Testing Fuel Efficiency of Tractors with both Continuously Variable and Standard Geared Transmissions

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Abstract. *A John Deere 8295R IVT tractor with a continuously variable transmission (CVT) and a John Deere 8295R PowerShift (PST) tractor (Waterloo, Iowa) with a standard geared transmission (GT) were tested for fuel consumption at three different travel speeds with six different load levels applied per speed. The JD 8295R PST tractor was tested both at full throttle (FT) and shifted up two gears and throttled back (SUTB) to achieve the same travel speed as at full throttle conditions. For each travel speed with each transmission mode, fuel consumption was determined to be linearly related to drawbar power. Linear regression results showed that the tractor with the CVT was more fuel efficient than the tractor with the GT at FT when the power was below 76% to 81% of maximum drawbar power depending on the travel speed. The results also showed that above 37% to 52% of maximum drawbar power, the GT at SUTB was more fuel efficient than the CVT equipped tractor. As travel speed increased, the percent of maximum power below which the CVT was significantly more fuel efficient than the GT at FT decreased slightly. Likewise, the percent of maximum power above which the GT at SUTB was more fuel efficient than the CVT decreased as speed increased. In order to determine differences in fuel consumption between the tractor transmission operating modes, testing with at least three loads and at least three travel speeds is recommended. Additional testing is needed on other models of tractors from other manufacturers to determine whether the trends found in this study pertain to all CVT equipped tractors or if they are specific to this tractor model and manufacturer.*

Keywords. *Tractor testing, Continuously Variable Transmission (CVT), Geared Transmission, Fuel Consumption, Fuel Efficiency.*

TESTING FUEL EFFICIENCY OF TRACTORS WITH BOTH CONTINUOUSLY VARIABLE AND STANDARD GEARED TRANSMISSIONS

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INTRODUCTION

Testing tractors to ensure they meet their advertised performance claims has been a central focus of the Nebraska Tractor Test Lab since the Nebraska Tractor Test Law passed in 1919. Besides ensuring that tractors meet their advertised performance claims, the standardized test protocol developed as a result of the law allows a means of comparison between tractors of different makes and models, as well. Since 1919, tractors have advanced significantly and are now available with numerous options.

One of these options for some tractors is the choice of different transmissions. Many tractor models are now available with both standard geared transmissions (GTs) and continuously variable transmissions (CVTs). Unlike traditional geared transmissions that operate using a series of fixed gear ratios, CVTs have the ability to operate over an infinite number of gear ratios within a certain range. They are equipped with control systems that have the capability to adjust the transmission ratio and engine speed to operate at the point of maximum fuel efficiency for the given conditions as described in Renius and Resch (2005). This approach is based on the "shift-up, throttle-back" (SUTB) or "gear up and throttle down" approach to driving a conventional geared transmission as described in Grisso and Pitman (2009). If less than full power is required, the same amount of required power can be developed with increased fuel efficiency by using a lower engine speed and a higher gear ratio. The CVT, ideally, is capable of giving the same performance as a standard geared transmission operated under SUTB conditions, but without the operator having to experiment to find the optimum gear/throttle position combination. The CVT has the additional advantage over the GT of being able to choose a more efficient transmission ratio than the limited discrete ratios available with GTs.

CURRENT TESTING PRACTICES

There is currently only a minimal standardized test protocol in place that allows the comparison of fuel efficiency between tractors that are available with both CVTs and GTs at settings other than full throttle. The Organisation for Economic Co-operation and Development (OECD) oversees the development and maintenance of world-wide tractor testing standards. Currently, the OECD Code 2 standard for official testing of agricultural and forestry tractors (OECD, 2010) is globally used as the standard by which tractors are tested. In the drawbar tests, only two points below maximum power (50% and 75% of pull at maximum power) are tested for fuel consumption comparisons. A test procedure that compares fuel efficiency between these two types of tractors over a range of loads would provide useful information both for the consumer looking to buy a new tractor and to the manufacturer who is looking to advertise the benefits of the different transmission options available.

Efforts have been made to develop a test procedure for comparing the fuel efficiency of tractors equipped with standard geared transmissions and CVTs. Coffman et al. (2010) performed drawbar testing on a John Deere 8530 IVT tractor in both manual and automatic modes. From this study, it was found that the order in which the loads were applied did not affect the steady state results. Also, the CVT operating at reduced engine speed in automatic mode was more efficient than the CVT operating at full throttle in manual mode at loads less than 78% of maximum power at rated engine speed. However, the performance of a CVT transmission operating in manual mode may not be the same as the performance of an actual geared transmission.

The German Agricultural Society (DLG) Test Center (Groß-Umstadt, Germany) has been developing a new test that can account for varying levels of drawbar load, PTO load, and hydraulic load all at the same time (Degrell and Feuerstein, 2005). This test, named the "DLG-PowerMix," uses 8 different load cycles to simulate the entire range of uses for an agriculture tractor. Each load cycle consists of a dynamic load curve that is applied over a fixed amount of time that can incorporate drawbar pull, PTO torque, hydraulic power, or any combination of the three depending on the type of work simulated. Theoretically, this test, using strictly drawbar loading, could compare the fuel efficiency between a tractor equipped with a standard geared transmission and a tractor equipped with a CVT. However, due to the dynamic load curve, it would be very difficult to replicate the test using a different load car (at a different test station) due to differences in the load car controllers and components. Also, the load cycles that DLG has chosen may not be appropriate for typical North American row-crop farming operations.

