JFPS International Journal of Fluid Power System
This article is a selected paper from the 12th JFPS International Symposium on Fluid Power, Hiroshima, Japan in October 22-25, 2024
18-1, 29/36, 2025
doi.10.5739/jfpsij.18.29

Design of Power-Split Hydrostatic CVT Agricultural Transmission for Enhanced Control and Efficiency

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Received 23 December 2024 / Accepted 2 March 2025

A newly conceived fully synchronized and full power-shift Output Coupled transmission for agricultural tractors was designed, based on the hypothesis of ideal components. This first version was simulated in a lumped parameter environment, including all the components of the transmission, the real hydraulic machines efficiencies, and a finite state machine based control system, to command hydraulic machines displacement and engagement/disengagement of clutches and synchronizers. The hydraulic transmission considered in the work, is subjected to challenging operating conditions, which cause deviations from the ideal command profiles due to volumetric and hydromechanical efficiencies. In the second phase, the parameters of the transmission are optimized, obtaining an upgraded design. The new set of parameters substantially increase the performances of the hydrostatic units, moving the operating points to favourable conditions. The latter design is finally compared to the first one, focusing on the performance of both the hydrostatic units and the overall interconnected system.

Keywords: Power-Split, Continuously Variable Transmissions (CVT), Hydrostatic Transmissions, Lumped Parameter Simulation.

1. Literature overview

In recent years, Power-Split CVT transmissions have gained popularity for their mass production in hybrid passenger cars, allowing the traction force to be transmitted either by mechanical or electrical power; this industrially relevant topic has fostered a quite large numbers of scientific papers on the Hybrid Electric Power-Split architectures especially for the automotive field.

Historically, the Power-Split concept had been commercially available in agricultural tractors since late 90es, however, the architectures for agricultural tractors have obviously different requirements with respect to automotive, in particular, high torque capability and relevant operating conditions at lower speeds.

In contrast to the large hybrid electric vehicle literature, the hydraulic Power-Split is less rich. A ground-breaking and forward-looking work by Kress[1)] introduces the main classification of Hydrostatic Power-Splitting architectures, Input Coupled (IC), Output Coupled (OC) and Compound Architectures with basic theory of operation, still valid today.

Linares et al.[2)] take up these concepts, offering an explanation of Power-Split CVTs and describing the design parameters, obtaining a general diagram for the design of the transmissions.

An important overview and history of CVT technology is provided by Renius[3)], where the main transmission types are introduced, referencing to the commercial applications of these concepts.

In the next literature, we report only the papers focused on hydraulic solutions; this technology, offering a good trade-off in terms of cost, robustness and controllability, is the most common among agricultural CVT transmissions.

Output Coupled, Input Coupled, Compound Coupled and dual stage Input Coupled architectures are compared by Carl et al.[4)] via the Matlab/Simulink software, using custom built component libraries, for applications such as wheel loaders and busses with a focus on efficiency, control effort, and system complexity.

Rossetti et al.[5)] propose an optimisation method of the average loss, calculated between zero and maximum vehicle speed. A two-range Power-Split agricultural tractor is considered, where the architecture, displacement of hydraulic machines and mechanical ratios are the degrees of freedom.

Additionally, dual stage Input and Output Coupled commercial architectures are considered by Schembri Volpe et al.[6)]. More in detail, the authors present an optimization procedure in order to achieve a significant reduction in power recirculation and an increase in mechanical efficiency.

Xia et al.[7)] show a new dual stage Power-Split architecture, which includes two Input Coupled planetary gear trains; moreover, the working principle, the basic characteristic formulae derived in this study for speed ratio, torque, and efficiency are reported.

Martelli et al.[8)] put forward a complete co- simulation of a commercial Power-Split tractor, based on a hydraulic Input Coupled and Compound architecture. The simulation takes into account the engine, the transmission, the multibody behaviour of the vehicle and the complex tyre-soil terramechanics.

Finally, Marani et al.[9)] show the comparison between a current production tractor Power-Split Input Coupled architecture and an Output Coupled designed and set up by the authors. The new architecture features Power-Shift characteristics, however, the transmission total efficiency plots highlight distortions at the extreme low displacements of the pump/extreme low speeds of the hydraulic motor.

The low speed behaviour of hydraulic machines is a well-known issue for fluid power industry specialists, which was scientifically investigated by Zarotti[10], Achten[11] and Mommers[12]. Zarotti[10] theoretically demonstrated that for near zero speed the volumetric and hydro-mechanical efficiency are characterized by an indeterminacy zone in these spots, essentially because the leakage and friction are non-zero at zero speed. Achten et al.[11] et al. experimentally studied torque losses at extremely low operating speeds, stating that "at near-zero operating speeds, the pump foremost needs to compensate the volumetric losses, i.e. leakage of the hydraulic motor to be tested". Finally Mommers et al.[12] measured the efficiency of different volumetric pumps, among which axial piston pumps, confirming the existence of a low operating speed indeterminacy zone.

In the present paper, the same architecture presented in Marani[9] will be equipped with a proper transmission electronic control and the drawbacks in terms of control deviation caused by low efficiency will be analysed and fixed.

