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Design considerations about the hydromechanical transmission IC2OC

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Abstract. The hydromechanical transmission IC2OC operates as an Input Coupled transmission at low speeds and as an Output Coupled transmission at high speeds; a series of clutches allows the shift between two layouts. In this way the transmission speed range can be almost doubled.
The design of this transmission requires the full mechanical speed value for each layout. Since these values heavily influence transmission performance, their choice must be made carefully.
In this paper two choice criteria for these two speeds are proposed and discussed: the first one refers to the optimal choice for each basic layout; the second sets these speeds in correspondence with the speed of the maximum energy processed in a working cycle.
The effectiveness of these two criteria, applied to an off-road vehicle transmission, demonstrates that the former guarantees greater efficiency than the latter, despite its good assumptions.

1. Introduction
The recent proposal of European Commission allowing sales of new cars with internal combustion engines fueled by e-fuels, let us to foresee the use of internal combustion engines for many more years after the fateful date of 2035. In this context, developments on engines and transmissions could still play a significant role, especially in the off-highway machinery sector, where the penetration of electrification appears more difficult.
The most recent novelty introduced in this sector is the hydromechanical transmission [1, 2, 3], which has been applied in high-power agricultural tractors since the 90s .
The performance of a hydromechanical transmission is strongly conditioned by its layout, which can have 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 have more planetaries and additional mechanical ranges [4,5].
The hydromechanical transmission have been the subject of many research studies aimed at the performance analysis [6 - 8], the design [9 - 14] and the optimization of the components [15]. In [16] the optimization concerns layout and components simultaneously.
The management of these transmissions plays an important role in achieving high performance of the powertrain. In fact, the hydromechanical transmission is a continuous transmission and therefore it separates the speed of the wheels from that of the engine, which can be managed according to minimum consumption criterion [17, 18]. The management criterion can also be aimed at minimizing emissions, as done by the authors in [19] using the DALY parameter as a summary indicator of emissions.

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Finally, these transmissions have also been designed as part of a hybrid system, both electric [20] and hydraulic [21, 22]; the latter system is justified by the low level of technology required and the good performance obtainable.
Simple layouts IC and OC cannot cover the usual vehicular speed ranges without operating at low efficiency conditions: therefore additional gears must be added [5]. For example: Vario from Fendt (OC + 3 ranges), HVT R2 from Dana Rexroth (IC + 3 ranges) [23], ZF cPower (OC + 2 ranges) [24].
The authors have recently proposed a new transmission configuration, called IC2OC [25], which, thanks to an appropriate clutch set, allows to switch from the IC configuration to the OC one and vice versa. In this way the speed range of the transmission can be extended without adding mechanical gears.
Both the IC and OC configurations show a particular operating condition, called full mechanical point (FMP), in which the power flows only through the mechanical branch, minimizing the losses in the hydraulic branch, now due only to the no-load losses of one of the two hydraulic machines. Since the IC2OC configuration shifts from the IC configuration to the OC one, it exploits two FMPs and, therefore, two high efficiency operating zones. Therefore, the FMP and gear positions within the transmission speed range are important to achieve high operating efficiency. However, as these positions have to be chosen in advance during the project, it becomes necessary to have an adequate criterion for the choice.
The criterion proposed in this work establishes the FMP and shift conditions as those most frequently used, identified through statistical analysis of data collected in the field. This criterion was applied to the design of the reach stacker transmission, exploiting measurements obtained by DANA during an experimental campaign [18]. The transmission was sized according to the criteria set out, and was simulated on working cycles and loads deduced from statistical analysis. To highlight any strengths of the IC2OC, the same vehicle was equipped with the IC and OC transmission and simulated with the same loads

2. The IC2OC hydromechanical transmission

The IC2OC transmission arises from the observation that the two simple layouts IC and OC differ only in the position of the epicyclic gear with respect to the engine and the hydrostatic unit (figure 1). Therefore, with a suitable series of connections, it is possible to pass from one layout to the other.
As shown in figure 1, by closing clutches A and B and opening clutches C and D, layout IC is formed; conversely, by opening the last two and closing the first two, layout OC is formed.