TRACTOR LOADING

In typical farming operations a single tractor may pull a variety of different implements with varying power requirements. Research has been conducted that illustrates the average power required to pull certain implements. Rickets and Weber (1961) conducted research to study the engine horsepower output of a single tractor for several farm operations. They found that operations that farmers generally called heavy work varied from 56 to 97 percent of the maximum horsepower available from the tractor at full throttle. Research was performed by McLaughlin et al. (2008) to determine the energy inputs for eight primary tillage implements applied to a clay loam soil over a four year period (2002-2005). The eight primary tillage implements included deep zone till, moldboard plow, chisel sweep, disk ripper, chisel plow, shallow zone till, fluted coulter, and disk harrow. The tractor used for this testing was a Case IH 7110 and the range of the tractor-implement matches was considered by the authors to be “typical” of that found on many farms. Values corresponding to a range of 26.4% to 81.4% of available tractor power with an average value of 51.5% were needed to pull these implements.

Changing soil conditions and topography play a significant role in determining the required drawbar power. One study on the spatial mapping of tillage energy (McLaughlin and Burt, 2000) showed that the draft force required to pull a combination disk-ripper varied significantly with respect to location in an agricultural field composed of clay-loam soil. The average maximum and minimum percentage of full power used was 46.6% and 28.0%, respectively. Due to the fact that averaged values from the spatial plot legends were used, the true maximum and minimum power values are most likely higher and lower, respectively than the calculated average maximum and minimum power values required to pull the disk-ripper.

Several other researchers have mapped soil mechanical resistance in agricultural fields with corn-soybean rotations. Results in Chung et al. (2008) showed minimum-to-maximum soil resistance ratios of 0.57 and 0.64. Results in Siefken et al. (2005) showed a minimum-to-maximum soil resistance value of 0.50 in fields that had previously been no-till. Likewise, results in Adamchuck et al. (2008) showed minimum-to-maximum soil resistance values of 0.45 and 0.55 for a field that had been in a no-till rotation for more than ten years. The types of soil varied widely for these studies and the minimum-to-maximum soil resistance values reported here are most likely slightly lower than what was actually experienced in the field due to the fact that averaged values from spatial map legends were used to calculate them. However, between the tillage energy study and the soil mechanical resistance studies, it was demonstrated that the amount of power needed to pull an implement can vary greatly within a field.

OBJECTIVES

The ultimate goal of this research was to be able to recommend an optional test procedure that can be added to the OECD Code 2 for determining the fuel efficiency of a CVT transmission at varying drawbar load levels. However, the specific objectives of this research were 1) to determine the partial load level at which statistically significant fuel consumption differences occur between the tractor equipped with the CVT and the tractor equipped with the GT operated at full throttle (FT), 2) to determine the partial load level at which statistically significant fuel consumption differences occur between the tractor equipped with the CVT and the tractor equipped with the GT operated under “shift-up-throttle-back” (SUTB) conditions, and 3) to determine if significantly different fuel consumption results are obtained when different travel speeds are tested.

MATERIALS AND METHODS

With support from Deere and Company (Waterloo, Iowa) and the Nebraska Tractor Test Laboratory (NTTL) (Lincoln, Nebraska), two large row-crop tractors were tested on the concrete test track of the NTTL (Figure 1), located in Lincoln, Nebraska (40° 49’ N, 96° 40’ W), with an elevation of 355 m.

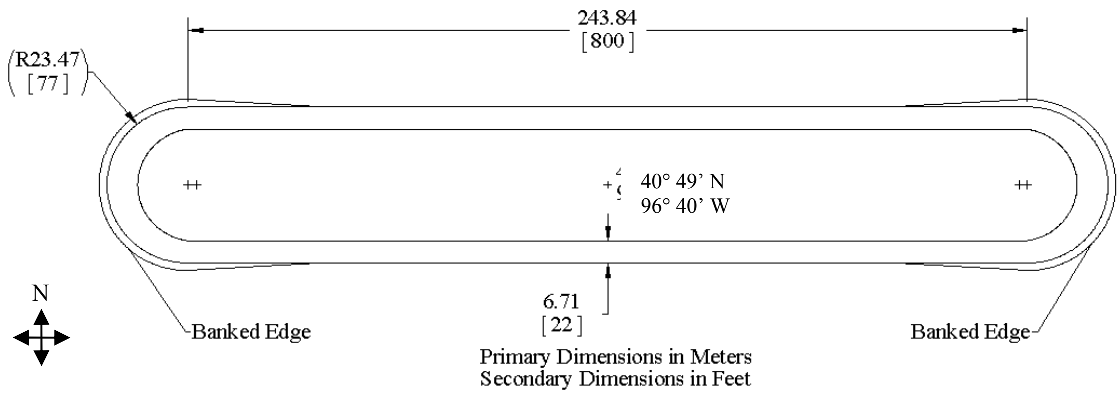

Figure 1. Test track at the Nebraska Tractor Test Laboratory, Lincoln, NE.

The two tractors tested were the John Deere 8295R PowerShift (PST) and the John Deere 8295R Infinitely Variable Transmission (IVT) tractors [Deere uses the term PowerShift (PST) to describe their version of a geared transmission (GT) and Infinitely Variable Transmission (IVT) to describe their version of a Continuously Variable Transmission (CVT)]. The tractors were ballasted to a common ballast configuration of 75 kg/PTO-kW with a weight split of 41%/59%, using the supplied tractor weights. This means that 41% of the tractor weight was on the front axle and 59% of the tractor weight was on the rear axle, which is a typical ballast configuration for Mechanical Front Wheel Drive (MFWD) row-crop tractors. The same Goodyear Dyna Torque Radial tires were used throughout the testing, which took place from June 4, 2010 to June 8, 2010.