2. Nomenclature

N_R	:	Number of teeth of the gear connected to the first hydraulic unit
N_{S1}	:	Number of teeth of the gear connected to the transmission Input
N_{S2}	:	Number of teeth of the gear connected to the even gear shaft
α_{HM}	:	Hydraulic Motor Displacement
α_{HP}	:	Hydraulic Pump Displacement
α_{HT}	:	Hydrostatic transmission command
τ_1	:	Ratio of First Planetary Gear Set
τ_2	:	Ratio of Second Planetary Gear Set
τ_{HM}	:	Hydraulic Motor Shaft Ratio
τ_{HP}	:	Hydraulic Pump Shaft Ratio
$\tau_{F1,3}$	:	First/Third Gear Ratio
$\tau_{F2,4}$	:	Second/Fourth Gear Ratio
$\tau_{T,req}$	:	Transmission ratio request
ω_E	:	Input of the Transmission Shaft Speed
$\omega_{o1,3}$	:	Output Speed, First/Third Gear
$\omega_{e2,4}$	:	Output Speed, Second/Fourth Gear

3. CVT, Power-Split and Power-Shift

The simplest CVT architecture is represented by a hydrostatic transmission, directly coupled in between the engine and the output of the transmission, however this solution suffers from two important drawbacks. The first is the relatively low efficiency of the hydraulic units, the second is the size of the hydrostatic transmission that should be able to match the engine power, and therefore, relatively large volumetric units should be used.

Power-Split architectures were developed to overcome these disadvantages, in fact in Power-Split architectures the engine power is divided into two distinct paths: fixed ratio mechanical path and infinitely variable path. In the fixed ratio path, the power is efficiently transmitted via mechanical components such as gears; in the infinitely variable path, a continuous variable transmission unit is provided to offer the CVT feature.

The infinitely variable transmission can be obtained by different technologies: mechanical, via chain/belt variator or toroidal variator, electrical, with an electric generator/motor couple or hydraulic, via the rather conventional closed loop hydrostatic transmissions.

The Power-Split architectures[1] can be classified in three categories: Input Coupled, Output Coupled or Compound. Input Coupled is characterized by the coupling of one of the hydraulic units (i.e. the hydraulic pump, HP) to the input to the transmission, while the second unit (i.e. the hydraulic motor, HM) is attached to one branch of the Planetary Gear Train (PGT). In all the architectures previously listed one of the units is conventionally referred to as hydraulic pump and the other as hydraulic motor; however it should be noted that in the effective working range both units functionally switch between pumping and motoring operation.

In Output Coupled transmissions one hydraulic unit (i.e. the hydraulic motor) is attached to the output of the transmission,

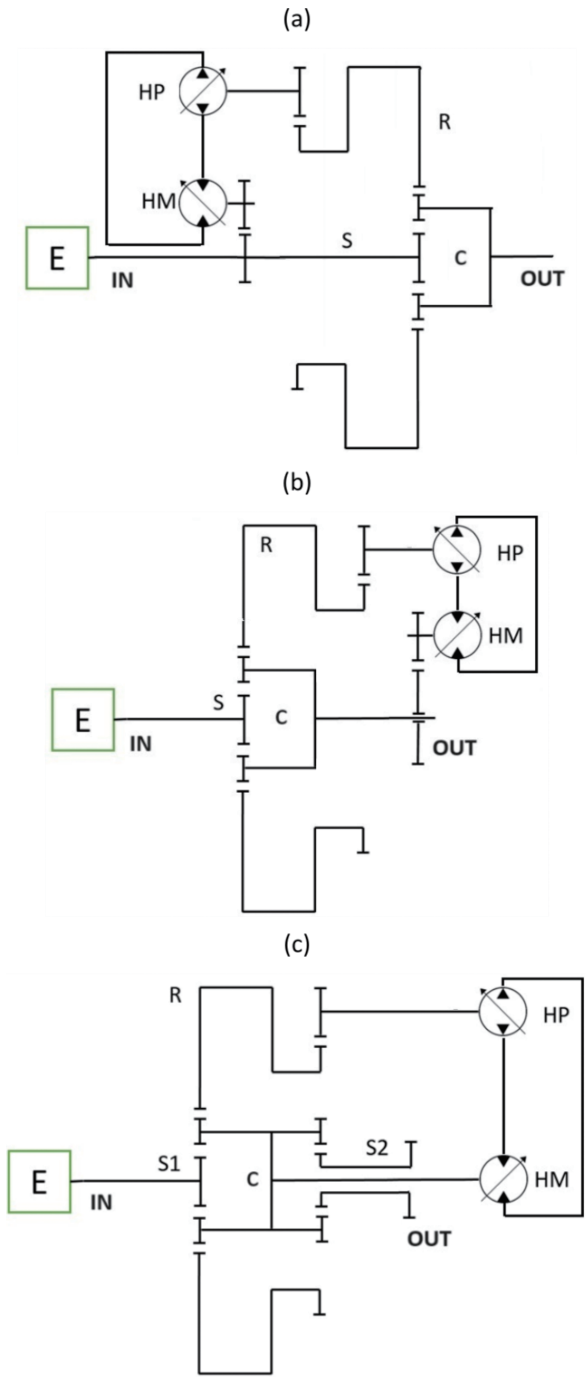

Fig. 1 (a): Input Coupled Power-Split transmission, (b): Output Coupled Power-Split transmission, (c): Compound Power-Split transmission whereas the second unit (i.e. the hydraulic pump), providing for the speed control, is connected to one of the planetary shafts.

In Compound architectures four active shafts are provided by a compound planetary, in which one member is common to the two stages (the planet carrier, generally), one shaft is for the input, one for the output and the two remaining are connected to the two hydraulic units.

