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Layout optimization of Input Coupled CVT transmission for agricultural tractors through real-world data

Luca Colendi[1] https://orcid.org/0009-0005-1512-0835 and

Michele Mattetti[2] https://orcid.org/0000-0002-2473-5250

[1,2] University of Bologna, Bologna, Italy
luca.colendi2@unibo.it

Abstract. The general trend of improving the fuel consumption efficiency of agricultural tractors goes hand in hand with the progressive increase in drivetrain design complexity. In this regard, the need to ensure low consumption, high performance, and operator comfort has widely recognized continuously variable transmissions (CVTs) as an optimal design solution. As part of the development of continuously variable transmissions for high-power tractors, several solutions have been proposed, showing an always strong interest in the type of hydromechanical CVT drivetrains. Given the intrinsic complexity of CVTs, the development of parametric-based drivetrain models is essential to accomplish predictive performance analysis and design development matching users' and manufacturers' requirements. This paper first presents a parametric modeling process of input-coupled multi-staged hydromechanical continuously variable transmissions (IHMCVT) for agricultural tractors. The procedure was designed to guarantee flexibility in terms of modeled numbers of transmission working stages, moreover, using an iterated model evaluation, it was possible to generate the output mesh for every operating point of the ICE and for each transmission ratio. A model validation activity was then carried out by applying the algorithm to a real-world IHMCVT architecture of a tractor already on the market. The tractor was properly instrumented so that real output data were available to be collected, while input data were extracted from the vehicle CAN-bus network. Validation was performed on both on-field and on-road tractor tasks. Finally, a constrained optimization algorithm was structured to define a possible objective procedure for the selection of optimal parameters in a generic considered IHMCVT transmission. This resulted in the identification of the best-suited design parameters for the given objective function and the number of requested working stages.

Keywords: CVTs, Efficiency, Parametric model, Validation, Optimization, CAN-data.

1 Introduction

An Input-coupled hydromechanical continuously variable transmission (IHMCVT) is a type of transmission commonly used in agricultural vehicles. Due to the considerable complexity of these drivetrains, designing an IHMCVT can be both time-consuming and costly. Therefore, selecting the most suitable transmission to meet the vehicle's performance requirements presents a significant challenge. Introducing systematic procedures for parameter selection represents a valuable solution, particularly in the current scenario where modularity is increasingly crucial for reducing design efforts. In addition, integrating a systematic approach with the utilization of real-world data can be an even more effective practice. Indeed, this approach can help identify the best design solution that meets performance requirements and aligns with users' habits regarding tractor use, thus potentially reducing fuel consumption.

2 Materials and methods

2.1 IHMCVT architecture and model

IHMCVT architecture. An Input-coupled hydromechanical continuously variable transmission (IHMCVT) is a type of drivetrain where power input is passed through a hydrostatic variator, allowing for continuous transmission ratio adjustment. An IHMCVT consists of three main elements: an ordinary gear (OG) connected to the ICE, a planetary gear (PG), and a hydrostatic variator (HST) composed of two swash plate hydraulic units (U1 and U2). More specifically, a two-staged PG was considered with power extraction from the carrier and the second solar, while a discrete gearbox (GB) was applied at the end of the main IHMCVT group. Finally, to ensure transmission ratio variation U1 was set to have variable displacement.

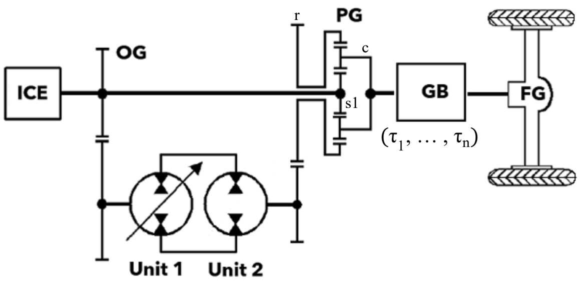

Fig. 1 Example of IHMCVT with single-staged PG.