EXPERIMENTAL DESIGN

There are two main ways of operating a standard geared transmission. The first is to simply pick the gear that will give the desired travel speed when the engine is at full throttle, and operate at full throttle. The other method is to select a gear that will give the desired travel speed with the engine at a reduced throttle setting but still with enough power to pull the load. The CVT transmission is designed to automatically and continuously select the optimum engine speed to maximize fuel efficiency and to produce the desired travel speed through the field. Therefore, it was decided to compare the two tractors at three different modes of tractor operation: 1) standard geared transmission with engine at full throttle (GT at FT), 2) standard geared transmission shifted up two gears and throttled back (GT at SUTB) to achieve the same forward speeds as in (1), and 3) CVT in automatic mode [controller set to allow engine speed to vary between 1200 rpm and full throttle depending on loading conditions (Deere and Co., 2009)] with set point travel speeds set to achieve the same speeds as in (1).

It was decided to test the tractors at six load levels ranging from 30% to 80% of drawbar load at maximum power in 10% increments based on the tractor loading research reported in the literature. There are already required tests in place that test the tractors at maximum power so it was deemed unnecessary to test the tractors at maximum power again. A speed range of 5 $km \cdot h^{-1}$ to 11 $km \cdot h^{-1}$ was chosen to encompass a wide variety of field applications. It was decided to pick three speeds out of this range for testing. Three speeds and six loads gave a total of 18 treatment combinations. To implement these treatment combinations, a Split-Plot Design with the whole plots arranged in randomized complete blocks was used. The main plot factor was speed and the subplot factor was load. Four replications were achieved by blocking by time. More detailed information on the load application order can be found in Howard (2010).

The three speeds were chosen based on the maximum speeds achieved in 6th, 8th and 10th gears for the John Deere 8295R PST tractor. The maximum speeds corresponding to these gears were 5.94 $km \cdot h^{-1}$, 7.97 $km \cdot h^{-1}$ and 10.64 $km \cdot h^{-1}$. According to the Nebraska Tractor Test Report (NTTL, 2010) for this tractor, the pull at maximum power for these gears was 107.40 kN, 80.02 kN and 58.42 kN, respectively. Therefore, these loads were used to determine the six partial load levels (30%, 40%, 50%, 60%, 70%, and 80% of pull at maximum power for each gear) at which both tractors were tested.

The testing was performed in a clockwise travel direction around the test track. All vehicles traveled on the flat portion of the track, not on the banked portions shown in Figure 1. At the start of the day, multiple warm-up rounds were completed to make sure that the tractor was at steady state operating conditions before the actual testing was conducted. Steady state operating conditions were met once the hydraulic temperature had reached its normal operating temperature. Once the tractor had reached steady state operating conditions, data collection began with the first load to be applied for the first speed in the first block.

The loads were tested by recording data over a 60.96 m (200 ft) length of straightaway on each side of the track for each load and averaging the results over that length. Therefore, two data runs could be taken per straight side of the track, as shown in Figure 2. Around the corners, the load car load controller was set to apply a pause load. This pause load was set to the same load as the load being tested, unless that load was greater than 66.72 kN (15,000 lb), to minimize the amount of transition coming out of the corners. For set point loads above 66.72 kN, a possibly damaging amount of side load might be applied to the tractor therefore the pause load was limited to a maximum of 66.72 kN.

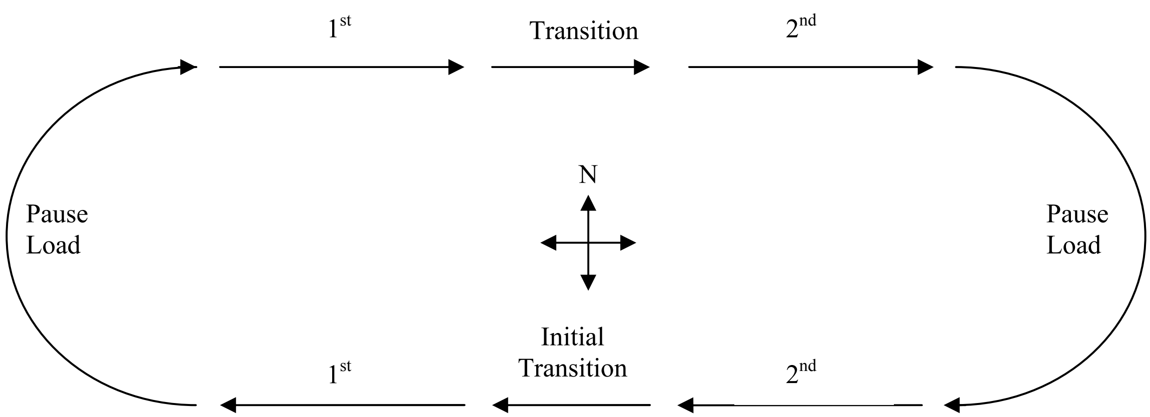

Figure 2. Test pattern used (trend continues for 3rd through 6th loads in randomized sequence of each block).