For reasons of efficiency and sizing, tractor CVT transmissions need to be equipped with a discrete number of mechanical gears, just like their non-CVT counterparts, especially for open field heavy duty models; in general, the higher the number of gears, the smaller the hydrostatic units are, and consequently, the smaller is the power flowing into the variable hydraulic path.

Another important characteristic for tractors is the so-called Power-Shift feature, which is the ability to avoid the interruption of torque transmission over the shifting. Looking at Fig.1 (a) and 1 (b), adding a series of mechanical gears downstream of the output shaft, would necessarily produce a non-Power-Shift transmission scheme. In fact, with a single PGT, during a gear shift, the hydrostatic motor must adjust its speed to the incoming gear (e.g. in an upshift, from the maximum speed of lower gear to minimum speed of the next gear) by changing accordingly its displacement, thus generating a torque discontinuity in the transient. Therefore, to achieve the Power-Shift feature, a second planetary gear is necessary, obtaining a Compound Architecture, such as the proposed architecture, which will be presented in the next paragraph.

4. Baseline transmission set up

The compound architecture considered in this paper is represented in Fig. 2, with odd and even gears on the first and second stage of a compound PGT. The resulting transmission can be classified as Output Coupled architecture for the odd gears and Compound Architecture for even gears. This configuration opens up the possibility to obtain a full Power-Shift architecture.

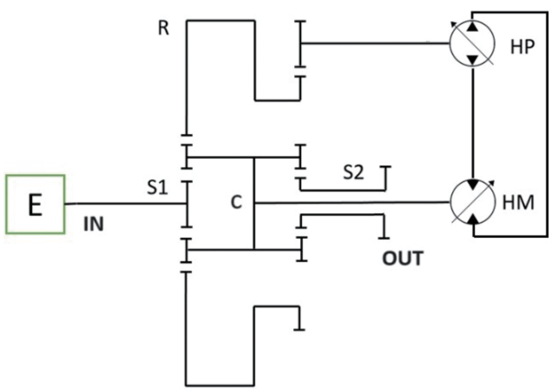

Fig. 2 Compound Architecture proposed by the authors.

For the Power-Shift feature, the ratios of the two PGTs must be opposite in sign, so that a positive increase in the hydrostatic transmission (HT) output speed corresponds to an increase/decrease of the transmission output speed for odd/even gears. In other words, the output speed will be an increasing function of hydrostatic command for odd gears and a decreasing function for even gears. With proper final reduction, these two functions will cross in a specific point (i.e. the synchronism point), in which we can theoretically shift from one gear to the other adjacent with minimum torque discontinuity, because the hydraulic units displacement and the odd and even shafts speeds are perfectly matching.

The proposed architecture features two identical hydraulic units capable of pump/ motor operation with variable displacement, from zero to maximum displacement on both the units; conventionally the unit on the output is labelled HM and the one on the ring of the PGT is labelled HP.

It is possible to write down speed equations to drive the design of the system. Before going on with the equations, it is important to note that the general schemes proposed in the figures serve as a reference, but can be generalized to obtain alternative architectures. Here we chose to connect the first sun (S1) to the input, the ring (R) to the hydraulic unit, the carrier (C) to the odd gears and the second sun (S2) to the even gears, but the same behaviour could be achieved with switched connections as well, since the same equations would still apply, once the transmission rations of the two PGT stages are defined. The first one is the number of teeth of the gear connected to the hydraulic unit (R in the scheme) over the number of teeth of the gear connected to the Input (S1); the second one is the ratio of number of teeth of the gear connected to even gears (S2) over the input shaft gear (S1). Please note that the number of internal/external teeth comes with negative/positive sign.

$$\tau_1 = \frac{N_R}{N_{S1}} \tag{1}$$

$$\tau_2 = \frac{N_{S2}}{N_{S1}} \tag{2}$$

Then, using the Willis formula, we can express the output speed in odd gears for ideal, unitary efficiency machines:

$$\omega_{o1,3} = \tau_{F1,3} \cdot \frac{\omega_E}{1 - \tau_1 + \frac{\tau_1}{\tau_{HP}\tau_{HM}} \cdot \frac{\alpha_{HM}}{\alpha_{HP}}} \tag{3}$$

Or, in even gears:

$$\omega_{e2,4} = \tau_{F2,4} \cdot \omega_E \left(\frac{1}{\tau_2} - \frac{1-\tau_2}{\tau_2} \cdot \frac{1}{1 - \tau_1 + \frac{\tau_1}{\tau_{HP}\tau_{HM}} \cdot \frac{\alpha_{HM}}{\alpha_{HP}}} \right) \tag{4}$$

By hypothesis, a sequential command is imposed on the hydrostatic transmission: the pump and motor start at null and full displacement, respectively, this condition corresponding to hydraulic motor standstill. Then, the pump displacement is increased until saturation at 1, with constant full motor displacement; finally, the motor displacement is decreased to 0, with constant full pump displacement. It is important to note that the motor can be brought to null displacement because the pump speed is not constant, instead it is decreasing with motor displacement, with 0 motor displacement corresponding to 0 pump speed.

As already mentioned, the speed of odd gears is increasing with the hydrostatic transmission command while the speed of even gears is a decreasing function. This property makes it possible to synchronize the two gears with proper gear ratios (τ_1, τ_2, τ_{F1}, τ_{F2}, τ_{F3}, τ_{F4}): the maximum speed obtained by a certain value of command on a particular gear, corresponds to the minimum speed obtained by the same command on the next gear.