IHMCVT model. The following section presents equations used to outline the drivetrain model. Given a generic OG, Eq. (1) and (2) express its kinematic and torque relations, where T_{in}, ω_{in}, T_{out}, ω_{out} are the input and output torque and angular speed and η_m is the efficiency.

$$\tau_{OG}=\frac{\omega_{out}}{\omega_{in}} \tag{1}$$

$$\eta_m = \frac{T_{out}}{T_{in}} * \tau_{OG} \tag{2}$$

The two-stage PG was modeled through the Willis formulas and ideal power balances, e.g., Eq. (3) expressed the first stage torque and angular speed relations.

$$\tau_{PG1} = \frac{\omega_r - \omega_c}{\omega_{s1} - \omega_c} = -\frac{T_{s1}}{T_r} \tag{3}$$

The HST was modeled via flow rate and pressure drop equations so that, e.g., for the first stage Eq. (4) and (5) were obtained.

$$K = \frac{\omega_{U1}}{\omega_{U2}} = \frac{\lambda * D_{U1}}{D_{U2}} * \eta_v^{sign(-\lambda)} \tag{4}$$

$$\frac{T_{U1}}{T_{U2}} = \frac{\lambda * D_{U1}}{D_{U2}} * \eta_{hm}^{sign(+\lambda)} \tag{5}$$

with $\lambda \in [+1, -1]$ a parameter representing the fraction selected displacement D_{U1} (D_{U2} is fixed), the sign function accounts for power direction in HST. On the other hand, η_{hm} and η_v are, HST mechanical and volumetric efficiency, while T_{U1}, ω_{U1}, T_{U2}, ω_{U2} are the torques and angular speeds of the units. Three additional equations are needed to solve the implicit problem: the pressure equation Eq. (6) here expressed for U2, the mechanical shaft power balance Eq. (7), and the final PG power balance.

$$p_{U2} = \frac{2\pi * T_{U2}}{D_{U2} * \eta_{hm_{U2}}^{sign(+\lambda)}} \tag{6}$$

$$T_{ICE} + T_{OG} + T_{s1} = 0 \tag{7}$$

A calculation loop similar to the one defined by Macor and Rossetti, (2011) was implemented to achieve convergence for the implicit problem for every possible transmission working point (T_{ICE}, ω_{ICE}, λ). Additional inner loops were defined to guarantee flexibility of the algorithm, allowing the calculation to repeat for the chosen number of discrete GB ratios ($\tau_1, \dots, \tau_n$), but also to ensure HST maximum pressure and angular speed limitation.

2.2 Model validation

An experimental validation activity was carried out. The model was applied to the commercial IHMCVT transmission architecture of the tractor New Holland T7.315 HD (CNH Industrial SpA, Basilon, UK), equipped with a four-gear GB. Experimental data were collected exploiting a data logger (Kvaser Memorator Pro 5xHS, KVASER, Molndal, SE) for vehicle CAN data extraction and Wheel Force Transducers (LW-2T-100K, Michigan Scientific Corporation, Charlevoix, US). Tests were conducted to collect data from pulling and transport scenarios. Drawbar pulling tests were performed using a second tractor (New Holland T8 Genesis™ 335 from CNH Industrial SpA, Basilon, UK) that applied a traction resistance to the first one according to a ramp-shaped technique defined by Angelucci and Mattetti (submitted). Transport tests consisted in real on-road activity, typically farm-to-field transports. Data extracted from the CAN-Bus network were: engine torque (T_{ICE}), engine angular speed (ω_{ICE}), and tractor rear axle angular speed (ω_W). CAN data were used to calculate the predicted pinion torque. Meanwhile, traction data on the four tractor wheels were collected using WFTs, so that total wheel torque was calculated and brought back to the drivetrain pinion. Simulated (T_{pin}^{sim}) and experimental pinion torque (T_{pin}^{exp}) were then compared through relative error ($e_{\%}$ as in Eq. (9)) to determine the statistical goodness of the method and thus enable the definition of the optimization procedure.