If comparable results were achieved on both the north and the south side of the track, then the next load set point was applied. If the results were not comparable then more data were collected until there was one north and one south run that showed comparable results. Drawbar power and fuel consumption values were used to determine whether the results were "comparable" or not. If the drawbar power values were within 0.75 kW of each other and the fuel consumption values were within 0.23 $kg \cdot h^{-1}$ of each other, then the results were deemed to be comparable. This trend continued until all six loads for the given speed had been tested. This process was then repeated for the next speed.

The GT tractor at FT was tested on June 4, 2010; the GT tractor at SUTB was tested on June 5, 2010; and the CVT tractor was tested on June 8, 2010, after the wheels and tires had been switched over from the GT tractor.

TEST EQUIPMENT

The test was conducted using the NTTL instrumented drawbar load car, which was equipped with a National Instruments data acquisition and load control system running LabVIEW v8.6 software (National Instruments Corp., Austin, Texas). A modified John Deere 5020 tractor was also pulled behind the load car during testing to provide additional drawbar load to that of the load car.

The drawbar load was measured using a hydraulic cylinder in the linkage between the load car and the tractor. The pressure in the cylinder was measured using a pressure transducer and then converted to force using the known cross-sectional area of the cylinder. Travel speed was measured using an unpowered "fifth wheel" which traveled under the load car. The rear axle speed of the tractor was also measured. The volumetric fuel flow rate was measured using a positive displacement flow meter, which was converted to a mass flow rate using the specific weight of the fuel (0.842 kg·l^{-1}). The engine and fan speed were measured using fiber optic sensors and the turbocharger boost was measured using a pressure transducer. Various temperatures were collected, as well, using K-type thermocouples. These temperature measurements included: fuel inlet and return temperatures, engine coolant temperature, engine oil temperature, air inlet temperature to the engine and the hydraulic oil temperature.

DATA ANALYSIS

There were small variations in the forward travel speeds and the applied loads for the different set-point loads and speeds. Because speed and load could not be set consistently at the same values, the relationship between hourly fuel consumption and drawbar power was found using regression analysis instead of Analysis of Variance (ANOVA) or some other approach. The same model was used to fit the fuel consumption curves for all three tractor operating modes for each individual speed and is shown below:

$$Q_{i,j} = \beta_0 + \beta_1 \cdot P_j + \beta_2 \cdot M_1 + \beta_3 \cdot M_2 + \beta_4 \cdot P_j \cdot M_1 + \beta_5 \cdot P_j \cdot M_2 + \varepsilon_j \quad (1)$$

where,

$Q_{i,j}$ = measured fuel consumption with speed i for the jth observation (kg·h^{-1})
$\beta_{0,\ldots,5}$ = Slope (kg·h^{-1}·kW^{-1}) and intercept (kg·h^{-1}) terms
P_j = actual drawbar power, jth observation (kW)
M = mode of operation

$M_1 = \begin{cases} 1 \textit{ for GT at FT} \\ 0 \textit{ otherwise} \end{cases}$

$M_2 = \begin{cases} 1 \textit{ for GT at SUTB} \\ 0 \textit{ otherwise} \end{cases}$

e_j = random error
i = 1, 2, 3 corresponding to speeds 1, 2 and 3, respectively,
j = observation number.

No differences in fuel consumption were found between the blocks so they were dropped from the model, which was implemented using SAS (SAS Institute Inc, Cary, NC). This model allowed the comparison of the differences in predicted fuel consumption values between the GT at FT and the CVT as well as between the GT at SUTB and the CVT. Using an alpha level of 0.05, the power level at which there was a significant difference between the predicted fuel consumption values for the different transmission operation modes was determined. The power level at which a significant difference was detected was compared to the maximum power for each speed to find the percent of maximum power at which the significant difference occurred. The percent of maximum power was plotted against travel speed to detect whether there was any trend based on travel speed. In addition to the regression analysis, residual analysis was performed to make sure that the regression model assumptions were not violated.

A similar model was used to compare the predicted fuel consumption values at different travel speeds for each transmission operating mode. Instead of representing transmission mode, the M values represented travel speed:

$M_1 = \begin{cases} 1 \textit{ for Speed } 1 \\ 0 \textit{ otherwise} \end{cases}$

$M_2 = \begin{cases} 1 \textit{ for Speed } 2 \\ 0 \textit{ otherwise} \end{cases}$

i = 1, 2, 3 corresponding to transmission modes GT at FT, GT at SUTB and CVT, respectively

As for the transmission mode comparison, the power level at which there was a significant difference between the predicted fuel consumption values for the different travel speeds was determined using an alpha value of 0.05.

RESULTS AND DISCUSSION

Regression analysis of the relationship between fuel consumption and drawbar power produced the following models for Speeds 1, 2 and 3, respectively:

$$\hat{Q}_1 = 2.565 + 0.250 \cdot P + 5.927 \cdot M_1 - 0.041 \cdot P \cdot M_1 + 2.095 \cdot M_2 - 0.031 \cdot P \cdot M_2 \quad (2)$$

$$\hat{Q}_2 = 4.141 + 0.239 \cdot P + 5.236 \cdot M_1 - 0.035 \cdot P \cdot M_1 + 1.051 \cdot M_2 - 0.024 \cdot P \cdot M_2 \quad (3)$$

$$\hat{Q}_3 = 5.205 + 0.240 \cdot P + 4.801 \cdot M_1 - 0.034 \cdot P \cdot M_1 + 0.845 \cdot M_2 - 0.024 \cdot P \cdot M_2 \quad (4)$$

where, $\hat{Q}$ = predicted fuel consumption (kg·h^{-1}). Separating the modes of transmission operation, these models can be rewritten as:

$$\hat{Q}_1 = \begin{cases} 8.49 + 0.209 \cdot P & \textbf{for GT at FT} \\ 4.66 + 0.219 \cdot P & \textbf{for GT at SUTB} \\ 2.56 + 0.250 \cdot P & \textbf{for CVT} \end{cases} \quad (5)$$

$$\hat{Q}_2 = \begin{cases} 9.38 + 0.204 \cdot P & \textbf{for GT at FT} \\ 5.19 + 0.215 \cdot P & \textbf{for GT at SUTB} \\ 4.14 + 0.239 \cdot P & \textbf{for CVT} \end{cases} \quad (6)$$

$$\hat{Q}_3 = \begin{cases} 10.01 + 0.206 \cdot P & \textbf{for GT at FT} \\ 6.05 + 0.216 \cdot P & \textbf{for GT at SUTB} \\ 5.20 + 0.240 \cdot P & \textbf{for CVT} \end{cases} \quad (7)$$

The measured fuel consumption data and the predicted models are shown in Figure 3. The fuel consumption values for the GT at FT and GT at SUTB are almost parallel, with the GT at FT having higher fuel consumption values at each power level. Since the GT at SUTB will always be more fuel efficient than running at FT, no further comparison was done between these two operating modes. The coefficients of determination (R^2) values for these lines were found to be 0.993 for Speed 1 and 0.995 for Speeds 2 and 3. There were no discernable trends with respect to drawbar power in the analysis of the fuel consumption prediction errors (Howard, 2010).

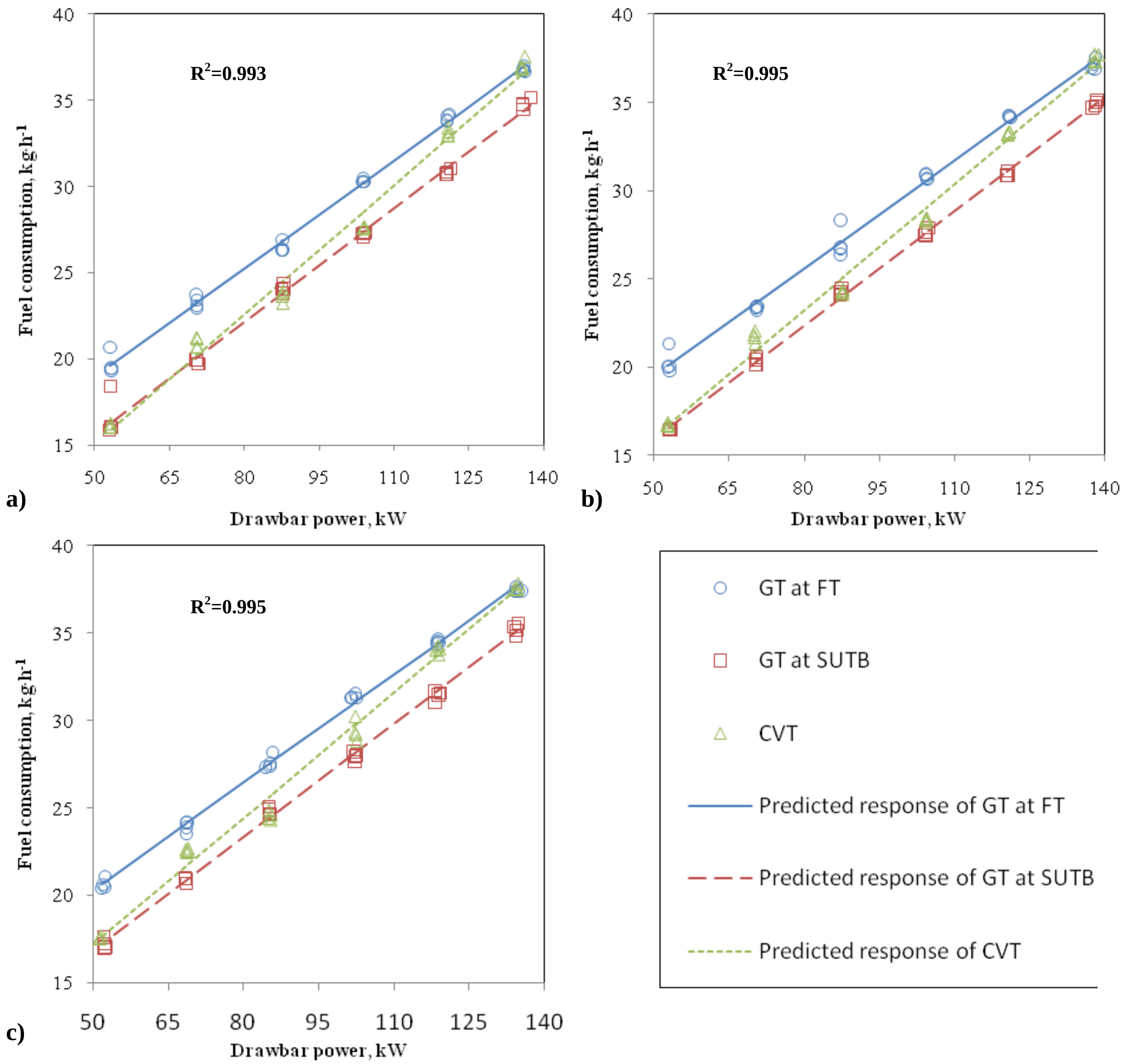

Figure 3. Hourly fuel consumption response to drawbar power for a John Deere 8295R PST and a John Deere 8295R IVT at (a) Speed 1, (b) Speed 2 and (c) Speed 3.