In the choice of transmission parameters, the energy aspect must be taken into account. Many papers[1)-3)] point out that, given that the power transmitted by hydraulics is variable during the speed regulation, it is in general more favourable to work in conditions of low hydraulic power usage, i.e. with most of the power being transmitted via the mechanical path, because the hydraulic path has a significantly lower efficiency.

According to this reasoning, the best efficiency is theoretically found in the full mechanical points or lockup condition, which is defined in Kress[1)] as the condition in which all the power passes through the mechanical path and consequently the hydraulic power is null. The above condition is obtained in two distinct control states: pump at null displacement and motor at full displacement and, symmetrically, pump at full displacement and motor at null displacement; in the first and in the second case the motor and the pump are at standstill, respectively.

Obviously, the speed regulation starts at standstill in first gear with the hydraulic pump at 0 displacement and hydraulic motor at 1, which is a first lockup condition.

As a first setup, we placed the synchronism between first and second gear and between third and fourth at the maximum command (pump at 1 and motor at 0), thus requiring a lockup condition at the end of the first and third gear and at the beginning of second and fourth.

Unlike for the previous cases, the synchronism between second and third gear should be located in the middle of the control range and not at lockup, because in third gear the first lockup point would correspond to zero speed, according to equation 3. The control map of the baseline model is shown in Fig. 3, with synchronism points placed according to the previous discussion.

5. Lumped Parameters Simulation

All the simulations were carried out in Simcenter Amesim[13)], a commercial software that has a couple of advantages with respect to self-built software. The first is the convenience of having a large

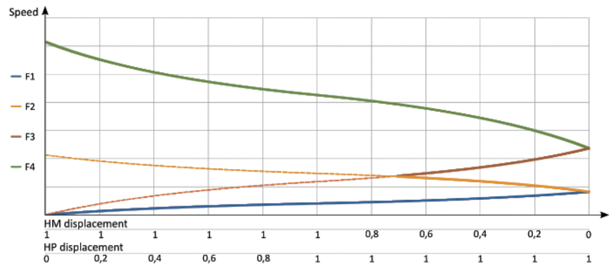

Fig. 3 Control Map of Baseline Model

library of the most common elements used in fluid power and automotive field, the second is the ease of communication and portability to industrial partners who have adopted the same software.

To objectively highlight the performance of the transmission, an ideal engine and an ideal load were set up, in this way it is possible to separate the effects of the transmission to the characteristics of the external environment. Once the design is validated, it will be useful to use realistic engine and load characteristics such as in Martelli et al.[8)].

For this study, the engine is a constant speed ideal motor and the load is a constant power with torque saturation resisting torque. The load starts at low speed at maximum torque linking up to a constant power hyperbole as the speed increases over the maximum torque and maximum load power point.

The core of the model is essentially composed of three parts: the mechanical transmission, the hydrostatic transmission and the electronic control.

The mechanical transmission, illustrated in Fig. 4, is made of the compound planetary with shared carrier, followed by the four gears with the respective synchronizers. First and third gear belong to the odd intermediate shaft, while second and fourth are connected to the even intermediate shaft; the two shafts are connected to the output shaft via the final pinion reduction thanks to a dual clutch. The transmission mechanical losses were concentrated to the final reduction and even and odd shafts using the data of an existing commercial transmission previously studied[8), 9)]. Additionally, the actuation of the synchronizers and clutches were simulated through linear mechanical components and hydraulic actuators.

The hydrostatic transmission is quite nonconventional, since it composed by two symmetric hydraulic machines both capable of functional reversing (pump/motor operation is allowed) and zero

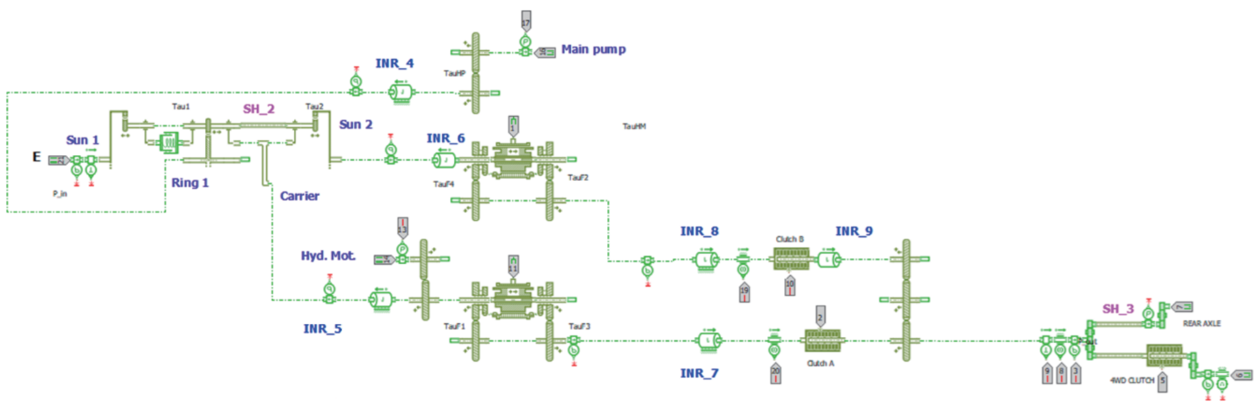

Fig. 4 Mechanical transmission sub-model

displacement control. The circuit is complemented by the boost pump, pressure relief valves and the flushing valve. The volumetric and hydro-mechanical efficiency of the volumetric machines were provided by the hydrostatic units' manufacturer. These efficiencies are defined by a matrix of operating points featuring 13 displacements, 21 speeds and 21 pressure levels.

