Objective evaluation of gearshift process of agricultural tractors

Published Version:
Mattetti, M., Michielan, E., Mantovani, G., Varani, M. (2022). Objective evaluation of gearshift process of agricultural tractors. BIOSYSTEMS ENGINEERING, 224(December 2022), 324-335 [10.1016/j.biosystemseng.2022.11.001].

Availability:
This version is available at: https://hdl.handle.net/11585/907867 since: 2024-11-13

Published:
DOI: http://doi.org/10.1016/j.biosystemseng.2022.11.001

Objective evaluation of gearshift process of agricultural tractors

Michele Mattetti[a], *Enrico Michielan*[a], *Giacomo Mantovani*[a], *Massimiliano Varani*[a]*

[a] Department of Agricultural and Food Sciences – Alma Mater Sudiorum - University of Bologna, viale G. Fanin 50, Bologna, Italy

* Massimiliano Varani, tel. +39 051 2096174, fax +39 051 2096189, e-mail: massimiliano.varani@unibo.it

KEYWORDS: transmission; comfort; gearshift quality; driveability; CANBUS

Nomenclature		
a	longitudinal acceleration	(m s^{-2})
$a_s^{0\rightarrow 40}$	shuffle acceleration in acceleration tests	(m s^{-2})
$a_f^{0\rightarrow 40}$	forward acceleration in acceleration tests	(m s^{-2})
$a_{f,p}^{0\rightarrow 40}$	peak value of forward acceleration in acceleration tests	(m s^{-2})
$a_{s,RMS}^{0\rightarrow 40}$	shuffle index	(m s^{-2})
a_s^s	shuffle acceleration in shuttling tests	(m s^{-2})
a_{sp}^s	shuttling peak prominence	(m s^{-2})
a_{op}^s	overshoot peak prominence	(m s^{-2})
$C_{V,\hat{P}_e}^{0\rightarrow 40}$	coefficient of variation of normalised engine power	(-)
M_e	actual engine percent torque	(Nm)
M_f	nominal friction-percent torque	(Nm)
M_r	engine reference torque	(Nm)
n_e	engine speed	(rpm)
P_e	engine power	(kW)
$\hat{P}_e$	normalised engine power	(-)
$P_{e,p}$	engine peak power	(kW)
R	global metrics	(-)
$R^{0\rightarrow 40}$	acceleration average metrics	(-)
R^s	shuttling average metrics	(-)
$t^{0\rightarrow 40}$	acceleration duration	(s)
t^s	shuttling duration	(s)
t_{sp}^s	shuttling peak duration	(s)
V_t	tractor ground speed	(km h^{-1})
$X_{*,i}^+$	metrics of + test of i_th tractor	(-)
$\hat{X}_*^+$	normalised metrics of + test of i_th tractor	
$\hat{X}_*^{0\rightarrow 40}$	normalised metrics for acceleration test	(-)
$\hat{X}_{*,i}^s$	normalised metrics for shuttling test	(-)
α	tractor heading angle	(deg)
$\mu_{\hat{P}_e}^{0\rightarrow 40}$	mean value of normalised engine power	(-)
σ_a^s	standard deviation of shuffle acceleration in shuttling tests	(m s^{-2})

Abstract

In the last 20 years, manufacturers have invested in continuously variable transmissions (CVTs) for agricultural tractors, since farmers have demanded more easy-to-use transmissions to increase comfort and driveability. Recently, stepped transmissions have been of interest due to their greater efficiency compared to continuously variable transmissions. To fully compete with CVTs, stepped transmissions must be fully automated and properly handle the operator's comfort during gearshifts. This paper aims to define a test procedure and data analysis method for evaluating the gearshift quality of transmissions for agricultural tractors. The procedure is composed of two tests and aims to reproduce the typical manoeuvres that occur during farming, such as accelerating and shuttling. This method was applied to 14 tractors with different transmissions and engine peak power. On each tractor, an inertial measuring unit and a CAN-Logger were installed to record vehicle and powertrain operating conditions. From the recorded data, a set of metrics was calculated to objectively quantify the impact of gearshift. These metrics were combined to obtain an overall score, which allows for the ranking of tractors in terms of gearshift quality and its dependence on design parameters. Based on our metrics, CVTs outperform stepped transmissions due to their ability to combine high performance with comfort measures. The approach outlined in this study enables manufacturers to quickly spot the weak points of the gearshift behaviour of a tractor so that they can address specific issues and increase operator satisfaction and the overall quality of their tractors.

Introduction

The transmission is one of the most studied and expensive components in agricultural tractors, and can account for up to 26% of the entire tractor cost (Renius, 2019). Recent studies on transmissions for agricultural tractors have mostly focused on newer architectures (Blumenthal et al., 2000; Rossi et al., 2014; Troncon et al., 2019; Varani et al., 2021), accelerated life testing (Mattetti et al., 2019; Wen et al., 2021), and investigation of power losses (Molari & Sedoni, 2008). Over the last 20 years, farmers have demanded more and more easy-to-use transmissions to increase operator comfort and tractor driveability (Tinker, 1993), which has increased the efforts of tractor manufacturers to develop continuously variable transmissions (CVTs) (Blumenthal et al., 2000; Brenninger, 2007; Casoli et al., 2007). Stepped transmissions (i.e., partial and full powershift transmissions) have also been of interest to manufacturers, due in part to their greater efficiency than CVTs, and also because they allow for modular design architectures, which is an important requirement for worldwide manufacturers (Birkmann et al., 2018; Jenkins, 1997; Otten & Wrisberg, 1990). This interest in stepped transmissions has been demonstrated by the recent commercial releases of double-clutch transmissions, for example, New Holland with its Dynamic Command, Case IH with its ActiDrive8, and John Deere with its DirectDrive. This type of transmission permits the enhancement of powershift transmission efficiency by reducing the number of hydraulically operated clutches (Seeger, 2012) and reducing the flow of oil, which is the major cause of power losses in transmissions (Molari & Sedoni, 2008).

Tractors operate in a wide variety of operating conditions (in terms of traction load and ground speed), which requires a great number of speeds to maximise operational efficiency. The operational load may change during operation due to different soil conditions or crop flows, requiring frequent gearshifts to maintain high operational efficiency. Since the ‘50s, manufacturers have provided transmissions that are equipped with systems which enable gearshifts without any power interruption (Harris & Jensen, 1964). These systems were based on clutches that were controlled by on-off hydraulic valves with no modulations, which led to abrupt gearshifts, especially during on-road driving (Elfes, 1961). Only in the ‘80s were electronic shift controls integrated into transmissions, which permitted accurate control of the clutch timing in the function of operating parameter (i.e., oil viscosity, input torque, clutch wear) (Ross & Panoushek, 1990). To fully compete with CVTs, stepped transmissions must be fully automated and properly handle the operator's comfort during gearshifts. Therefore, shifting elements should be actuated with certain shift parameters to consistently provide a quick, smooth, and efficient gearshift (Goetz et al., 2005; Tanelli et al., 2011). Typically, shift parameters are tuned based on the subjective impressions of transmission calibrators when driving a testing vehicle through a series of representative manoeuvres (Sorniotti et al., 2007). However, calibrators may not have a consistent and repeatable subjective judgment and they are not able to discern subtle changes in gearshift behaviour, leading to a potential unclear sense of dissatisfaction. To compare different parameters of the gearshift controller, or different vehicles, metrics to objectively evaluate the quality of the gearshift are required. These must be able to quantify the quality of shifts in terms of both performance and comfort so that different vehicles can be compared. For example, a well-known metric in the automotive industry for evaluating performance is the duration of the shift, while a metric for evaluating comfort is the jerk of longitudinal acceleration (Huang & Wang, 2004; Wheals et al., 2002). Most of the studies on gearshift quality are focused on on-road vehicles, and there is a significant lack of studies on gearshift metrics for agricultural tractors. There is only one recent study on agricultural tractors, but it is focused on simple metrics such as shifting skid work, impact, and shift response time (Xia et al., 2020). However, these simple metrics cannot capture the discomfort caused by gearshifts. Moreover, each study proposed a testing procedure and results are limited to a single tractor and this become problematic when comparing results from different studies. This paper aims to fill this gap by reporting a standardised testing procedure for evaluating the gearshift behaviour of tractors with automated transmissions which permits to rank tractors in function of their behaviours. This procedure was applied to several tractors so that the impact of tractor’ parameters could be outlined.