The difference between predicted fuel consumption values for the three transmission modes as a function of drawbar power was plotted for the three different travel speeds (Figure 4). Based on the analysis of fuel consumption difference between the GT at FT and the CVT in automatic mode, shown in Figure 4a, c and e, the fuel savings of using the CVT in automatic mode increased as the power level decreased, but the fuel consumption was similar at higher loads. A comparison of the values of the predicted fuel consumption difference between the GT at FT and the CVT ($\hat{Q}$ for the GT at FT minus $\hat{Q}$ for the CVT) with the 95% confidence interval for this difference showed that the CVT reduced fuel consumption significantly below certain power levels. The CVT was more fuel efficient below 128 kW for Speed 1, 131 kW for Speed 2 and 124 kW for Speed 3, which corresponded to 81%, 79% and 76%, respectively, of the maximum drawbar power obtained during the unballasted portion of the official OECD test (NTTL, 2010), as shown in Table 1.

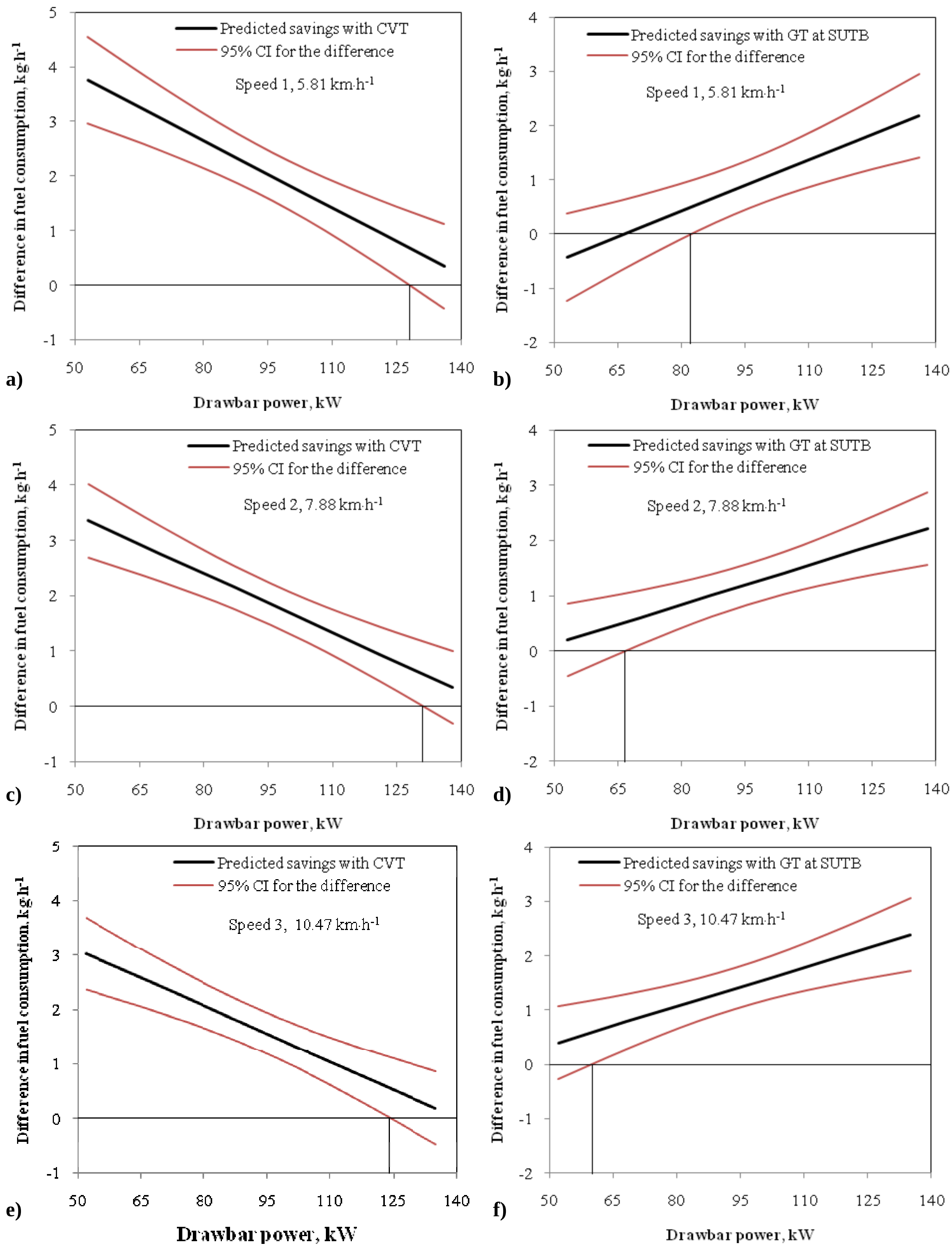

Figure 4. Difference in hourly predicted fuel consumption response to drawbar power between the GT at FT and the CVT (GT at FT – CVT) for (a) Speed 1 (5.81 km·h^{-1}), (c) Speed 2 (7.88 km·h^{-1}), and (e) Speed 3 (10.47 km·h^{-1}), also between the CVT and the GT at SUTB (CVT – GT at SUTB) for (b) Speed 1 (5.81 km·h^{-1}), (d) Speed 2 (7.88 km·h^{-1}), and (f) Speed 3 (10.47 km·h^{-1}).