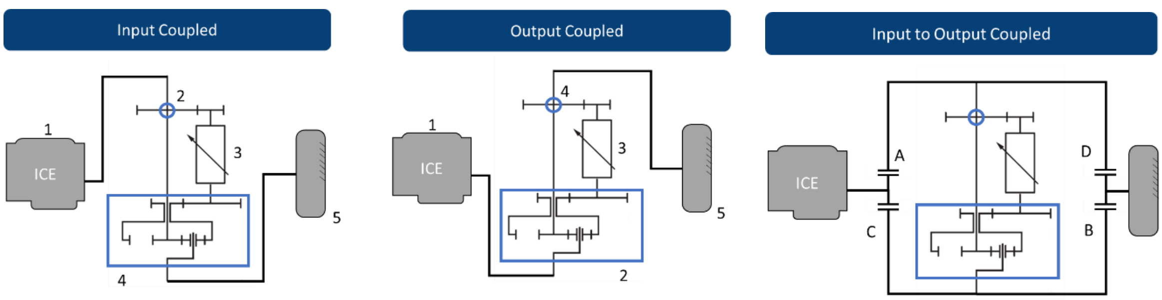

Figure 1. IC, OC, and IC2OC layouts.

So the change of layout does not take place by changing the transmission but only by acting on a series of clutches. It remains to be seen whether this configuration is advantageous over the simple two. It is known that each of the two simple layouts has a functional condition in which the power is transmitted only through the mechanical way, called full mechanical point, which is therefore characterized by the highest efficiency (figure 2). In the IC2OC layout there are two FMPs, guaranteeing two high-performance zones. A second advantage of this layout is the extension of vehicle speed without having to adopt additional ratios. Finally, observing the graphs in figure 2, it can be deduced that going from

IC to OC, both FMP can be reached by regulating only one hydraulic machine, while the other has a fixed displacement. To avoid slipping in the clutches when switching between the two layouts, the engine speed must be equal to that of the output shaft on the differential, as can be deduced from figure 1.

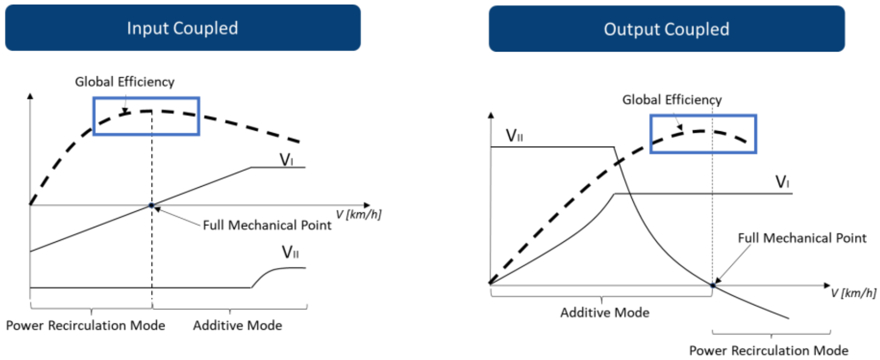

Figure 2. IC and OC characteristic curves.

2.1 Transmission design

The transmission design requires the calculation of the transmission ratios of the planetary gear and of the ordinary gears, the definition of the shifting condition, and the calculation of the displacement of the two hydraulic machines.

With reference to the scheme of figure 3, the design procedure develops as follows.