Materials and methods

The authors have been testing tractors as part of a project with New Business Media srl aimed to develop methods for testing tractor drivetrains. In total, 14 tractors from different brands and with different types of transmissions have been tested, and their specifications are reported in Table 1. The types of tractors tested cover a wide range in terms of mass (and power) levels and types of transmission.

To avoid any commercialism, brands and models of tractors were not reported. Tractors were denoted with an identification code composed of the type of transmission—CVT for continuously variable transmissions and ST for stepped transmissions (i.e., powershift and dual-clutch transmissions)—followed by a progressive number increasing with tractor mass. All tractors were equipped with automatic transmission mode. All tested tractors were brand new.

Table 1– Main specifications of the tractors used for the test.

Identification Code	Mass [kg]	Eng. max power [kW]	Transmission typology	Number of forward rations	Seat base height from the ground [mm]	Power-to-mass ratio [kg kW^{-1}]
ST1	5705	107	Powershift	24	1470	53
ST2	5960	98	Powershift	16	1400	61
ST3	6335	103	Dual-clutch	24	1370	62
ST4	7320	99	Dual-clutch	24	1520	74
ST5	8200	193	Powershift	19	1560	50
ST6	8520	121	Powershift	30	1500	70
ST7	11967	195	Powershift	28	1760	61
CVT1	5910	96	CVT	[-]	1370	62
CVT2	5950	102	CVT	[-]	1550	58
CVT3	7470	147	CVT	[-]	1900	51
CVT4	8680	125	CVT	[-]	1600	69
CVT5	10040	206	CVT	[-]	1800	49
CVT6	12230	213	CVT	[-]	1850	57
CVT7	12570	228	CVT	[-]	1880	55

All tractors were homologated for the Italian market; therefore, their top speed was limited to 40 km h^{-1}. Tractors were tested on a paved track since on-road driving is the most critical condition for the gearshift process since the hard ground does not to damp the eventual shuffle movement caused by the variability of the wheel torque. All the tractors were tested in bare conditions (i.e., with no front or rear implement) to avoid any change from the mass and its distribution set by each manufacturer. To ensure sufficient test repeatability, prior to the tests, the tyre pressure was set in the function of the axle static load as suggested by tyre manufacturers. Each tractor was tested in unladen condition (i.e., with no front or rear ballasts) and with the same driver to avoid any driver bias in the results.

Data acquisition

The data acquisition system used for each tractor is reported in the following:

- CAN-BUS data logger (CANCaseXl log, Vector Informatik GmbH, Stuttgart, Germany), which permitted the recording of signals with the following Suspect Parameter Numbers (SPNs) and Parameter Group Numbers (PGNs):
    - SPN 544 and PGN 65251: "*Engine reference torque,*" reports the maximum torque delivered by the engine (M_r). The sampling rate was 10 Hz.
    - SPN 513 and PGN 61444: "*Actual engine per cent torque,*" which reports the torque as a per cent of M_r (M_e). The sampling rate was 10 Hz.
    - SPN 513 and PGN 5398: "*Nominal friction-per cent torque,*" which reports the frictional and thermodynamic loss of the engine itself, pumping torque loss, and the losses of fuel, oil and cooling pumps as a per cent of M_r (M_f). The sampling rate was 10 Hz.
    - SPN 190 and PGN 61444: "*Engine speed,*" which reports the revolution speed of the engine crankshaft (n_e). The sampling rate was 10 Hz.
- Inertial Measurement Unit (IMU), (MV5-AR, LORD MicroStrain®, Cary, North Carolina, United States of America) was mounted at the seat base for a consistent placement among all the tractors so the influence of the IMU location on the measured linear acceleration was minimised (Molari et al., 2011). This IMU permitted the measurement of the longitudinal acceleration (a). The samping rate was set at 500 Hz.
- Global Navigation Satellite System (GNSS) receiver (IPESPEED, IPETronik GmbH, Baden, Germany) with an updated frequency of 25 Hz, which permitted the

recording of the absolute value of the tractor ground speed (V_t) and its heading angle (α), which allows the detection of tractor direction.

To evaluate the gear shift quality, an ad hoc testing procedure was developed to evaluate tractor behaviour in the most important driving manoeuvres carried out during tractor use where any gearshift process occurs. Those manoeuvres are acceleration and shuttling. The acceleration test aimed to reproduce starting the tractor from a standstill position during on-road driving and its procedure is briefly reported as follows. Starting with the transmission in neutral and in the automatic mode, the tractor was set to full throttle. After holding the tractor in a standstill position for 5 s, the driver moved the power shuttle into the forward position. When the tractor reached 40 km h^{-1}, this speed was kept for at least 3 s. This test was repeated 5 times for each tractor. The shuttling test aimed to reproduce tractor direction reversal occurring at headland turns where the power shuttle lever is typically used and its procedure is briefly reported as follows. The gear ratios for forward and rearward directions were chosen as the closest gear ratio that permitted each tractor to travel at 15 km h^{-1} with n_e at 1700 rpm in the two tractor directions (i.e., forward and rearward). Subsequently, the throttle position to maintain for the entire test was determined. The combination of n_e and V_t was chosen based on the results from the real-world data collected and analysed by Mattetti (2021). Thanks to this combination, all the tractors were tested with consistent gear ratios, and the influence of the gear ratio variability from tractor to tractor was minimised (Kim et al., 2007; Yin et al., 2014). Tractors equipped with CVTs were completely automated and, therefore, the transmission ratio could not be controlled on those tractors. The test procedure was composed of 10 shuttling inversions described in the following. The tractor was kept at a standstill for 5 s with the target n_e, throttle position, and gear ratio. Then, the operator moved the power shuttle lever in the forward direction, and the tractor accelerated until it reached the target V_t in the forward direction, which was kept for at least 5 s to reach a steady state condition. After that, the power shuttle lever was moved in the rearward direction, and the tractor decelerated until reaching the target V_t in the rearward direction. At the end of the 10th inversion, the power shuttle lever was moved into the neutral position.

Data analysis

All the signals were interpolated at 500 Hz with a cubic spline for a consistent sampling rate. The engine power (P_e) and its normalised value ($\hat{P}_e$) were calculated with Eq. 1 and Eq. 2, respectively, where $P_{e,p}$ is the engine max power from Table 1.

$$P_e = M_r \cdot \frac{M_e - M_f}{100} \cdot n_e \cdot \frac{2\pi}{60} \qquad \text{Eq. 1}$$

$$\hat{P}_e = \frac{P_e}{P_{e,p}} \qquad \text{Eq. 2}$$

For both tests, a was the superimposition of vehicle forward acceleration $\left(a_f\right)$ and vehicle shuffle (a_s), which is an oscillation of vehicle acceleration around the mean forward acceleration due to sudden changes in engine load (Reddy et al., 2019) (Figure 1). a_f was calculated by filtering a with a zero-lag low-pass filter with a cut-off frequency of 1 Hz, while a_s was calculated by filtering a with a zero-lag high-pass filter with a cut-off frequency of 1 Hz. For the acceleration test, the portions of signals occurring when V_t crossed 0.1 and 39.9 km h^{-1} with a positive slope were associated to a single acceleration manoeuvre. In Figure 1, an example of acceleration manoeuvre is reported.