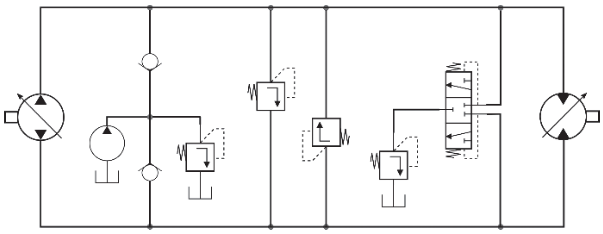

Fig. 5 Hydrostatic Transmission sub-model.

The electronic control outputs a single command for the hydrostatic transmission (α_{HT}), ranging from 0 to 2, as a function of the operator's input (requested vehicle speed).

α_{HT} is then translated to the corresponding individual displacement settings (in the range [0, 1]) for the 2 hydraulic units, (α_{HP}, α_{HM}), to implement a sequential control:

$$\alpha_{HP} = \max(\min(\alpha_{HT}, 1), 0) \quad (5)$$

$$\alpha_{HM} = \max(\min(2 - \alpha_{HT}, 1), 0) \quad (6)$$

Given the active gear, the engine speed and the relevant vehicle parameters (wheel rolling radius and final drive ratio), the requested vehicle speed is converted to a requested transmission ratio, and the control output is simply calculated, in open-loop, via a look-up table based on the map in Fig. 3.

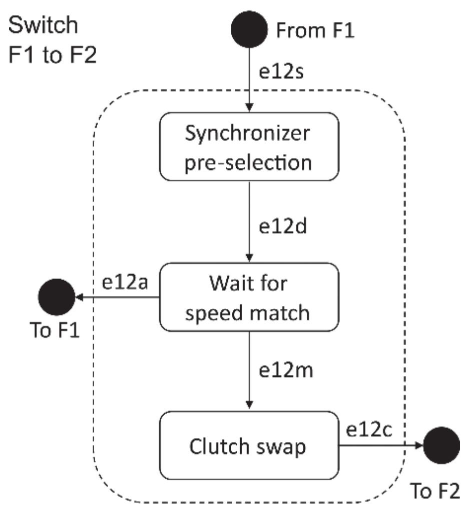

Fig. 6 Electronic control FSM – Internal FSM for state "Switch F1 to F2"

To compensate for system non-idealities (neglected by the open-loop map), we added a speed feedback on the odd- and even-gear output shaft, to detect the actual point of speed match and trigger the clutch swap for a smooth gear shift. The overall control is managed by a top-level finite state machine (FSM), which includes stable states *Fx* (where *x* is the forward gear number) and transition states *Switch Fo to Fi* for gear shifts.

All *Switch Fo to Fi* states (*e.g. Switch F1 to F2*) implement an internal, second-level FSM, as the one in Fig. 6.

With the vehicle in *F1* and accelerating (i.e. the operator is increasing the transmission ratio request, $\tau_{T,req}$), when the ratio request exceeds a predefined threshold value, event *e12s* triggers the transition from *F1* to *Switch F1 to F2*. In the *Synchronizer pre-selection* sub-state, the *F2* position is pre-selected in the *F2-F4* synchronizer (according to the possible upcoming shift to *F2*); then, when the pre-selection is done (event *e12d*) the sub-state is switched to *Wait for speed match*.

At this point, if the operator requests a deceleration and $\tau_{T,req}$ drops below the threshold (event *e12a*), the gear shift is aborted; otherwise, when the speed difference between the two shafts drops below a given threshold (speed match, event *e12m*), the sub-state is switched to *Clutch swap*, clutches *A* and *B* are swapped, following the predefined time-dependent curves (for the engaging and disengaging clutch). Finally, at the end of this operation (*event e12c*), top-level state *F2* is reached.

The closed-loop compensation is implemented as a second control output, adding to the base open-loop map; it's an exponentially decaying function of time, so that at steady state only the open-loop control is active.

The speed difference threshold for speed matching is set to 50 rpm.

The same applies to all the 6 possible gear shifts: *1* and *2* just need to be replaced by the relevant gear numbers; for downshifts, with respect to the previous upshift example, "exceeds" must be replaced by "drops below" (and vice versa), while "deceleration" becomes "acceleration".

6. Results of baseline model

The test on the transmission is carried out by increasing linearly the requested transmission ratio from zero to a maximum vehicle speed of 45 km/h, then the vehicle is symmetrically decelerated until standstill. The control is based on the mapping shown in Fig. 3, starting from pump at zero displacement and motor at full displacement, at the end of gear 1, the theoretical synchronism with gear 2 is reached at null hydraulic motor displacement and full pump displacement, which is a full mechanical point. Then, the control action is reversed increasing the motor displacement until the next shift. The synchronism between gears 3 and 4 is analogous to the one between gears 1 and 2.

In Fig. 7 we can identify the requested tractor speed in dotted pink, the actual speed in red, and the external load in blue, with the dashed vertical lines indicating the transitions between the even-odd gears.