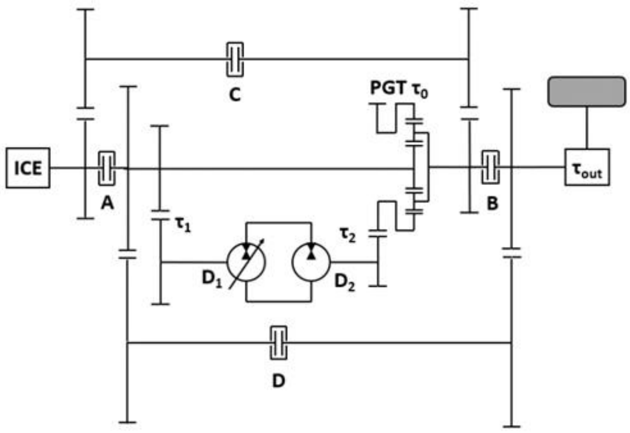

Figure 3. IC2OC characteristic curves.

In the full mechanical point, the planetary ring is still. So, the two FMP conditions for the transmission can be obtained via the Willis equation for the planetary gear (equation (1)) by setting to zero the ring shaft speed.

$$\tau_0 = \frac{\omega_r - \omega_c}{\omega_s - \omega_c} \tag{1}$$

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

In IC mode, i.e. A and B clutches closed,

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$$\omega_s = \omega_{ice} \qquad \omega_c = \omega_{ice}\frac{\omega_w}{\tau_{out}} = \omega_{ice}\frac{V}{\tau_{out}R_w} \tag{3}$$

In OC mode, i.e. C and D clutches closed,

$$\omega_c = \omega_{ice} \qquad \omega_s = \omega_{ice}\frac{\omega_w}{\tau_{out}} = \omega_{ice}\frac{V}{\tau_{out}R_w} \tag{4}$$

Combining equation (3) with equation (2), and equation (4) with equation (2), the expression of the FMP speed for both modes ca be obtained.

$$V_{fmp}^{IC} = \frac{\tau_0}{(\tau_0-1)}\omega_{ice}\tau_{out}R_w \tag{5}$$

$$V_{fmp}^{OC} = \frac{(\tau_0-1)}{\tau_0}\omega_{ice}\tau_{out}R_w \tag{6}$$

Once the two FMPs speeds are chosen, equations (5) and (6) can be used to calculate the standing gear ratio τ_0 of the planetary gear and the axle gear ratio τ_{out}. Since the FMPs are the functional conditions with the highest efficiency, their choice could be made in correspondence with the speeds of most frequent use of the vehicle. Following this suggestion, in the Section 3.1 a more general method based on the statistical treatment of data collected in the field will be proposed.

The synchronous shift between IC to OC modes requires the two speeds in clutch B to be equal, i.e. ω_s=ω_{ice}. So the shift condition can be expressed by means of equation (4):

$$V_{shift} = \omega_{ice}\tau_{out}R_w \tag{7}$$

Observe that the shifting speed is equal to the geometric mean of IC and OC FMP speeds (equations (5) and (6)).

The gear ratios t1 and t2 are designed to limit the hydraulic units speed below the maximum values established by the manufacturer.

$$\tau_1 = \frac{\omega_{Hy\,Max}}{\max\left(\omega_{ICE}, \frac{1}{\tau_{out}}\frac{V_{max}}{R_W}\right)}; \quad \tau_2 = \frac{\omega_{Hy\,Max}}{\max\left(\omega_r \middle| \begin{matrix} V=0 \\ V_{shift} \\ V_{max} \end{matrix}\right)} \tag{8}$$

In the first equation, the denominator is the maximum speed that the shaft can assume respectively in the functional modes IC and OC. In the second equation the denominator is the maximum ring speed under the starting, shifting and maximum speed conditions. The hydraulic units have 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:

$$D_1 = \frac{2\pi\,T_1}{\Delta P_{max}};\, D_2 = \frac{2\pi\,T_2}{\Delta P_{max}} \tag{9}$$

$$\alpha_1\,\omega_1 D_1 = \omega_2 D_2 \;\text{ with } \alpha_1 < 1 \tag{10}$$

The torques T_1 and T_2 are the most severe conditions and must be identified considering the two following situations:

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

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3. Case study

The sizing method presented in the previous section will be applied to the case of an off-highway vehicle: the reach stacker. The vehicle is Kalmar Cargotec DRG Gloria 450 model, already described by Mercati [23], whose main characteristic data are summarized in table 1.

