R_{tire}}\right)} \tag{12}$$

For the hydraulic unit connected to the power merge node (i.e. the planetary gear), the highest velocity of ω_c, whichever it occurs at zero speed, at shifting speed V_{shift} or maximum velocity V_{max} has to be verified:

$$\tau_2 = \frac{\omega_{Hy\,Max}}{\max\left(\omega_c \Big|_{\substack{V=0 \\ V_{shift} \\ V_{max}}} \right)} \tag{13}$$

Once the gears ratios τ_1 and τ_2 are known, the hydraulic units can be sized. Each unit has to be sized to maintain the hydraulic system pressure below the maximum design value and to provide a coherent flow rate between the two units:

$$D_1 = \frac{2\pi\, T_1}{\Delta P_{max}}, \quad D_2 = \frac{2\pi\, T_2}{\Delta P_{max}} \tag{14}$$

$$\alpha_1\ \omega_1 D_1 = \omega_2 D_2 \text{ with } \alpha_1 < 1 \tag{15}$$

To calculate the torques T_1 and T_2, the most severe condition has to be identified for the specific vehicle considering the two following situations:

- Vehicle still, maximum pulling force, IC mode;
- Shifting speed, maximum power, OC mode.

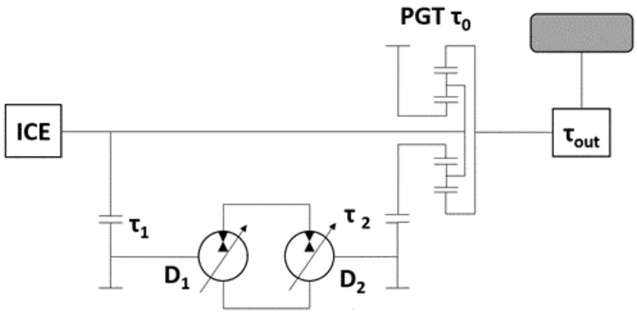
3. SIZING AND MODELLING OF THE TRANSMISSIONS

The three transmissions under comparison were sized according to the suggestions of Section 2. They are designed for a 5 t forklift with 75 kW, whose characteristics are shown in Table 3.1.

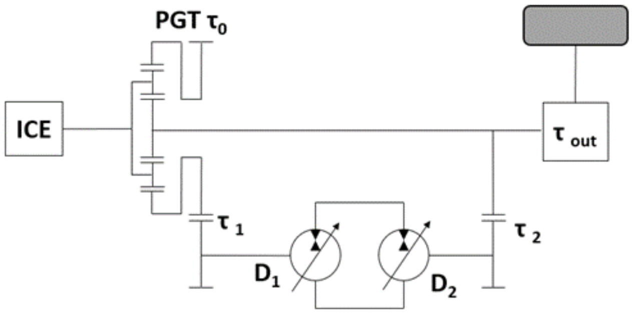
Table 3.1. Characteristic data of the vehicle

	Value	Unit
$Engine\ Power$	75	[kW]
$Engine\ Speed$	2400	[rpm]
$Maximum\ Wheel\ Torque$	17290	[Nm]
$Maximum\ Vehicle\ Speed$	24	[km/h]
$Wheel\ Radius$	0.38	[m]

Figure 3.1 shows the schemes of the IC and OC transmissions, while the scheme of the ICtoOC is in Figure 2.1. The sizing of the IC transmission requires the assumption of V_{fmp}. Since the full mechanical point is the highest efficiency point, the value of its speed has been chosen equal to the one with the highest use frequency of the forklift, which is around 4 km/h.



(a)



(b)

Figure 3.1. Scheme of the IC (a) and OC (b) transmissions

The full mechanical point speed for the OC transmission, on the other hand, is assumed to be equal to the maximum speed of the vehicle, as is usually done in order to prevent the power recirculation mode.

The ICtoOC sizing maintains the two previous full mechanical point speeds and assumes the shift speed equal to their geometric mean, as established in Section 2.

Table 3.2 summarizes the values of the full mechanical point and shift speeds together with the parameters produced by the sizing. It should be noted that the reduction in the speed range due to the change from IC to OC mode in the ICtoOC layout reduces the overall displacement of the hydraulic units, potentially leading to a more compact CVT unit.