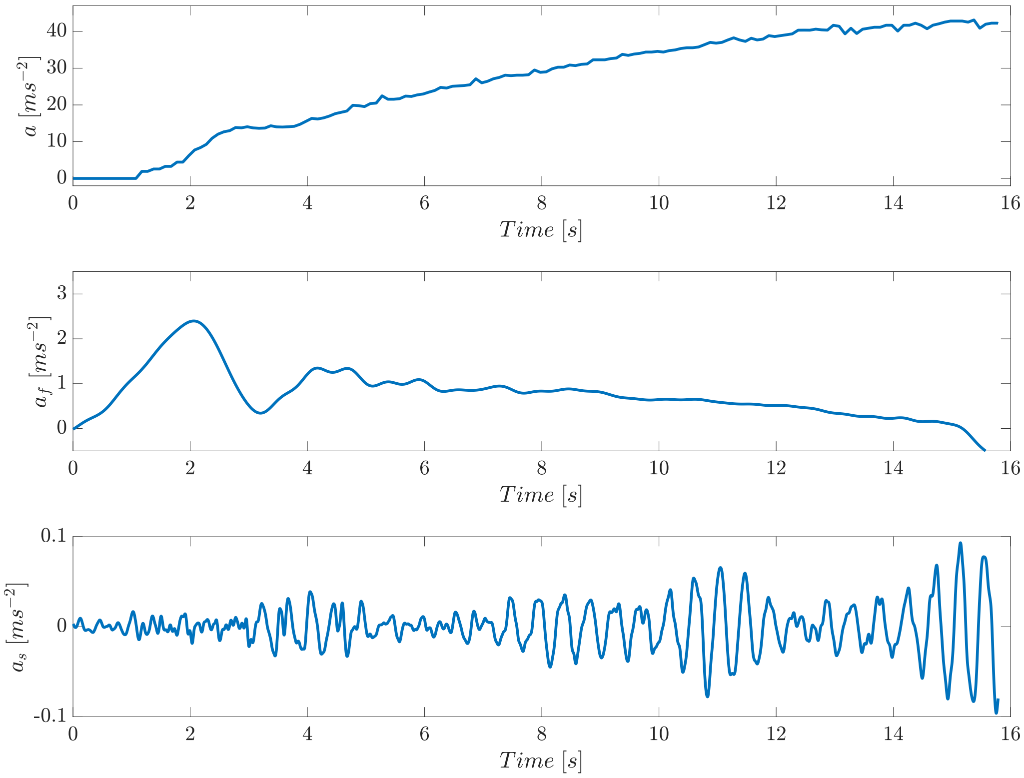

Figure 1: Tractor ground speed (V_t) *(top), tractor forward acceleration* $\left(a_f\right)$ *(centre), and tractor shuffle acceleration* (a_s) *(bottom).*

During the gearshift process, the vehicle dynamic state is usually changed due to the fact there a change a traction forces at the wheels. This change may induce a change a tractor ground speed, and tractor shuffling, and jerking. The greater is this change and the greater are this inducement. To capture such phenomenon, a certain metrics were calculated for each test. In particular, during each acceleration manoeuvre, the following metrics were calculated:

- acceleration duration ($t^{0\rightarrow 40}$), calculated as the time elapsed between the initial and the final instants of the acceleration manoeuvre;
- peak value of a_f ($a_{f,p}^{0\rightarrow 40}$);
- shuffle index ($a_{s,RMS}^{0\rightarrow 40}$), which was calculated as the root-mean-square (RMS) of a_s filtered with the w_d weighting function described in ISO 2631-1 (ISO, 1997). This filtering was conducted to take into consideration the most critical frequency band for the human body according to Griffin (1996);

- mean value of $\hat{P}_e$ ($\mu_{\hat{P}_e}^{0\to 40}$), permitting the evaluation of the power level delivered by the engine with respect to its maximum value;
- coefficient of variation of $\hat{P}_e$ ($C_{V,\hat{P}_e}^{0\to 40}$), permitting the quantification of the consistency of the engine power.

A shuttling manoeuvre (forward-to-rearward or vice versa) occurred anytime the peak-to-peak value of V_t in a moving window of 1 s was above 0.5 km h^{-1}. In Figure 2, a for a reference shuttling manoeuvre is reported together with the main metrics.

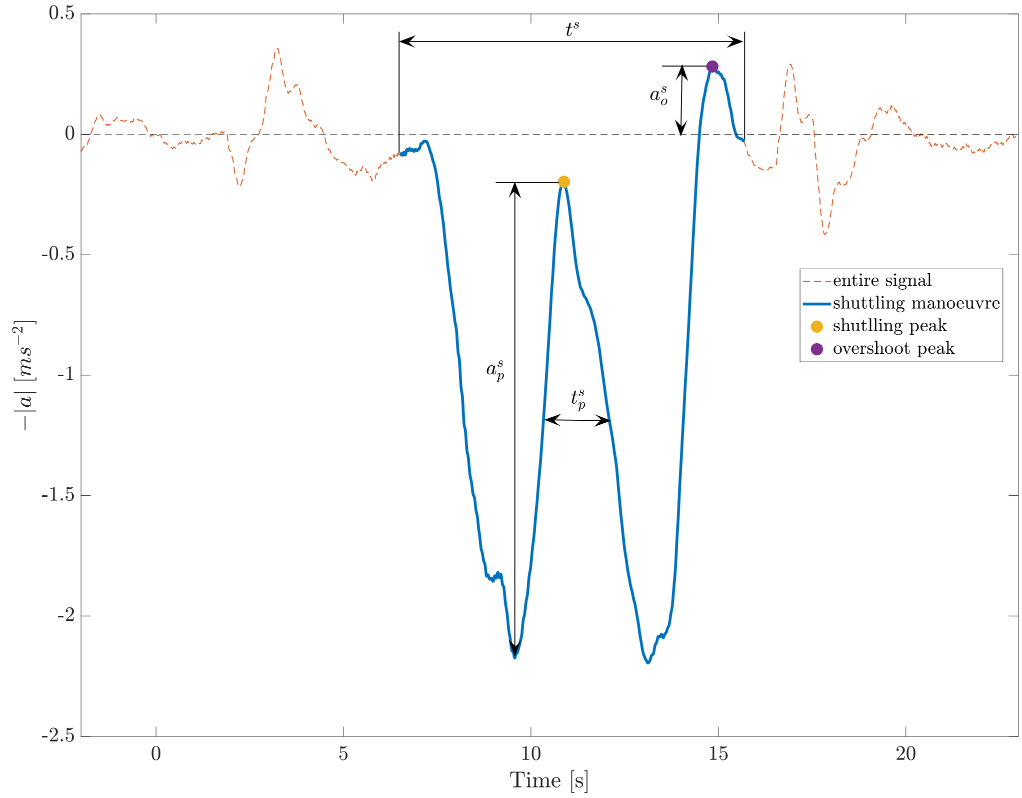

Figure 2: Tractor longitudinal acceleration (a) and the shuttling and overshoot peaks during a shuttling event are reported.

The steady-state portion prescribed in the test procedure was not included in the calculation of the metrics to avoid any driver bias. For subsequent analysis, a was forced to be negative for all manoeuvres; thus, any significant change in a was evaluated by detecting the peaks that occurred during manoeuvres. In particular, peaks with a prominence greater than 0.2 m s^{-2} were detected. This threshold was chosen by the authors after a visual inspection, and it permitted the detection of the most important peaks without any detection of secondary smaller peaks. During manoeuvres, two peaks were detected— a *shuttling peak* occurring at the inversion point (where V_t was approximately 0 km h^{-1}) and an *overshoot peak* occurring at the end of manoeuvres (Figure 2). In each manoeuvre, the following metrics were calculated (Figure 2):

- manoeuvre duration (t^s), calculated as the time elapsed between the initial and the final instants of shuttling manoeuvre;
- shuttling peak prominence (a_{sp}^s);
- shuttling peak duration (t_{sp}^s), using half of the peak prominence as a reference value for its calculation;
- overshoot peak prominence (a_{op}^s);
- standard deviation of a_s (σ_a^s);

For tractors where no shuttling or overshoot peaks were observed, a_{sp}^s, t_p^s, or a_{op}^s were set to 0.