Table 1: Main data of the test vehicle.

Engine Data		Unit:	Vehicle Data		Unit:
Engine Model	TAD-1150-VE	-	Maximum Load	112400	Kg
Max power @ speed	235 @ 1450 - 2000	KW@rpm	Tyre Radius	0.825	m
Max torque @ speed	1581 @ 950-1450	Nm@rpm	Maximum Torque at Wheel	128000	Nm
			Max Vehicle Speed	26	Km/h
			Gradeability at rated load	17	%

The design procedure seen at the previous chapter requires the values of the two FMP speeds, which are unknown a priori. In the following, two approaches for a non-arbitrary choice of this speeds will be presented and discussed.

3.1 Optimal approach

The traditional design criterion of OC and IC transmissions refers to optimal solutions identified by the authors in [16]. In summary, the FMP of an IC layout must be positioned at about 30-35% of the maximum vehicle speed; while the FMP for the OC layout must be positioned around the maximum speed, in order to prevent the power recirculation mode, which is notoriously dissipative.
These suggestions applied to the IC2OC transmission lead to the following conditions:

-IC: FMP at 35% shift speed; but, as shown before, the shift speed is the geometric mean of the FMP speed and maximum speed of the vehicle, so the result is:

$$V_{fmp}^{IC} = 0.35 \cdot \sqrt{V_{max} \cdot V_{fmp}^{IC}} \;\rightarrow\; V_{fmp}^{IC} = 0.123 \cdot V_{max}$$

-OC: FMP positioned at maximum vehicle speed.

$$V_{fmp}^{OC} = V_{max}$$

3.2 Experimental-statistical approach

The value of the FMP speed can be chosen on the basis of a working cycle deduced from experimental measurements.
The spectrum of velocities, powers and frequency that characterize a specific vehicle can drastically influence the performances of the powertrain. For this reason, in the design of the IC2OC layout, the speed of the characteristic points (full mechanical points and the shifts between IC and OC) will be defined based on the expected working cycle of the reach stacker.
The reference cycle is described in detail in [23] and consists of three main phases (figure 4). In the first phase the vehicle reaches a pile of three containers of 10, 30, and 24 t respectively moving without load and at high speed. Then the three containers are moved from the first pile to another pile 50 m away and then moved back, resulting in six cycles of load – transport- unload – return.

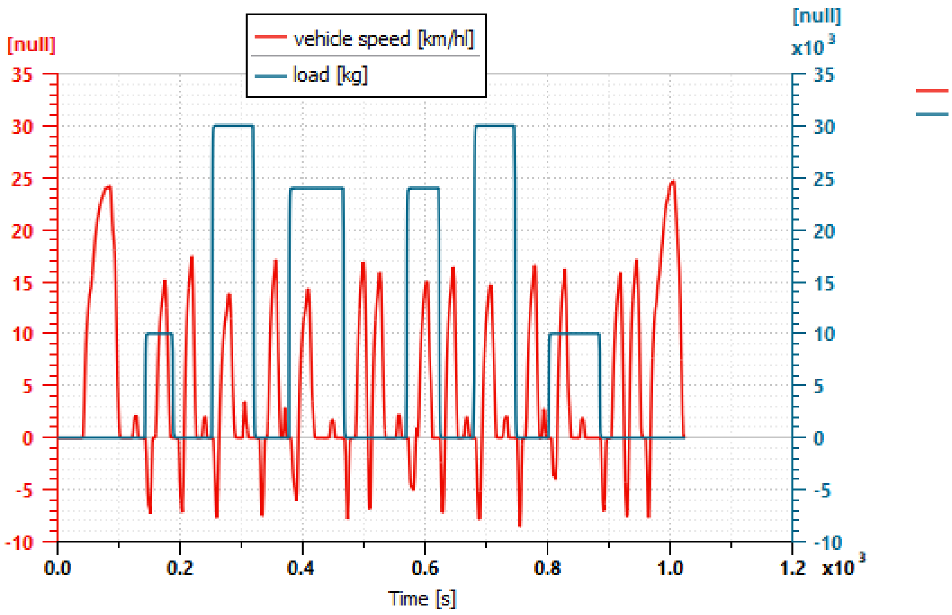

Figure 4. Working cycle

Data were acquired with a frequency of 100 Hz.
Raw data were first filtered with a robust linear regression using 1 s window. The actual power request at the wheel when then computed as follows:

$$P_{out} = \omega_{out}\, T_{out} \tag{11}$$

Which gives the work corresponding to the i-th acquisition by multiplying by the acquisition period:

$$W_{out}^{i} = P_{out}^{i}\, \frac{1}{100}[s] \tag{12}$$

The data were then ordered in terms of transmission ratio $\tau = \frac{\omega_{out}}{\omega_{in}}$. The transmission range of the vehicle was then discretized in intervals τ^j to obtain the frequency distribution f^j weighted on the transmission work as follow:

$$f_W^j = \frac{\sum_i\, W_{out}^{i}\,(|\tau_i|>\tau^j)(|\tau_i|<\tau^{j+1})}{\sum_i\, W_{out}^{i}} \tag{13}$$

The discretization of the transmission range has been performed using a step of 0.1 between the 0 and 1.5. The resulting curve is reported in figure 5.
The distribution shows that the first FMP speed must be chosen at about 50% of the maximum speed, while the second one must be chosen close to the maximum speed. The latter condition is also foreseen by the traditional sizing criteria of the OC transmissions, which suggest limiting the operation in power recirculation.

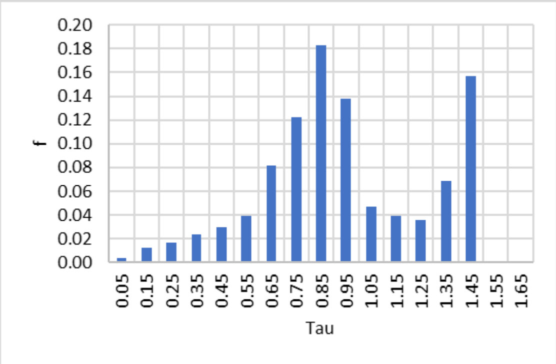

Figure 5. Distribution of the frequency weighted on the transmission work.

In the following, the transmissions designed according to the optimal approach and to the experimental statistical approach will be called respectively IC2OC_1 and IC2OC_2. The main parameters of the two transmissions are summarized in table 2.
For comparison purposes, also the IC and OC transmissions were sized based on the suggestions provided by [16]. Their main parameters are also shown in table 2.
It should be noted that the optimal approach leads to overall smaller displacements than those of the IC and OC, while the experimental-statistical approach leads to higher displacements and to a planetary gear with a ratio lower than 1. This fact could increase the energy losses in the transmission.

Table 2. Main design data of the transmissions.

		IC	OC	IC2OC_1	IC2OC_2
n_{ICE}	[rpm]	2000	2000	2000	2000
n_{Hymax}	[rpm]	3000	3000	3000	3000
Δp_{max}	[bar]	465	465	465	465
V_{max}	[km/h]	26	26	26	26
V_{fmp}	[km/h]	6	26	3.2	13
V_{shift}	[km/h]	/	/	9	18
D_1	[cm^3]	145	115	320	362
D_2	[cm^3]	340	480	120	254
τ_I	/	0.67	1	0.6	1.06
τ_I	/	1.28	0.5	1.51	0.6
τ_0	/	-1/2	-1/2	-1/1.5	-1/0.4
τ_{out}	/	26	71	60	34

3.3 Simulation model

The four transmissions were modeled using Amesim [26]. Figure 5 shows only the model of the reach stacker with IC2OC transmission. 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 model experimentally derived [21]. Losses in ordinary gears were defined by an average efficiency equal to 0.98, while for the epicyclical gears, the Benedict and Kelley [26] method was assumed for each pair of engaged gears. In figure 6 is shown the work cycle, which was derived from experimental data