Table 1.Power levels below which the CVT was more fuel efficient than the GT at FT and above which the GT at SUTB was more fuel efficient than the CVT.

	Average forward travel speed from CVT fuel efficiency test ($km \cdot h^{-1}$)	Maximum drawbar power of JD 8295R PST from official OECD test (kW)	Highest power level at which fuel consumption for CVT < for GT at FT (kW)	Percent of maximum power at selected speed (%)	Lowest power level at which fuel consumption for GT at SUTB < for CVT (kW)	Percent of maximum power at selected speed (%)
Speed 1	5.81	158.10	128	81.0	82	51.9
Speed 2	7.88	165.58	131	79.1	66.5	40.2
Speed 3	10.47	163.26	124	75.9	60	36.7
Average	-	-	127.7	78.7	69.5	42.9

The analysis of difference of fuel consumption between the GT at SUTB and the CVT showed that the GT at SUTB was more fuel efficient at higher loads, but the fuel consumption was similar at lower loads as shown in Figure 4b, d, and f. A comparison of the values of the predicted fuel consumption difference between the CVT and the GT at SUTB ($\hat{Q}$ for the CVT minus $\hat{Q}$ for the GT at SUTB) with the 95% confidence interval for this difference showed that the GT at SUTB had significantly lower fuel consumption above certain power levels. The GT at SUTB became more fuel efficient above 82 kW for Speed 1, 66.5 kW for Speed 2 and 60 kW for Speed 3, which corresponded to 52%, 40% and 37%, respectively, of the maximum drawbar power obtained during the unballasted portion of the official OECD test (NTTL, 2010), as shown in Table 1.

In general, at drawbar power levels less than 75 percent of maximum, the CVT was more fuel efficient than the GT at FT. For drawbar power levels above 35 to 50 percent depending on speed, the GT at SUTB was more fuel efficient than the CVT within the power range tested. Field operations often require 30% to 80% of maximum power and for much of this range, the GT at SUTB consumed fuel at the lowest rate. If the fuel use data is displayed, some operators may choose to actively engage in changing gears and throttle settings to achieve the lowest fuel use rate. However, some operators may prefer to minimize shifting gears with the tractor under load as they believe the shock loading that occurs with these gear shifts to be damaging to the tractor. In situations where the operator chooses not to use a GT at SUTB, the tractor with a CVT transmission will be more fuel efficient than a GT at FT.

There was a linearly decreasing trend for both the percent of maximum drawbar power below which the CVT was found to be more fuel efficient than the GT at FT and the percent of maximum drawbar power above which the GT at SUTB was found to be more fuel efficient than the CVT. Therefore, as speed increased, the percent of maximum power below which the CVT was significantly more fuel efficient than the GT at FT decreased slightly. Likewise, the percent of maximum power above which the GT at SUTB was more fuel efficient than the CVT decreased as speed increased.

Results from the analysis of the differences between predicted fuel consumption values at the different speed levels are shown in Figure 5. A comparison of the values of the predicted fuel consumption difference between Speed 1 and Speed 2 ($\hat{Q}$ for Speed 2 minus $\hat{Q}$ for Speed 1) with the 95% confidence interval for this difference showed that operating at Speed 1 produced significantly lower fuel consumption values for certain power ranges with certain transmission modes (Figure 5a, c and e). For the GT at FT, operating at Speed 1 instead of Speed 2 consumed fuel at a lower rate for drawbar power levels below 93 kW. For the GT at SUTB, there was no significant difference in fuel consumption between Speeds 1 and 2. For the CVT, operating at Speed 1 instead of Speed 2 consumed fuel at a lower rate for drawbar power levels between 58 kW and 85 kW.