In order to compare different metrics which may have a different range of values, each metric was transformed to fall in the range between 0 and 1, where 0 was assigned to the worst value among the tractors and 1 was assigned to the best value. Thus, the generic metric of i_th tractor ($X_{*,i}^+$), where the lower the values are the better they are for the operator, such as $t^{0\to40}$, $a_{s,RMS}^{0\to40}$, and t^s, $C_{V,\hat{P}_e}^{0\to40}$, t^s, a_{sp}^s, t_{sp}^s, a_{op}^s, and σ_a^s, were transformed into the variable $\hat{X}_{*,i}^+$ with Eq. 3, while the others, such as $a_{f,p}^{0\to40}$, and $\mu_{\hat{P}_e}^{0\to40}$, were transformed into the variable $\hat{X}_{*,i}^+$ with Eq. 4.

$$\hat{X}_{*,i}^+ = \frac{\max\limits_i X_{*,i}^+ - X_{*,i}^+}{\max\limits_i X_{*,i}^+ - \min\limits_i X_{*,i}^+} \qquad \text{Eq. 3}$$

$$\hat{X}_{*,i}^+ = \frac{X_{*,i}^+ - \min\limits_i X_{*,i}^+}{\max\limits_i X_{*,i}^+ - \min\limits_i X_{*,i}^+} \qquad \text{Eq. 4}$$

In the above equations, + is the symbol for the generic test (i.e., $0 \rightarrow 40$ for the acceleration test and s for the shuttling test), and * is the symbol for the generic metric. For each tractor, the mean value of $\hat{X}_*^{0\rightarrow 40}$ and $\hat{X}_*^s$ were calculated to obtain $R^{0\rightarrow 40}$ and R^s, respectively. These two parameters can be considered as a single parameter which permitted, for each test, to rank the tractors from the worst (i.e., the one with the lowest value) to the best (i.e., the one with the greatest value). Finally, global metrics (R) were calculated as a weighted average of the mean value between the ranking in the two tests (Eq. 5).

$$R = 0.3 * R^{0\rightarrow 40} + 0.7 * R^s \qquad \text{Eq. 5}$$

The weighting coefficients (0.3 and 0.7) were determined using the approach reported in the Appendix.

Results

Acceleration test

In Figure 3, example behaviour of V_t (top) and $\hat{P}_e$ (bottom) is reported during two extreme acceleration manoeuvres for tractors ST2 and CVT1. One can note that V_t increases nonlinearly exponentially during the test for both tractors, which is consistent with the literature (Rakha et al., 2001). On the other hand, $\hat{P}_e$ for tractor ST2 shows several drops occurring during the gearshift, which led to a reduction of the load applied to the engine due to the decoupling between engine and transmission. This behaviour is common to many ST tractors. In these circumstances, no power is transmitted to the rear wheels and a slight reduction of V_t can be observed. This reduction is mostly dependent on the tractor mass, tire rolling resistance, and durations of gearshifts. When the gearshift was completed, $\hat{P}_e$ increased. This increase varied for different tractors, most likely because it was dependent on the controller management of the engine during gearshifts. A completely different behaviour was observed for CVT tractors, as they provide more consistent engine power throughout a manoeuvre because engine power can be delivered to the wheels with no power interruption.

Several studies on heavy-duty trucks have assumed that engines deliver the maximum engine power over the entire acceleration manoeuvre (Rakha et al., 2001; Searle, 1999). This assumption was considered quite sufficient to predict the maximum acceleration of such vehicles; however, it cannot always be applied to agricultural tractors equipped with stepped transmissions, where a large number of gearshifts occur during acceleration manoeuvres. Moreover, due to the geometrical progression of transmission ratios typically adopted in agricultural tractors (Naunheimer et al., 2010), gearshift spacing increases at high-speed and, therefore, decreases in $\widehat{P}_e$ were less frequent at the final part of the acceleration (i.e., high speeds). The trend for a was consistent with another study related to heavy-duty trucks (Shin et al., 2014). Specifically, a reached a major peak at the beginning of the manoeuvre and then gradually decreased, oscillating around 0 m s^{-2} at the end of the manoeuvre (Figure 2 – top and centre). With respect to heavy-duty vehicles, the major peak of a is more damped, so the variation between two consecutive peaks is smaller than those observed in on-road vehicles (Grotjahn et al., 2006; Shin et al., 2014) (Figure 2), likely because agricultural tractors have greater inertia than heavy-duty vehicles. a oscillated around the mean of the forward acceleration due to the gearshifts, which led to sudden changes in engine load and caused twisting and untwisting of the driveshafts due to compliances and backlashes in the transmission and to changes in traction forces at the wheels (Reddy et al., 2019). This effect is called shuffle, which is well-discussed in the literature and is more evident in lower gears due to the greater transmission torque ratios (Biermann et al., 2000). Shuffle is probably more critical for agricultural tractors than for on-road vehicles (i.e., passenger cars) due to the greater number of gearshifts occurring during acceleration manoeuvres, the smaller gear ratios, and the eventual impulses coming from towed implements/trailers during their service. Shuffle can be very critical in terms of discomfort since it usually lies in a critical frequency band for the human body (i.e., 4-8 Hz).

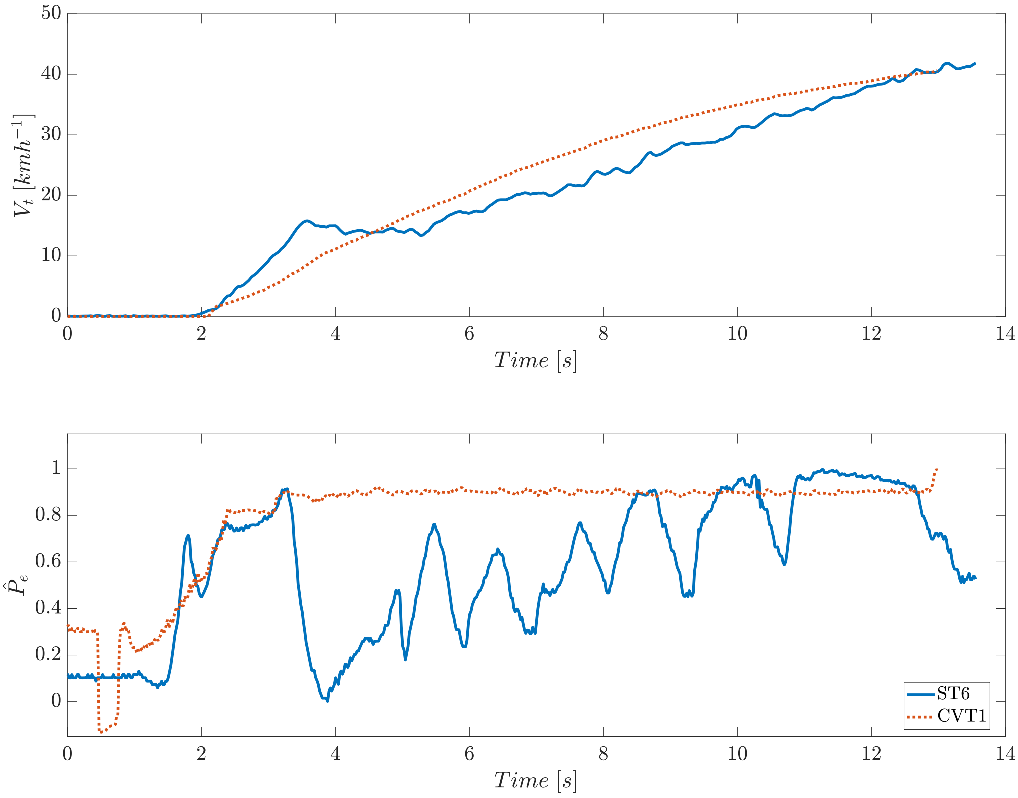

Figure 3:Tractor ground speed (V_t) (top) and engine power ($\hat{P}_e$) (bottom) during an acceleration manoeuvre for ST6 and CVT2.

$t^{0\rightarrow 40}$ was negatively correlated with mass-to-power ratio (the Pearson's correlation coefficient was -0.58), and the correlation appeared to be stronger in CVT than in ST tractors. This is probably caused by the fact that lower values of $C_{V,\hat{P}_e}^{0\rightarrow 40}$ and greater values of $\mu_{\hat{P}_e}^{0\rightarrow 40}$ were observed for CVT tractors compared to ST tractors for gearshifts occurring during acceleration manoeuvres. This agrees with the result of CVT tractors achieving significantly better values of $t^{0\rightarrow 40}$ than ST tractors. Indeed, CVT1 and ST3 were manufactured by the same company and had the same engine peak power, tires, and similar masses; therefore, they should achieve similar theoretical acceleration performance (Rakha et al., 2001). However, $t^{0\to40}$ of ST3 was 28% longer than that of CVT1 due to the installed transmission, which may differ for efficiency and shifting behaviour (Figure 4 – top). For tractor ST3, a change of range occurred during the acceleration test (located between 2 and 4 s in Figure 4 – centre), which was responsible for a 49% greater value of $C^{0\to40}_{V,\hat{P}_e}$ and 15% lower value $a^{0\to40}_{f,p}$ with respect to CVT1. Moreover, the change of range caused a severe reduction of a, and a secondary major peak at the engagement of the subsequent higher range was observed (Figure 4 – centre). ST3 also had a slightly greater shuffle than CVT1 (Figure 4 – bottom); indeed, it had a 14% greater $a^{0\to40}_{s,RMS}$ than ST3 on average.