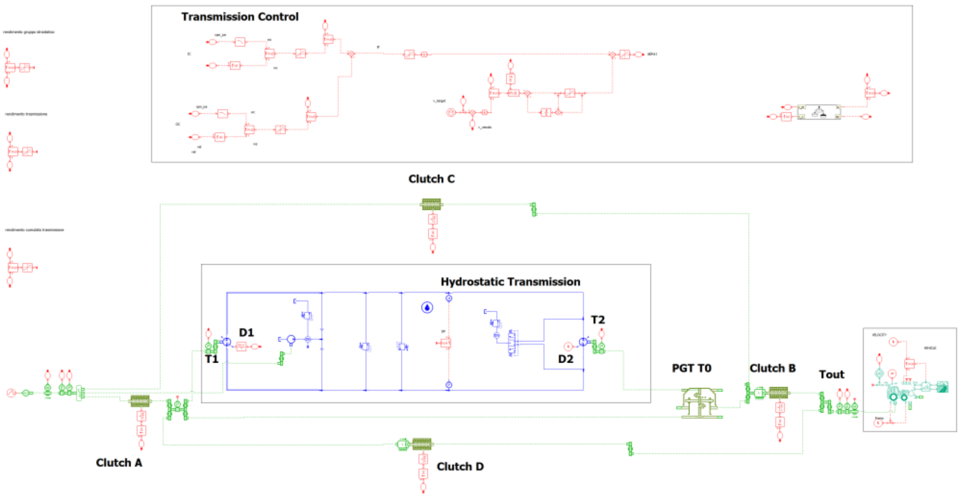

Figure 6. Model of the reach stacker with IC2OC transmission.

3.4 Results and discussion

Figure 7 summarizes the energy consumption of the four transmissions examined in the previous points. As can be seen, the IC transmission shows the highest consumption; compared to it, the OC and the IC2OC_1 have respectively lower consumptions of 10.5% and 13.9%. This confirms what has already been found in [25] with constant and maximum load conditions, even if now the consumption differences are more marked than those (5.2% and 9% respectively). This underlines once again the importance of the precise definition of the loads in the simulation of these machines.

It is interesting to compare the behavior of the IC2OC_2 and the IC transmissions.

At the start, where average speeds are high, the former works in conditions close to its two FMP, while the IC works in areas far from its unique FMP, i.e. in areas of lower efficiency. This results in a lower consumption curve for the IC2OC in the first 150 seconds with respect to the IC. Instead, in the intermediate area of the working cycle, where the average speeds are lower, the IC2OC_2 works more often in power recirculation. At the same time the IC works more frequently in conditions close to its FMP. As a consequence, the consumption of the former grows almost to reach that of the CI.

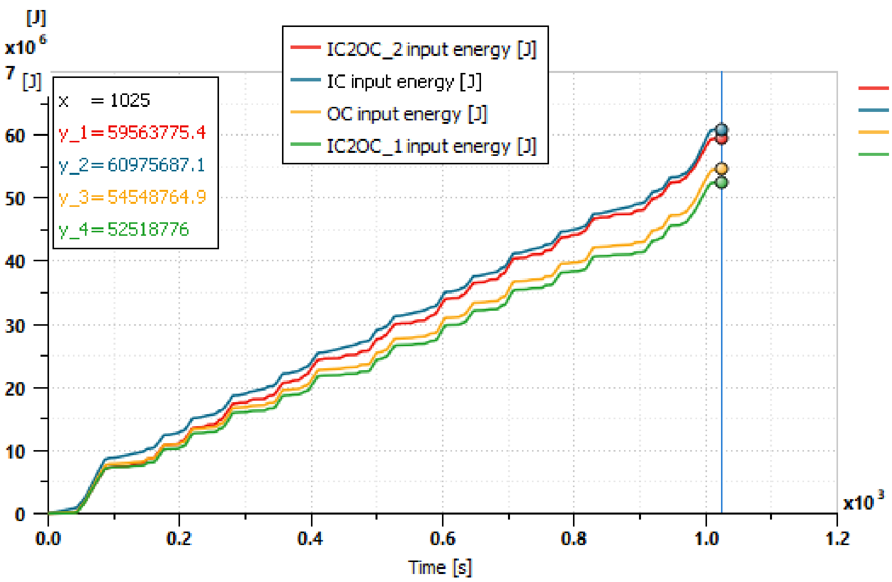

Figure 7. Input energy to the for transmissions.