The speed output correctly matches the requested speed in the central range of gears, but important discontinuities takes places at shifting, between first and second and especially between third and fourth.

To investigate this behaviour we focus on the analysis of the shift from third to fourth gear, which highlights the most pronounced discontinuity.

Before the shift, to reach the end of the third gear, with pump at full displacement, the hydraulic motor displacement must be decreased towards zero; consequently, the motor increases its velocity until maximum speed and the pump decelerates until standstill. This condition corresponds to one of the full mechanical points, which are, in theory, optimal efficiency points.

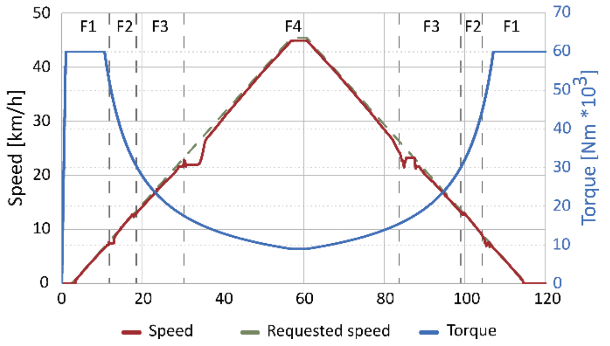

Fig. 7 Baseline Design simulation results: speed and load

After the clutches swap at about 30 s, the fourth gear is engaged and the control action is reversed with the pump accelerating and the motor decelerating.

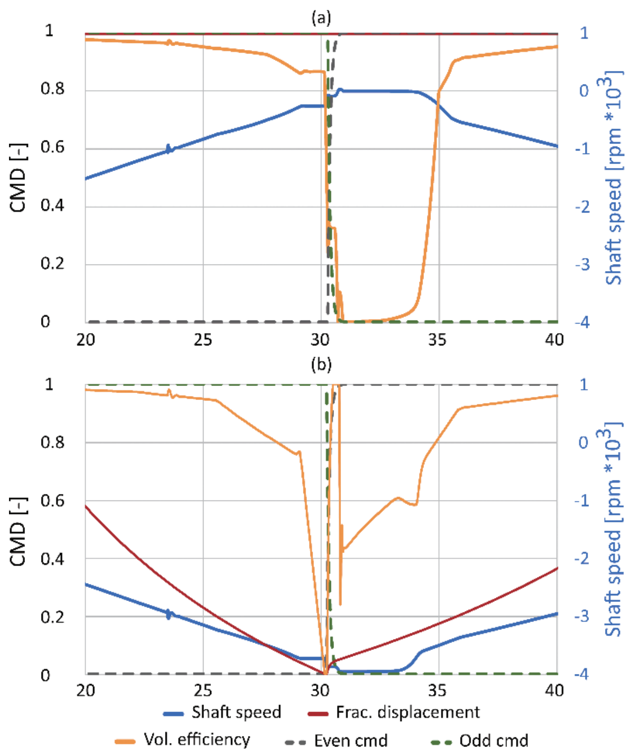

Fig. 8 Baseline Design 3-4 Clutches Swap: (a) Hydraulic Pump behaviour, (b) Hydraulic Motor behaviour

Figures 8 (a) and 8 (b) represent the pump and hydraulic motor behaviour during the third to fourth gear change; the most relevant parameters to be monitored are the displacement (red), the speed (blue) and the volumetric efficiency (yellow).

In Fig. 8 (a), we can note that after the clutches swap there is a functional switch (the pump acting as motor and vice-versa) and the speed drops to zero. In this condition, the pump has zero volumetric efficiency, the electronic control increases the hydraulic motor displacement to increase pump speed, but the high volumetric losses hampers the pump movement, causing an unwanted speed plateau, which explains the behaviour highlighted in Fig. 7.

In Fig. 8 (b), as the displacement decreases, the motor is accelerated and reaches the maximum speed before the clutches swap. The volumetric efficiency decreases with the displacement, converging to zero displacement/zero efficiency, at the swap. After the swap, the functional and control reversing occurs, the displacement is increased, decreasing the motor speed, and, as the efficiencies improve, the behaviour gets more linear and regular. The behaviour highlighted by the baseline design is clearly unacceptable from the functional point of view; for this reason, an upgraded design is proposed in the next chapters.

Finally, it should be remembered that the validity of these results are subjected to the faithful representation of efficiencies, especially at low speed, which were experimentally collected by the manufacturer. However, the validity of this data can be reconfirmed by the consistency with the efficiency plots shown in Mommers[12].

7. Optimized design and comparison

The results of previous simulations inspired the upgrade of the baseline design. The new design is obtained by shifting the theoretical synchronism point from the boundary points of hydrostatic control range (one of the machines at max displacement and the other at zero) to a point within the range. After different trials, the value chosen was at 10% of motor displacement, with pump at full displacement. This means reducing the control range, but, on the other hand, it allows to operate with a better efficiency. To do that, the third and second gear ratios were increased, while first and fourth gears remained unchanged to ensure the rated starting torque and maximum speed, respectively. The choice of these new synchronism points is the result of a trade-off between two requirements, the first is the avoidance of low efficiency spots, the second one is the decrease of the control range and increase of ratio of second and third gear. Increasing the gears' ratio raises consequently the torque and the pressure on the hydrostatic transmission, and since the transmission was designed to have the smaller possible hydraulic units, the hydrostatic units will be working near to the torque/pressure operative limits but they absolutely shouldn't exceed them.