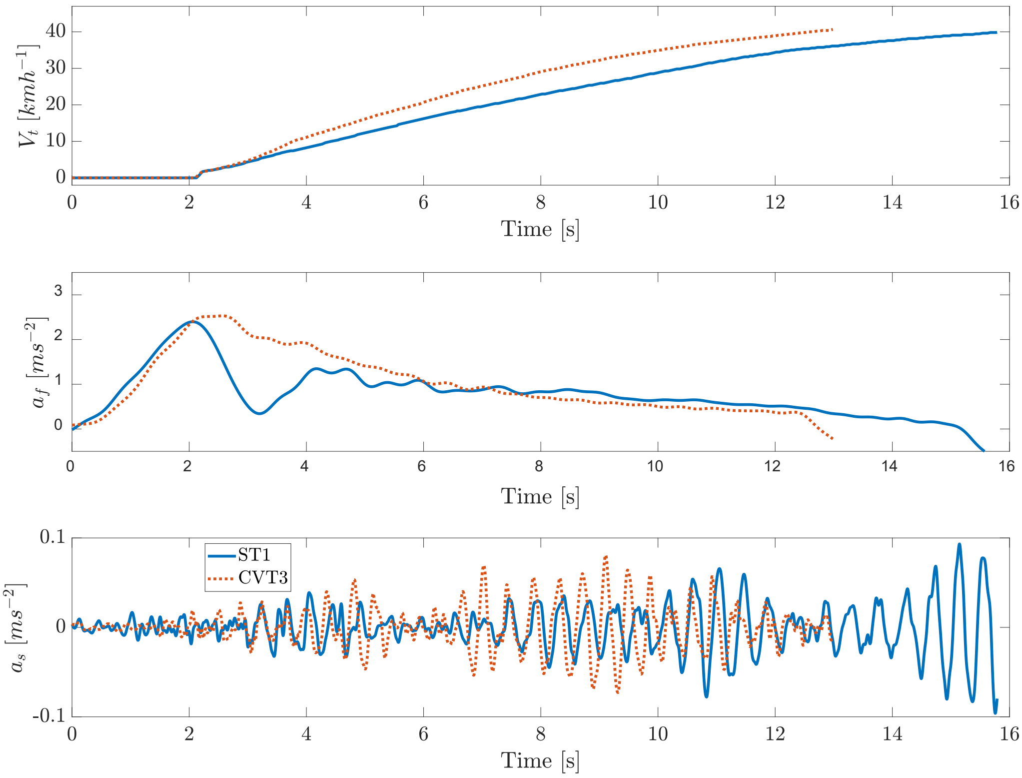

Figure 4: Tractor ground speed (V_t) (top), tractor forward acceleration (a_f) (centre), and tractor shuffle acceleration (a_s) (bottom) for a representative manoeuvre for ST3 and CVT1 tractors.

In Figure 5, the range (i.e., the difference between the minimum and the maximum values) of $\hat{X}_*^{0\to40}$ for each tractor is reported. One can note that many tractors have a very large range (e.g., ST7 and CVT7), while others exhibited a small range (e.g., CVT5 and CVT4). Several CVT tractors have higher rankings, while several ST tractors are ranked lower. This is due to the fact that CVT tractors exhibited significantly better $t^{0\to40}$, $\mu_{\hat{P}_e}^{0\to40}$, and $C_{V,\hat{P}_e}^{0\to40}$ than ST tractors. One can note that tractors ST5 and CVT1 (reported in Figure 3) are placed in two extreme positions in the ranking of Figure 4. Tractor ST6 had the greatest possible range, because it displayed the smoothest acceleration signals but also the highest $t^{0\to40}$. Tractors CVT6 is the with the best performance since is the one with the greatest $R^{0\to40}$.

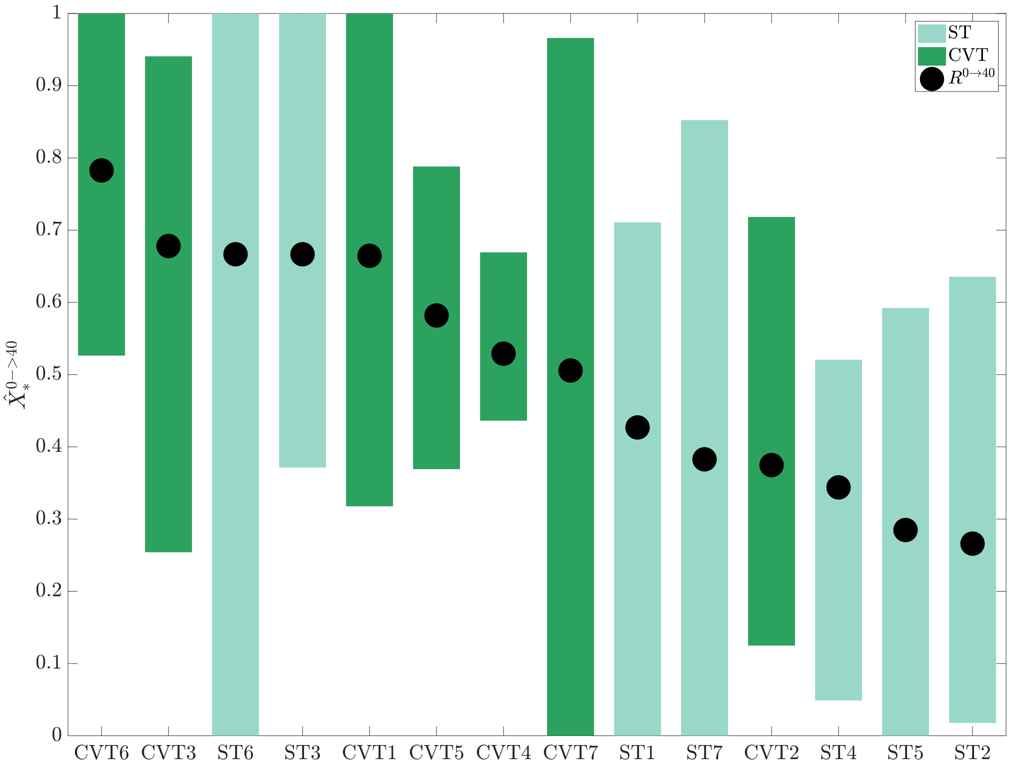

Figure 5: Range of the scaled metrics of the acceleration test ($\hat{X}_*^{0\to40}$), sorted by $R^{0\to40}$ mean values.

Shuttling test

Figure 6 reports two extreme shuttling manoeuvres recorded for tractors CVT4 and CVT6. In high quality shuttling manoeuvres, a increased rapidly and smoothly at the engagement of the shuttle lever, reaching a maximum value at the inversion point of tractor direction (where V_t was 0 km h^{-1}). During these manoeuvres, no shuttling and overshoot peaks were observed (Figure 6 – top). In low quality manoeuvres, shuttling and overshoot peaks were clear (Figure 6 – bottom). In particular, the shuttling peak occurred at the inversion of the point (i.e., when V_t is 0 km h^{-1}) due to the engagement process of the power shuttle clutch, which may lead to a temporary reduction in the power transmitted to the wheels. During the shuttling, peaks lasted for almost 2 s while a approached 0 m s^{-2} (Figure 6 – bottom). The overshoot peak led to a sign inversion of a, which led to a ground speed overshoot (this overshoot was not detected in V_t due to the low sampling rate of the GNSS receiver adopted in the test). As reported in the literature, when vehicles accelerate in a certain direction, the sign inversion of longitudinal acceleration must be avoided since it leads to significant driver discomfort (Goetz et al., 2005). This led to better-scaled metrics for CVT4 than CVT6 and, therefore, to a foreseeable better judgment of drivers.