The three transmissions were modeled through Amesim [21] (Figure 3.2). Particular attention was given to the definition of the loss models of the two hydraulic units, since the performance of the transmission essentially depends on their energy losses. The volumetric losses and the hydromechanical losses of the hydraulic machines have been described by a polynomial model derived from experimental measurements [22]. Losses in ordinary gears were defined by an average efficiency equal to 0.98, while for the epicyclical gears, an average efficiency of 0.97 was assumed for each pair of engaged gears.

For the sake of simplicity, the internal combustion engine has been replaced by a constant speed input. Each of the three transmissions is managed by a PI-type control system that tracks the vehicle speed by acting on the displacement of unit 1 (see Figure 2.1).

Table 3.2. Functional data of the transmissions

	IC	OC	ICtoOC
V_{fmp} [km/h]	4	24	4 and 24
V_{shift} [km/h]	/	/	9,8
D_1 [cc]	55	160	110
D_2 [cc]	140	80	40
τ_1	1.25	0.477	0.51
τ_2	0.77	1.25	1.26
τ_{out}	14	40	35
τ_0	-5.1	-1.8	-1.5
ΔP_{max} [bar]	400	400	400

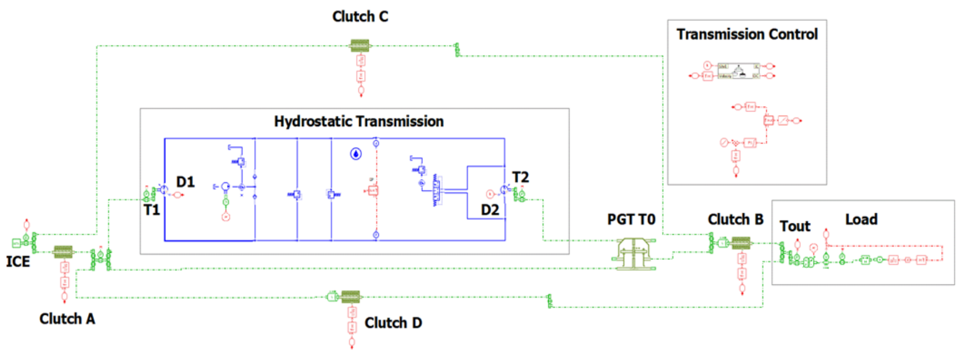


Figure 3.2. Amesim model of the ICtoOC transmission.

4. PERFORMANCE COMPARISON

The comparison between the three transmissions was carried out by means of two quantities: the energy consumption, i.e. the energy supplied by the engine, and the transmission efficiency, defined as the ratio between the power at the wheels and the power supplied by the engine. The calculations were carried out at maximum power condition, with gradually increasing speed between zero and the vehicle maximum speed.

Figure 4.1 shows the trend of the displacement and the speed of the hydraulic machines of the ICtoOC transmission along with the truth table of the clutches (Table 4.1).

The energy consumptions of the three transmissions are shown in Figure 4.2. IC is the least efficient layout, while the OC and the ICtoOC have lower consumption by 5.2% and 9% respectively. These results are the consequence of the losses in the three transmissions. While IC and OC layouts have only one full mechanical point, 4 km/h and 24 km/h respectively, the ICtoOC has two at the same speeds, as shown by the efficiency curve in figure 4.3. It is worth remembering that full mechanical point is the functional condition with maximum efficiency, since here all the power is transmitted via the most efficient mechanical path, and not via the hydraulic path, which is always the site of a double energy conversion. Furthermore, the ICtoOC benefits from the small size of the hydraulic machines, which leads to fewer losses.

As expected then, the ICtoOC can take the best response from the two configurations.