$$\frac{e_{\%}}{100} = \frac{T_{pin}^{sim} - T_{pin}^{exp}}{T_{pin}^{exp}} \tag{9}$$

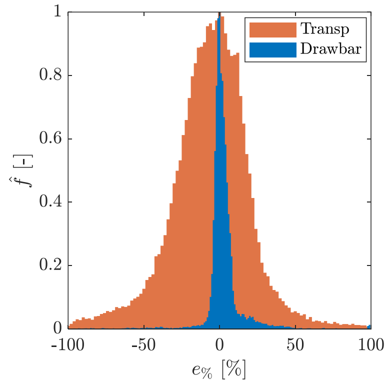

Fig. 2 Relative error distribution of drawbar and transport tests.

Fig. 2 presents the relative error distribution of the drawbar pulling and transport tests. The frequencies ($\hat{f}$) are normalized. Both distributions are centered at less than 1.4% relative error, thus making modeling outputs statistically significant. On the other hand, there is discrepancy between drawbar and transport test results in terms of data dispersion (standard deviations are σ_d=12.9% and σ_t=27.1%), with transport test having a wider data distribution due to the presence of obvious dynamic phenomena.

2.3 Optimization problem

The optimization problems (OPs) are now presented. An optimization is a systematic procedure to identify the best feasible solution to a specific problem. Hence, the optimization algorithm must search for the combination of design variables that minimizes the function Φ_f subject to a set of equality and inequality constraints. To determine the construction parameters of an IHMCVT, two optimization problems were formulated. Both OPs were implemented over the following design variables $\bar{x}_i$=(D_{U1}, D_{U2}, τ_{PG1},

τ_{PG2}, τ_1, … , τ_n). The constraints applied to OPs were linear (LC) and non-linear (NLC). LCs consisted of the hierarchy inequality relations between PG first and second stages gear ratios and the hierarchy inequality relations between consecutive GB ratios. On the other hand, NLCs ensured the shifting between GB gearwheels using the shift-point equation (see Eq. (11)) for two consecutive GB gears. Finally, two more NLCs were applied to ensure a minimum drawbar force at low tractor speed and a minimum tractor speed target.

$$K_{i\rightarrow i+1}=\frac{\tau_{PG1}}{\tau_{GB}}*\frac{\frac{(\tau_{PG2}-1)}{\tau_{PG2}}*\tau_i-\tau_{i+1}}{\frac{(\tau_{PG2}-\tau_{PG1})}{\tau_{PG2}}*\tau_i-\tau_{i+1}} \tag{11}$$

The first optimization (OP1) was structured so the algorithm would search for the set of design variables maximizing the drivetrain arithmetic mean efficiency (η_{AM}), calculated over the tractor speed span. The objective function was thus defined as in Eq. (12).

$$\Phi_{OP1}=\eta_{AM}=\frac{1}{n}*\sum_{j=1}\eta_{CVT}(v_j) \tag{12}$$

$$\Phi_{OP2}=\eta_{WM}=\frac{1}{100}*\sum_{j=1}\eta_{CVT}(v_j)*t_{\%}(v_j) \tag{13}$$