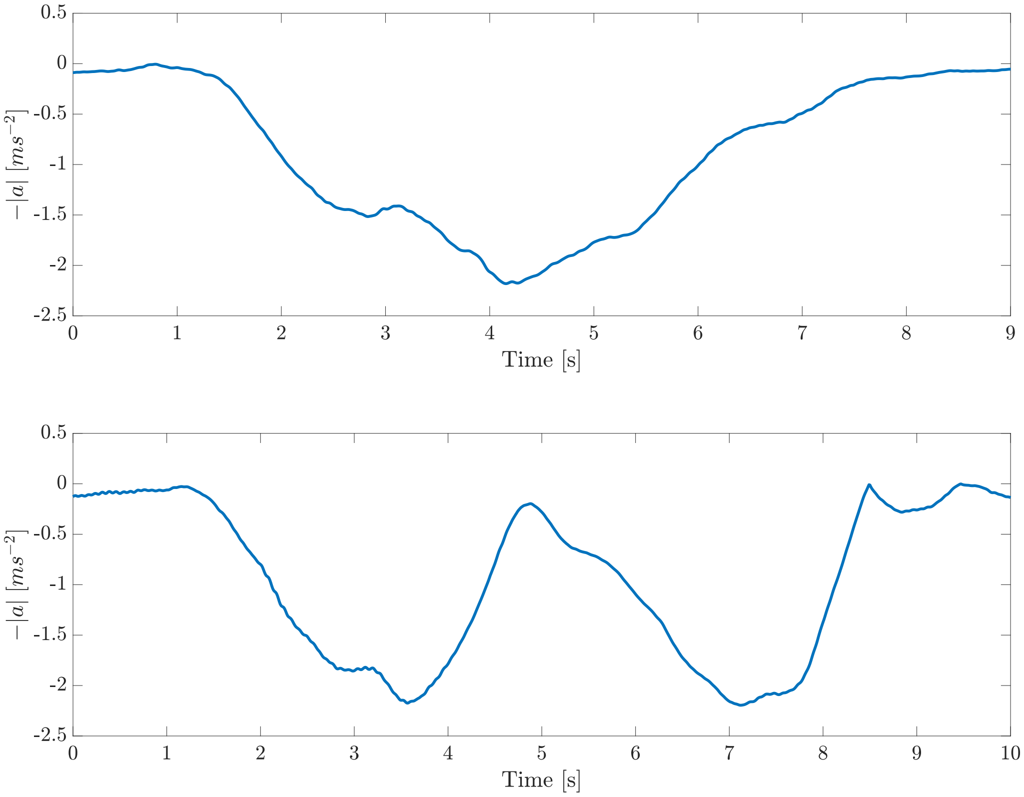

Figure 6: Tractor longitudinal acceleration for CVT4 (top) and CVT6 (bottom).

The metrics among different manoeuvre repetitions were quite concentrated around the median value, however, for a few tractors, a certain dispersion was observed mostly due to different behaviour between forward-to-rearward and rearward-to-forward manoeuvres. This was likely caused by a slightly different gear ratio between front and rear directions. Most of the calculated metrics were not correlated with each other, confirming that the metrics capture different aspects. Indeed, the Pearson's correlation was lower than 0.44 in the absolute value of all metrics pairs, except for the pair t_{sp}^{s} and a_{sp}^{s}, which was 0.73. This was probably caused by the fact that the greater was t_{sp}^{s} and the greater was the reduction of a during the engagement of the shuttling clutch and consequently the prominence of the shuttling peaks. Unexpectedly, there was no significant correlation between t_{sp}^{s} and t^{s} (the Pearson's correlation was 0.36), even if the greater was t_{sp}^{s} and the lower was the average value of a during manoeuvres and the longer was expected to be the time required for achieving the steady-state condition. One can expect that the shorter was the shuttling duration and the greater should be the irregularity in the acceleration signal (Raikwar et al., 2015); however, a small negative correlation between t^s and σ_a^s was observed (the Pearson's correlation coefficient was -0.32). Moreover, as expected, tractor mass is the parameter with the greatest correlation to t^s (the Pearson's correlation coefficient was 0.57), since during shuttling manoeuvres, the entire tractor mass must be inverted. 6 of 14 tractors consistently exhibited an overshoot peak in all manoeuvres with a sign change of a, while for 2 tractors this effect was seen only in a few manoeuvres. On average, tractors equipped with CVTs achieved 20% shorter t^s than those equipped with STs. This is probably caused by the ability of the transmission to exactly set the desired transmission ratio. In Figure 7, the range of $\hat{X}_*^s$ is reported for each tractor, and tractors are sorted by the mean value of $\hat{X}_*^s$. Tractor CVT5, that was ranked as the best one in the shuttling test, performed very well for all metrics since the range of $\hat{X}_*^s$ was markedly smaller than almost all the others, as $\hat{X}_*^s$ was greater than 0.6 for any metric. On the other hand, other tractors (e.g., ST7) showed a much higher range of $\hat{X}_*^s$ (i.e., from 0 to 1), but were placed in the middle of ranking. This high range occurred because the $-|a|$ did not exhibit any overshoot peak and, therefore, a_o^s was equal to 0 m s^{-2} (i.e., the highest achievable value). Tractor ST6 exhibited a high range, and the mean value of $\hat{X}_*^s$ was close to the maximum value of $\hat{X}_*^s$. This is because this tractor exhibited a smooth shuttling manoeuvre and also had the highest t^s (i.e., the worst value). As expected, the type of transmission does not have any influence on the quality of the shuttling manoeuvre, in contrast to the acceleration test, so CVT tractors were not placed in a certain region of the ranking. One can note that the two shuttling manoeuvres reported in Figure 6 were from the tractors ranked as the highest and lowest. Indeed, the two shuttling manoeuvres were selected on the basis of R^s.

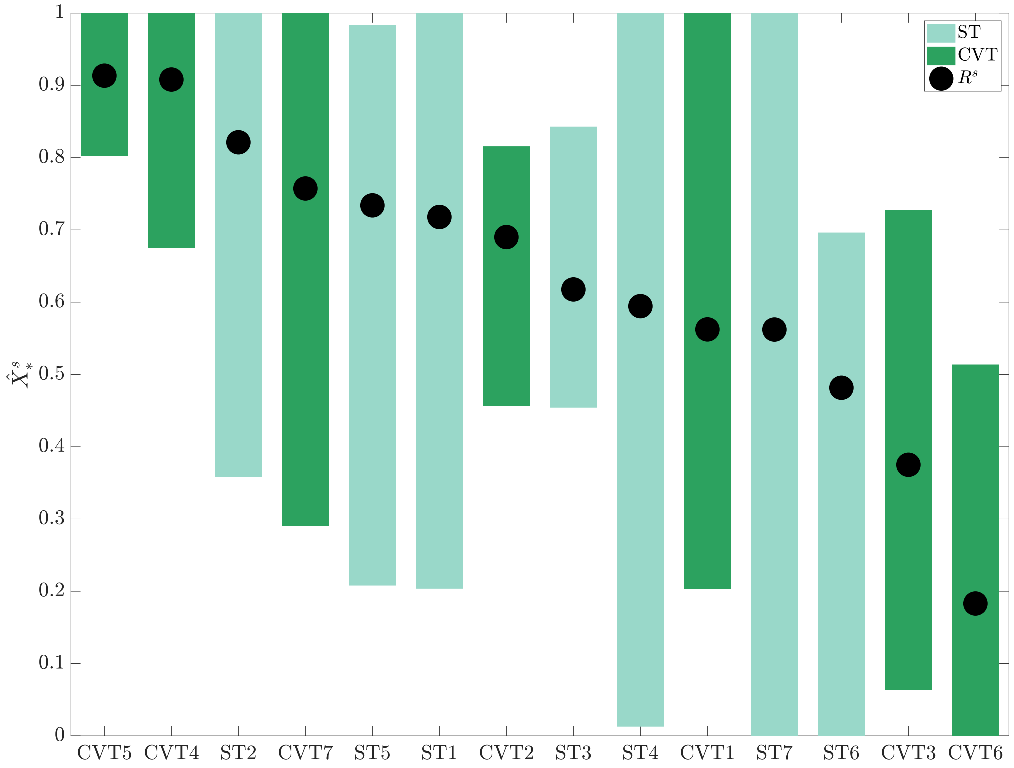

Figure 7: Range of scaled metrics of the shuttling test ($\hat{X}_*^s$), sorted by R^s.