Then, as a feasibility check, the pressure on hydrostatic transmission will be later evaluated in simulation, to ensure that the load is within the maximum allowed pressure.

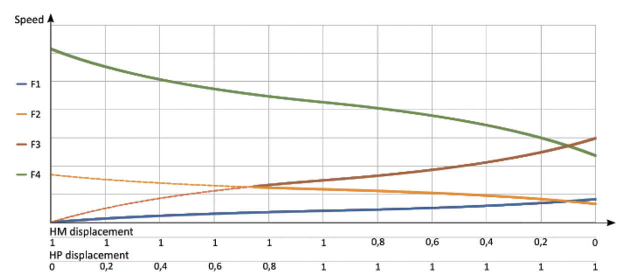

Fig. 9 Control Map of Optimized Design

The modified mapping of the transmission ratio control is shown in Fig. 9; please note that, in contrast to baseline, the theoretical point of first-second and third-fourth synchronism points is found at 0.1 hydraulic motor displacement, as said before.

Figure 10 is the analogous of Fig. 7, highlighting a better speed control, with a minimal bump on the shift, which can be easily corrected by electronic control fine tuning, not presented here but to be scheduled in next steps; in fact, for the sake of fair comparison, we decided to keep the same control of baseline case.

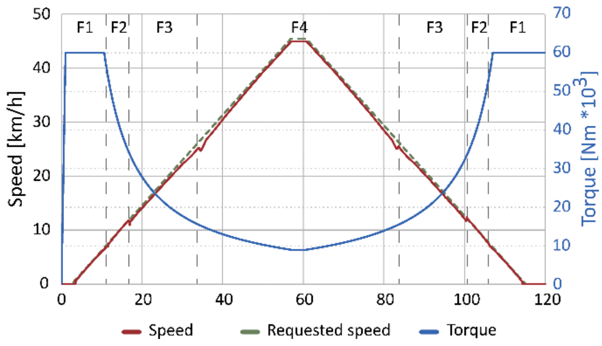

Fig. 10 Optimized Design simulation results: speed and load

To confirm the correctness of the method it is possible to notice, from Fig. 11 (a) and 11 (b), that the control of the hydraulic machines can be managed by the control algorithm, to obtain a smooth shift control. In particular it is noteworthy that neither the pump speed, nor the hydraulic motor displacement reach zero; therefore the volumetric efficiency is kept at acceptable levels from the control point of view.

Looking at the hydraulic efficiencies in the full speed range, it is possible to note that in the baseline case (Fig. 12 (a)), the efficiencies drop at the synchronism because in the full mechanical points one of the hydraulic units works at zero speed, which is a zero efficiency condition, which in turn causes control deviations.

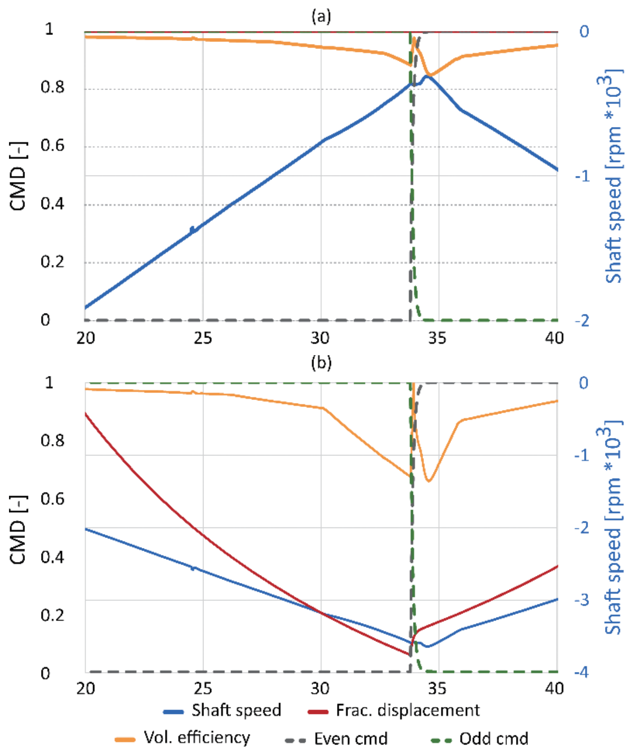

Fig. 11 Optimized Design 3-4 clutch swap: (a) Hydraulic Pump behaviour, (b) Hydraulic Motor behaviour

On the other hand, Fig. 12 (b) highlights the efficiency variation of the optimized design, avoiding extremely low volumetric efficiency spots, thus allowing a satisfactory speed control.