The second optimization (OP2), was implemented to maximize a weighted arithmetic mean efficiency (η_{WM}) of the drivetrain (see Eq. (13)), again calculated over the tractor speed span. In this last case, the weight for each efficiency point of the tractor was defined as the mean time-percentage spent at the corresponding speed point by a monitored worldwide tractor fleet. Given the absence of specific information regarding the ICE utilization in the tractor fleet, the most frequent (high-power) working point was chosen as the input of both OPs to ensure comparability between the two OPs. Since there are many possible operating points for internal combustion engines (ICE), selecting one specific point for driveline testing is common practice. For example, Macor and Rossetti (2011) and Scamperle (2017) considered the design power level and the design angular speed of the ICE. The tractor fleet comprised roughly four hundred medium tractors New Holland T7.270 equipped with the studied IHMCVT. Indeed, the type of usage, the driving style, and the use scenario are extremely variable from farmer to farmer, therefore the global mean percentage of time spent at a particular speed interval was employed as an overall usage indicator. The distribution of the percentage of time is shown in Fig. 3 (only the central value for each speed interval is represented). Speed intervals were properly chosen according to the data variability. It was pointed out that tractors worked most of the time at low-speed intervals, with an overall usage percentage of time of 75% for speed between 0 km/h and 12 km/h, and peak at 6.5 km/h. On the other hand, a second area was found in the range of 40-54 km/h, with a 9% peak at 53 km/h. A similar OP was proposed by Pettersson and Krus (2013), although frequency data were related to traction working points and were limited to the record of a single vehicle.

3 Results and discussion

3.1 Optimization results

Both OPs produced realistic drivetrain solutions. Fig. 3 also represents the ICE-to-wheels efficiency plots, for the considered ICE working point. The efficiency peaks in each plot reveal the four GB gears. OP1 shows a more consistent trend compared to OP2. The first one has almost no points of discontinuity in the efficiency plot when the gearbox shifts occur, except for a modest efficiency gap when shifting from the second to the third gear. This is a direct effect of the average efficiency OP. Truly, second and third gear are extremely close and positioned in a region of little significant use (range 12-34 km/h). Also, the last gear efficiency peak is placed at 45 km/h, again in a not frequently used speed range.

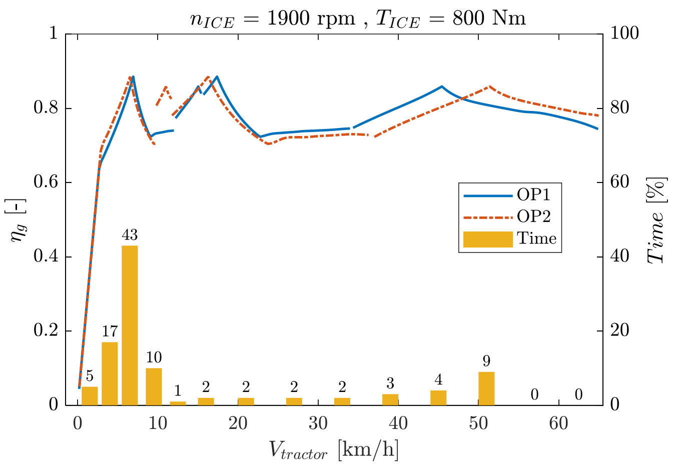

Fig. 3 OPs efficiency plots and speed frequency distribution.

On the other hand, the OP2 plot shows the effect of the time-based optimization, with the second gear moved at lower tractor speeds, to cover a larger region of intermediate use (range 0-12 km/h), ensuring high efficiency, but with a stronger tendency in generating gaps. In addition, OP2 has slightly moved the position of the first gear toward lower speed (effect of the 17%-time interval), while the third gear was moved to cover the speed ranges of the OP1 second and third gear. Finally, the fourth gear was relocated to set the efficiency peak close to the center of the second main speed range (52 km/h).

3.2 Optimization test

Given the variability of an IHMCVT efficiency with the transmission working point, a performance assessment procedure was executed on each OP solution. In particular, an artificial cycle, previously synthesized by Angelucci and Mattetti (2024), was applied to the OPs. It consisted of time-based signals T_{ICE}, ω_{ICE} and $V_{tractor}$, arranged in a sequence of typical activities like road transport, field passes with and without PTO, and idling. This sequence was designed to statistically describe the real use of an available tractor New Holland T7.270 by recombining 800 hours of work into a 2-hour trace. The cycle definition was achieved by selecting the most representative working sequences from the available tractor Can data. The signal-based cycle was fed to the OP models to perform the test, and the time-based pinion power difference between the two solutions was calculated as in Eq. (14), and the resulting points are shown in Fig. 4.