The transmission IC2OC_2, on the other hand, shows higher consumption than IC2OC_1. This may seem contradictory given that the experimental-statistical approach places the FMP, the highest efficiency point, right on the conditions of greatest use of energy.
This discrepancy can be justified by two main reasons.
First of all, it must be observed that moving the FMP towards high speeds means forcing the IC to work in recirculation mode in wider ranges. Since recirculation is less efficient, lower operating efficiency must be expected. In other words, the wide range in recirculation cancels the advantage due to the positive action resulting from having placed the FMP at the highest frequency.
Furthermore, high FMP speed values mean high displacements, as shown in table 2. If, on the one hand, larger displacements imply lower pressure levels and lower losses, on the other, they imply greater leakages and greater mechanical losses, due to the increased dimensions of the internal areas of the machine.
To clarify these points, simulations of the IC2OC transmission were carried out for different values of the FMP speed values, as summarized in table 3.

Table 3. Performance of IC2OC transmission for different value of FMP speed.

V_{FMP} [km/h]	13	6.2	5	3.2	2.8	1.5
τ_0	-0.4	-1.05	-1.2	-1.5	-2	-3
τ_{out}	34	48	55	60	70	95
τ_1	1.06	0.748	0.65	0.6	0.51	0.38
τ_2	0.6	1.51	1.51	1.51	1.51	1.51
D_1 [cm^3]	362	253	268	298	332	377
D_2 [cm^3]	254	118	112	110	105	88
E_{input} [MJ]	59.56	54.07	53.58	52.51	53.66	54.72

The table 3 shows that as V_{FMP} decreases, consumption decreases, confirming what was said about the effects of recirculation. However, beyond the value of 3.2 m/s, consumption slightly increases, due to the increase in losses resulting from the higher displacements of the machines.
These results suggest that the statistical method cannot avoid also considering the behavior of the transmission. Therefore it should be refined by weighing the distribution in equation (13) with both the energy required by the vehicle and the efficiency of the transmission. This will be the next step of this research.

4. Conclusions

The sizing of the IC2OC hydromechanical transmission requires the definition of two particular vehicle speeds, in which the power inside the transmission is transmitted only mechanically: the full mechanical point speed. Since the values of these speeds heavily influence the performance of the transmission, they must be carefully chosen. In this work, two selection criteria for the aforementioned speeds have been proposed and discussed. The first is based on the optimum conditions for each of the two configurations that make up the IC2OC transmission; the second is based on the concept of correspondence between the FMP speed (the highest efficiency point) and the speed of the most frequently used vehicle. In this work the latter speed has been replaced with the speed that processes the most energy. To obtain it, data from an experimental field campaign were used. These two criteria were applied to the IC2OC transmission designed for a reach stacker. The vehicle, equipped with two transmissions and the two basic IC and OC transmissions, was modeled in the Amesim environment, and a working cycle measured in the field was applied to it. The energy consumption of the transmission based on the first criterion was found to be the lowest compared to both basic solutions, confirming the potential of the IC2OC solution. On the other hand, the transmission based on the second criterion was not particularly efficient, partly contradicting the basic hypotheses of the criterion itself. In effect, this criterion pushes the FMP value for IC operation towards high values, forcing the transmission to work more frequently in power recirculation, i.e. in an area with less efficiency. For the refinement of this criterion, therefore, it will be necessary to take into account both the estimate of the energies processed and the estimate of the transmission efficiencies.

Acknowledgments

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