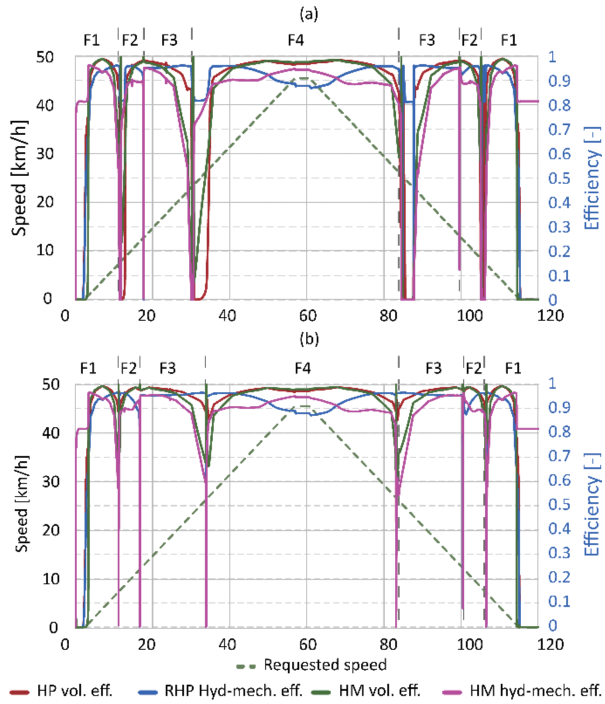

Fig. 12 (a): Baseline Design, full range efficiencies plot, (b) Optimized Design full range efficiencies plot

Additionally, a careful check must be done on the hydrostatic transmission load since, in the optimized design, we increased the second and third gear ratios. This change increases the torque load on the hydraulic units, thus the hydrostatic transmission plot can give more insight of the transmission operation, as shown in Fig. 13. The

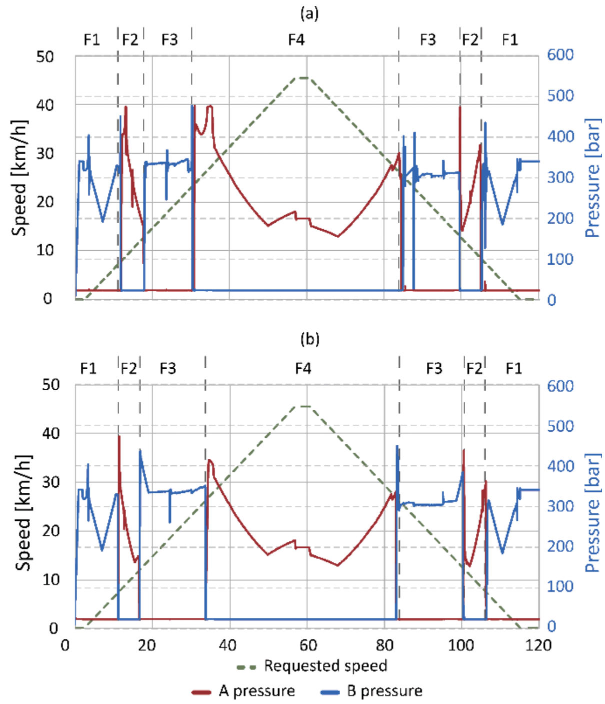

Fig. 13 (a) Baseline Design, Hydrostatic Transmission pressure plot, (b) Optimized Design Hydrostatic Transmission pressure plot plots of Fig. 13 (b) highlight an increase in hydrostatic pressure in second and third gear, but the pressure range is still below the pressure threshold of the relief valves.

A final assessment has to be made on the transmission efficiency, in particular, to check if the relocation of synchronism from full mechanical points can adversely affect the total efficiency.

The plot in Fig. 14 confirms that the total efficiency of the optimized design is overall better than baseline; in fact, the sacrifice of the full mechanical points is more than outweighed by a significant increase in hydrostatic transmission efficiencies. In particular, is it possible to note that, during the run up phase, at the shift to second and to fourth gear, the efficiency is significantly better in the optimized transmission.

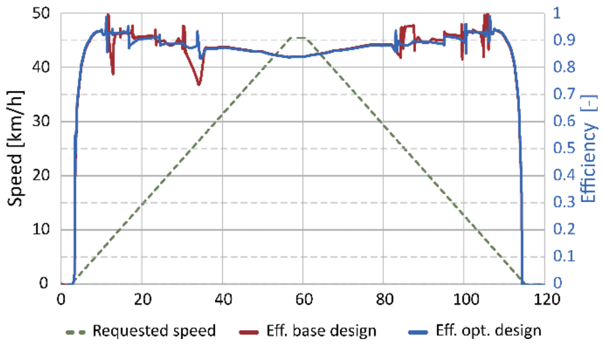

Fig. 14 Total Efficiency of Baseline and Optimized Transmission.

8. Conclusion

A Compound Output Coupled Power-Shift transmission, provided with an appropriate control algorithm, was introduced and simulated; firstly, the gear ratios were set up based on the ideal hydraulic machines behaviour, and imposing the condition of lockup at the first-second and third-fourth gear synchronism points. This condition is known from literature for having properties of optimal efficiency, and it is considered be a preferred operating condition. However, the results demonstrated that zero speed and zero displacement can be deemed as singularities from the control point of view; in fact, due to the extremely low values and non-linearity of the volumetric efficiency of the hydraulic units, the control of said machines is sub optimal, leading to unsatisfactory speed control. Consequently, the transmission ratios were optimized to obtain an acceptable transmission control. The results highlighted an improved operation of the hydraulic machines, now working in better efficiency regions. The optimized design was further validated, checking the consistency of the pressure loads in the hydrostatic transmission. Finally, the transmission total efficiency was calculated finding out that, globally, the optimized model is more energy efficient, even though the full mechanical points are no longer exploited.

As a final remark, it is worth reaffirming that the theoretically optimal zero hydraulic power/full mechanical condition must be treated with caution, since it can be a critical condition from both control and energy perspectives.

In the next steps, the project will continue a systematic study of Full Power-Shift transmissions, possibly to obtain a unitary and systematic framework for the development of these architectures.

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