$$\Delta P_{pin}(t)=P_{OP2}-P_{OP1}=(T_{pin}^{OP2}-T_{pin}^{OP1})*\omega_{pin} \tag{14}$$

The power difference is shown in Fig. 4. Although optimization data were collected from an international tractor fleet, the tractor used for the cycle extraction reached a maximum speed of 44 km/h, thus limiting the maximum speed of the test. Given the statistical meaning of the cycle, which is meant to present a mix of agricultural activities based on their frequency, and given the possible sample-to-sample comparison between the two OPs, the total power extracted from the pinion is defined as in Eq. (15), where a positive ΔP indicates better global performance for OP2.

$$\Delta P=\sum_{t}\Delta P_{pin}(t) \tag{15}$$

For the 2-hour test case ΔP=5.303 e+4kW, equivalent to 1.9% more power collected with OP2 than OP1, making OP2 globally more efficient.

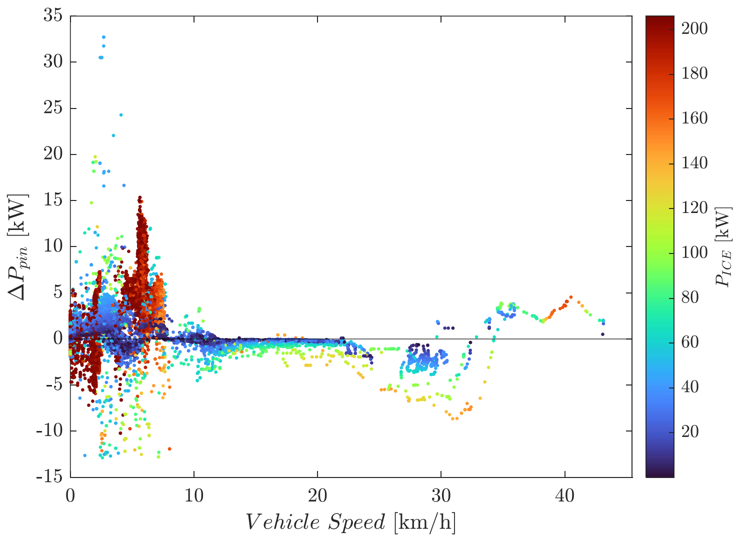

Fig. 4 Pinion power difference points from cycle test.

Fig. 4 shows four main areas: the 0-12 km/h, the most populated, the 13-21 km/h, the 22-33 km/h and the 34-44 km/h intervals. This range division is coherent based on the frequency distribution plot of tractor speeds and the intended optimization effects. The first interval is populated with low to high-power activities where ΔP_{pin} assumes extremely variable values, ranging from -13 kW up to +37 kW. This is due to the great variety of the IHMCVT working points. It is important to note in this interval 62% of points have a positive ΔP_{pin}. On the other hand, the second interval shows slightly negative ΔP_{pin} values since OP2's third gear decently overlaps the second and the third of OP1. OP2 third gear's final section covers the third interval (22-33 km/h) and slightly reduces efficiency. It's worth noting that the last interval (speeds greater than 33 km/h) does not align with the efficiency plot behavior since only positive ΔP_{pin} are found. This discrepancy is again attributed to the wide variety of transmission working points available.

4 Conclusion

In recent years, reducing emissions, minimizing fuel consumption, and customizing products have become increasingly important in the design phase of vehicles. The design phase involves numerous complexities, so developing a systematic method for identifying suitable parameters can be valuable. Moreover, it is possible to generate customized solutions by integrating real data into this chain. This study proposed a method for optimizing the parameters of an IHMCVT transmission. First, it outlined a general mean approach and then used worldwide real data to search for appropriate driveline parameters meeting the average operational needs. This approach allowed a reduction in fuel consumption, by generating a more efficient drivetrain.

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