Global analysis

In Figure 8, R is reported for all tractors. Considering the higher weight of R^s than that of $R^{0\to 40}$, tractors that ranked among the top of the shuttling test were also the ones that ranked higher in the global ranking, such as CVT4 and CVT5, while others were ranked lower since they had a lower ranking in the shuttling test, even if they performed very well in the acceleration test (e.g., CVT3 and CVT6). Although the first three positions of the ranking were occupied by CVT tractors, the last positions were also occupied by CVT tractors, leading to the conclusion that R was not affected by the type of transmission.

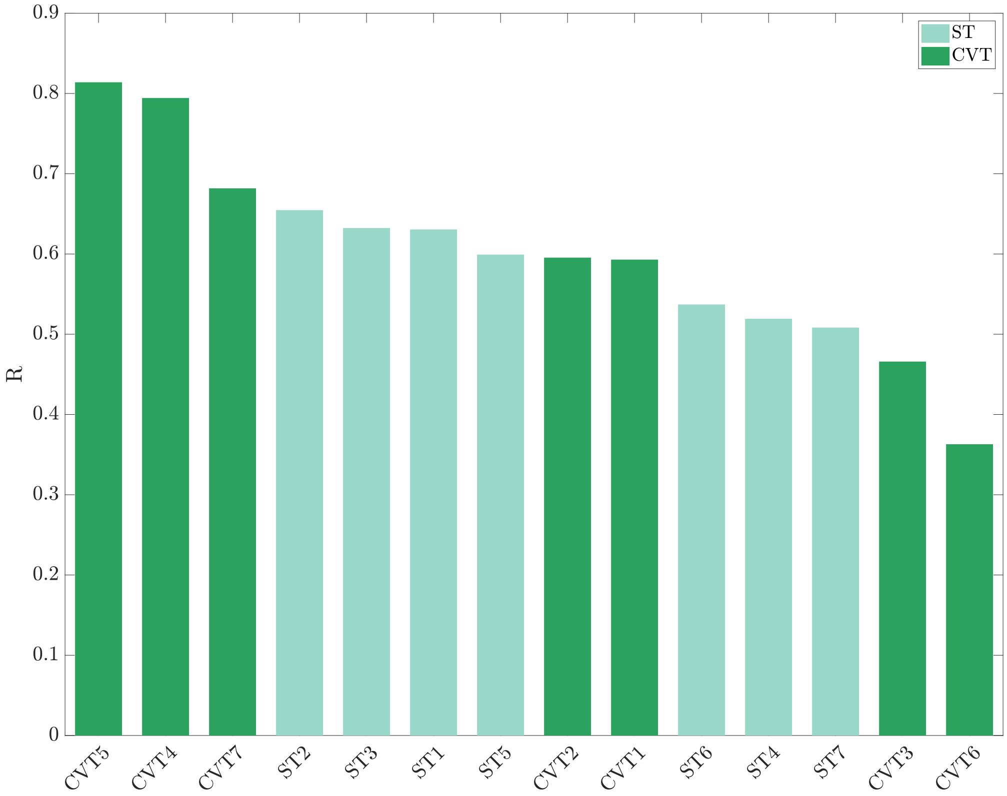

Figure 8: Value of R for all tractors.

4. Conclusions

Until the '90s, tractor transmissions had to be mainly efficient, durable, and reliable, but nowadays, they must also be highly functional. Therefore, the transmissions of modern tractors must be fully automated; thus, gearshifts are not anticipated by the operator, leading to a great perception of the gearshift process. For this reason, the gearshift process cannot be overlooked by engineers, especially nowadays when tractors are used more and more for transportation activities—in some cases, for up to 18% of their service time (Mattetti et al., 2021). This study was devoted to evaluating the quality of the gearshift. For high-quality gearshifts, gearshift actuators must be properly controlled to provide a smooth and quick gearshift. A simple procedure to evaluate the gearshift quality was developed and tuned based on two typical field manoeuvres. From each manoeuvre, a set of metrics able to objectively describe the irregularity caused by the gearshift process was reported, and their correlations with the main tractor parameters were calculated. It was found that CVTs provide a great benefit, especially for acceleration tests, in terms of both performance and comfort; on the other hand, stepped transmissions can provide a similar comfort level to that of CVT tractors, but at the expense of performance. The approach outlined in this study allows manufacturers to quickly spot the weak points of tractor gearshift behaviour so they can address specific investigations for improving the overall quality of the tractor and driver satisfaction. The method could be further improved by properly evaluating the latest features of modern gearshift transmissions, such as the ability of the transmission to bring the tractor to a standstill on slopes where the controller should set the tractor to operate with a transmission ratio of zero (Codecà et al., 2008).

In the future, an objective-subjective correlation should be carried out to find the optimal value of each metric and the weight of each metric on driver subjective rating. This will allow for the optimisation of the shift control strategy with numerical models when a prototype is not physically available.

Acknowledgements

The activities reported in this paper were supported by New Business Media Srl through the project titled “Test on tracks for measuring the performance and comfort of agricultural tractors.”

References

Biermann, J. W., Reitz, A., & Schumacher, T. (2000). *The Clonk Phenomenon—A Load Change Reaction to be Balanced in Terms of Comfort and Engine Response* (SAE Technical Paper N. 2000-05–0327). SAE International. https://www.sae.org/publications/technical-papers/content/2000-05-0327/

Birkmann, C., Fedde, T., & Frerichs, L. (2018). Drivetrain, chassis and tire-soil contact influence on power shift operations in standard tractors. *Landtechnik*, *73*(5), 146–161.

Blumenthal, U., Hansen, T., & Obermeier-Hartmann, R. (2000). *Claas CVT Development for Different Applications in Commercial Vehicles* (SAE Technical Paper N. 2000-01–2642). SAE International. https://doi.org/10.4271/2000-01-2642

Brenninger, M. M. (2007). *Fendt Vario CVT in Agricultural Tractors* (SAE Technical Paper N. 2007-01–4205). SAE International. https://doi.org/10.4271/2007-01-4205

Casoli, P., Vacca, A., Berta, G. L., Meleti, S., & Vescovini, M. (2007). *A numerical model for the simulation of Diesel/CVT power split transmission* (SAE Technical Paper N. 2007-24–0137). SAE International. https://doi.org/10.4271/2007-24-0137

Codecà, F., Tanelli, M., Savaresi, S. M., Frignati, D., Taroni, F., & Tascedda, L. (2008). Management of angular speed sensors for control at the power-zero condition of a continuously variable transmission: A describing-function approach. *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, *222*(12), 2269–2282. https://doi.org/10.1243/09544070JAUTO925

Elfes, L. E. (1961). *Development of a Hydraulically Shifted Tractor Transmission*. 610453. https://doi.org/10.4271/610453

Goetz, M., Levesley, M. C., & Crolla, D. A. (2005). Dynamics and control of gearshifts on twin-clutch transmissions. *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, *219*(8), 951–963. https://doi.org/10.1243/095440705X34720

Griffin, M. J. (1996). *Handbook of Human Vibration*. Elsevier.

Grotjahn, M., Quernheim, L., & Zemke, S. (2006). Modelling and identification of car driveline dynamics for anti-jerk controller design. *2006 IEEE International Conference on Mechatronics*, 131–136. https://doi.org/10.1109/ICMECH.2006.252510

Harris, K. J., & Jensen, J. K. (1964). *John Deere Power Shift Transmission* (SAE Technical Paper N. 640052). SAE International. https://doi.org/10.4271/640052

Huang, Q., & Wang, H. (2004). *Fundamental Study of Jerk: Evaluation of Shift Quality and Ride Comfort* (SAE Technical Paper N. 2004-01–2065). SAE International. https://doi.org/10.4271/2004-01-2065

ISO. (1997). *2631-1: Mechanical vibration and shock—Evaluation of human exposure to whole-body vibration—Part 1: General requirements* (N. 2631-1:1997; pagg. 1–31).

Jenkins, W. G. (1997). Principles of Product Planning for Worldwide Design. *ASAE Distinguished Lecture Series*, *21*, 1–16.