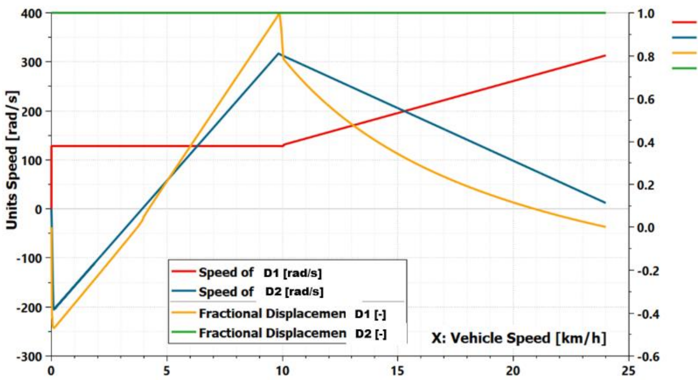


Figure 4.1. Displacements and speeds of the hydraulic units of ICtoOC transmission.

TABLE 4.1. Truth table of the clutches of the ICtoOC transmission.

	A	B	C	D
0 - V_{shift}	✓	✓	×	×
V_{shift} - V_{max}	×	×	✓	✓

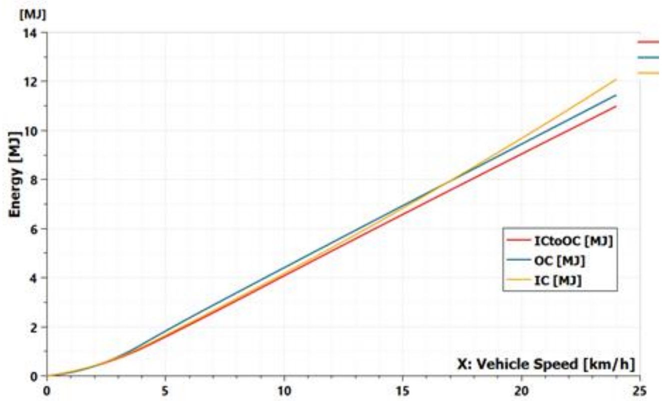


Figure 4.2. Energy consumption of the transmissions.

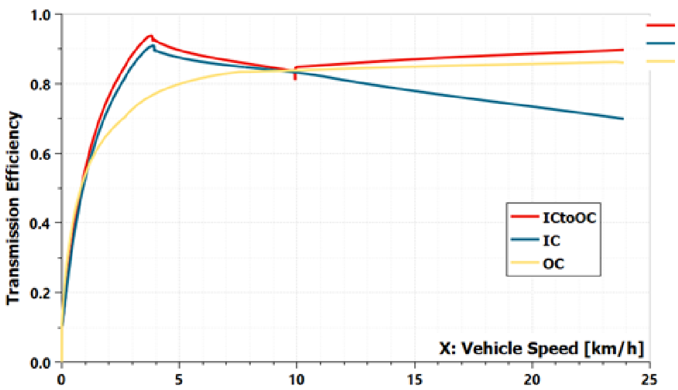


Figure 4.3. Efficiency of the transmissions.

5. CONCLUSIONS

The paper proposes a new three-shaft layout in which a single hydromechanical group (planetary + hydrostatic group) can switch from the IC layout to the OC layout, thereby extending the vehicle's speed range without adding mechanical gears. The powertrain is designed so to realize the shift between the IC and the OC layout at synchronous condition between the engaging shafts, to avoid discontinuity in the vehicle speed and to reduce the friction engaging time.

With this solution some advantages can be obtained:

- the layout has two full mechanical points, in which the power is transmitted only mechanically, therefore with high efficiency;

- the prevalent operation of the transmission is the additive mode, the most efficient, except at start-up in which the mode is power recirculation;

- the overall displacement of the hydraulic units is reduced compared to the corresponding one for the IC and OC layouts, with advantages in terms of losses, costs and encumbrance.

The simplification due to the lack of additional ranges is partially offset by the need to introduce four clutches (three considering one still present between engine and transmission).

The new layout was compared with IC and OC layouts applied to the case of a fork-lift. Under maximum power condition, the energy consumption of ICtoOC resulted 4% and 9% lower than the one of IC and OC layouts respectively.

Future developments of this solution will have to include simulations on realistic loading and unloading cycles and the optimal sizing to adapt the design parameters to the characteristics of the vehicle. Further interaction between this solution and possible energy recovery strategies will be also analysed.

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Biographies

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