Kim, D. C., Kim, K. U., Park, Y. J., & Huh, J. Y. (2007). Analysis of shifting performance of power shuttle transmission. *Journal of Terramechanics*, *44*(1), 111–122. https://doi.org/10.1016/j.jterra.2006.04.001

Mattetti, M., Maraldi, M., Lenzini, N., Fiorati, S., Sereni, E., & Molari, G. (2021). Outlining the mission profile of agricultural tractors through CAN-BUS data analytics. *Computers and Electronics in Agriculture*, *184*, 106078. https://doi.org/10.1016/j.compag.2021.106078

Mattetti, M., Maraldi, M., Sedoni, E., & Molari, G. (2019). Optimal criteria for durability test of stepped transmissions of agricultural tractors. *Biosystems Engineering*, *178*, 145–155. https://doi.org/10.1016/j.biosystemseng.2018.11.014

Mattetti, M., Medici, M., Canavari, M., & Varani, M. (2022). CANBUS-enabled activity-based costing for leveraging farm management. *Computers and Electronics in Agriculture*, *194*, 106792. https://doi.org/10.1016/j.compag.2022.106792

Mattetti, M., Molari, G., & Sereni, E. (2017). Damage evaluation of driving events for agricultural tractors. *Computers and Electronics in Agriculture*, *135*, 328–337. https://doi.org/10.1016/j.compag.2017.01.018

Molari, G., Mattetti, M., Pesce, M., Grillo, M., Forte, M., & Sedoni, E. (2011). Evaluation of a Tractor's Driving Performance on the Road. *Transactions of the ASABE*, *54*(1), 13–23. https://doi.org/10.13031/2013.36248

Molari, G., & Sedoni, E. (2008). Experimental evaluation of power losses in a power-shift agricultural tractor transmission. *Biosystems Engineering*, *100*(2), 177–183. https://doi.org/10.1016/j.biosystemseng.2008.03.002

Naunheimer, H., Bertsche, B., Ryborz, J., & Novak, W. (2010). *Automotive Transmissions: Fundamentals, Selection, Design and Application*. Springer Science & Business Media.

Otten, R. L., & Wrisberg, W. V. (1990). *New Series of Mid-size Agricultural Tractors Engineered for Worldwide Markets* (SAE Technical Paper N. 901677). SAE International. https://doi.org/10.4271/901677

Raikwar, S., Tewari, V. K., Mukhopadhyay, S., Verma, Cmde. R. B., & Sreenivasulu Rao, M. (2015). Simulation of components of a power shuttle transmission system for an agricultural tractor. *Computers and Electronics in Agriculture*, *114*, 114–124. https://doi.org/10.1016/j.compag.2015.03.006

Rakha, H., Lucic, I., Demarchi, S. H., Setti, J. R., & Aerde, M. V. (2001). Vehicle Dynamics Model for Predicting Maximum Truck Acceleration Levels. *Journal of Transportation Engineering*, *127*(5), 418–425. https://doi.org/10.1061/(ASCE)0733-947X(2001)127:5(418)

Reddy, P., Darokar, K., Robinette, D., Shahbakhti, M., Blough, J., Ravichandran, M., Farmer, M., & Doering, J. (2019). *Control-Oriented Modeling of a Vehicle Drivetrain for Shuffle and Clunk Mitigation* (SAE Technical Paper N. 2019-01–0345). SAE International. https://doi.org/10.4271/2019-01-0345

Renius, K. Th. (2019). *Fundamentals of Tractor Design*. Springer Nature Switzerland AG.

Ross, L. F., & Panoushek, D. W. (1990). *Adaptive Electronic Tractor Shift Control System* (SAE Technical Paper N. 901592). SAE International. https://doi.org/10.4271/901592

Rossi, C., Pontara, D., & Casadei, D. (2014). E-CVT Power Split Transmission for Off-Road Hybrid-Electric Vehicles. *2014 IEEE Vehicle Power and Propulsion Conference (VPPC)*, 1–6. https://doi.org/10.1109/VPPC.2014.7007055

Searle, J. (1999). *Equations for Speed, Time and Distance for Vehicles Under Maximum Acceleration* (SAE Technical Paper N. 1999-01–0078). SAE International. https://doi.org/10.4271/1999-01-0078

Seeger, J. (2012). New dual clutch transmission for tractors; Neues Doppelkupplungsgetriebe fuer Traktoren. *ATZ offhighway*. https://www.osti.gov/etdeweb/biblio/22007246

Shin, C. W., Kim, H., Kim, M. K., Lim, W., & Cha, S. W. (2014). Development of an evaluation method for quantitative driveability in heavy-duty vehicles. *Journal of Mechanical Science and Technology*, *28*(5), 1615–1621. https://doi.org/10.1007/s12206-014-0107-5

Sorniotti, A., Galvagno, E., Morgando, A., Velardocchia, M., & Amisano, F. (2007). *An Objective Evaluation of the Comfort During the Gear Change Process* (SAE Technical Paper N. 2007-01–1584). SAE International. https://doi.org/10.4271/2007-01-1584

Tanelli, M., Panzani, G., Savaresi, S. M., & Pirola, C. (2011). Transmission control for power-shift agricultural tractors: Design and end-of-line automatic tuning. *Mechatronics*, *21*(1), 285–297. https://doi.org/10.1016/j.mechatronics.2010.11.006

Tinker, D. B. (1993). Integration of Tractor Engine, Transmission and Implement Depth Controls: Part 1, Transmissions. *Journal of Agricultural Engineering Research*, *54*(1), 1–27. https://doi.org/10.1006/jaer.1993.1001

Troncon, D., Alberti, L., & Mattetti, M. (2019). A Feasibility Study for Agriculture Tractors Electrification: Duty Cycles Simulation and Consumption Comparison. *2019 IEEE Transportation Electrification Conference and Expo (ITEC)*, 1–6. https://doi.org/10.1109/ITEC.2019.8790502

Varani, M., Mattetti, M., & Molari, G. (2021). Performance Evaluation of Electrically Driven Agricultural Implements Powered by an External Generator. *Agronomy*, *11*(8), 1447. https://doi.org/10.3390/agronomy11081447

Wen, C., Xie, B., Song, Z., Yang, Z., Dong, N., Han, J., Yang, Q., & Liu, J. (2021). Methodology for designing tractor accelerated structure tests for an indoor drum-type test bench. *Biosystems Engineering*, *205*, 1–26. https://doi.org/10.1016/j.biosystemseng.2021.02.007

Wheals, J. C., Crewe, C., Ramsbottom, M., Rook, S., & Westby, M. (2002). *Automated Manual Transmissions—A European Survey and Proposed Quality Shift Metrics* (SAE Technical Paper N. 2002-01–0929). SAE International. https://doi.org/10.4271/2002-01-0929

Xia, G., Yu, X., Tang, X., Zhao, L., & Sun, B. (2020). Power Shift Control of High-Power Tractor Based on Pattern Search. *Applied Engineering in Agriculture*, *36*, 565–578. https://doi.org/10.13031/aea.13876

Yin, X., Zhong, Y., Wu, X., & Lu, H. (2014). *Shift Quality Improvement through Integrated Control of Dual Clutches Pressure and Engine Speed for DCT* (SAE Technical Paper N. 2014-01–1748). SAE International. https://doi.org/10.4271/2014-01-1748

Appendix

Utilizing the dataset used by the Authors in a previous paper (Mattetti et al., 2022), which contains 1323 hours of real-world data, the shuttling and on-road acceleration manoeuvres were detected and counted. In particular, a sliding window approach was used for the detection of manoeuvres (Mattetti et al., 2017). The length and shifting factor of the sliding window were 25 s and 2 s, respectively. In the sliding window, the minimum and maximum value of the tractor's ground speed (V_t) was calculated. The two driving manoeuvres were detected by evaluating the rules reported in Table 2 in each sliding window.

Table 2: Rules adopted for the detection of the manoeuvres in real-world data

Manoeuvre	Rule
Acceleration	$max(V_t) - min(V_t) > 30\ km\ h^{-1}$
Shuttling	$max(V_t) - min(V_t) > 5\ km\ h^{-1}\ AND\ \max(V_t)\,\min(V_t) < 0$

To avoid the same manoeuvre being counted twice, a minimum temporal distance of 2 min between two detected manoeuvres was imposed. From the dataset, 8336 manoeuvres were counted, with 30% of them being classified as acceleration manoeuvres while the others were classified as shuttling manoeuvres.