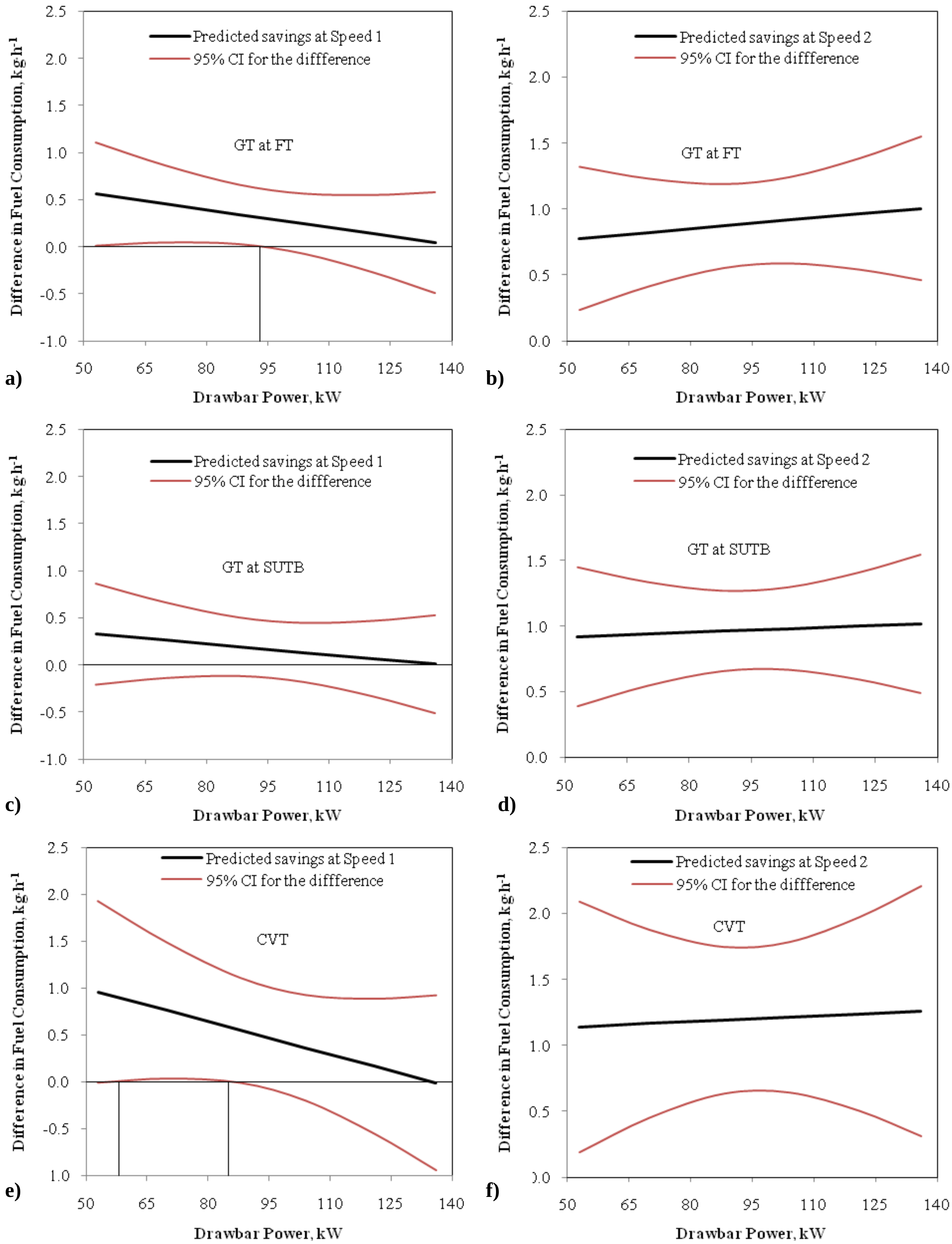

Figure 5. Difference in hourly predicted fuel consumption response to drawbar power between Speeds 1 and 2 (Speed 2 – Speed 1) for (a) GT at FT, (c) GT at SUTB and (e) CVT, and between Speeds 2 and 3 (Speed 3 – Speed 2) for (b) GT at FT, (d) GT at SUTB and (f) CVT.

A comparison of the values of the predicted fuel consumption difference between Speed 2 and Speed 3 ($\bar{Q}$ for Speed 3 minus $\bar{Q}$ for Speed 2) with the 95% confidence interval for this difference showed that operating at Speed 2 consumed fuel at a lower rate for all three transmission modes (Figure 5b, d and f). Since a significant difference was found between Speeds 2 and 3, no analysis was performed between Speeds 1 and 3 because Speed 1 was guaranteed to produce significantly lower fuel consumption values than Speed 3.

The average difference in hourly fuel consumption between Speed 3 and Speed 2 was 2.59 $km \cdot h^{-1}$, while the average difference between Speed 2 and Speed 1 was 2.07 $km \cdot h^{-1}$. The smaller difference between Speeds 1 and 2 may be the reason that the predicted fuel consumption values were not all significantly different. Even though the predicted fuel consumption values for Speeds 1 and 2 were not always significantly different, this analysis still shows that there are differences in fuel consumption based on travel speed and that multiple speeds should be tested to determine predicted fuel consumption values for different field applications.

In an effort to gain a deeper understanding of why the transmission operating modes differ where they do, an investigation was carried out on the engine speed of the tractors in relationship to drawbar load, as shown in Figure 6. The engine speeds as a function of drawbar power for the GT at FT and the GT at SUTB seem parallel as did the fuel consumption lines. However, there were noticeable differences in slope between the GT at FT and the CVT as well as between the GT at SUTB and the CVT. The differences between the engine speeds at the point where the two transmissions were found to produce significantly different fuel consumption results are marked by vertical lines and these differences are presented in Table 2.

As shown in Table 2, the minimum difference (GT at FT – CVT) in engine speed at which fuel consumption for the CVT became significantly less than for the GT at FT decreased as travel speed increased. Conversely, the maximum difference (GT at SUTB – CVT) in engine speed below which fuel consumption for the GT at SUTB was significantly less than for the CVT increased as travel speed increased. In general, the CVT equipped tractor had to reduce its engine speed roughly 400 rpm to 450 rpm below that of the GT equipped tractor at FT when operated at the same power level before any significant fuel savings occurs. On the other hand, the GT equipped tractor operating under SUTB conditions was more fuel efficient than the CVT equipped tractor as long as the GT at SUTB engine speed was less than 230 rpm to 340 rpm higher than that of the CVT equipped tractor operating at the same power level. These results show that to compensate for the parasitic losses associated with the CVT, the engine on the CVT equipped tractor must run at a significantly lower speed than that of the GT equipped tractor at FT to achieve a reduction in fuel consumption.

Table 2. Differences in engine speeds when the fuel consumption became significantly different between the two transmission types.

Speed Designation	Average travel speed ($km \cdot h^{-1}$)	Minimum difference in engine speed at which fuel consumption for CVT < for GT at FT (GT at FT – CVT) (rpm)	Maximum difference in engine speed at which fuel consumption for GT at SUTB < for CVT (GT at SUTB – CVT) (rpm)
Speed 1	5.81	440	230
Speed 2	7.88	415	330
Speed 3	10.47	395	340
Average	-	417	300

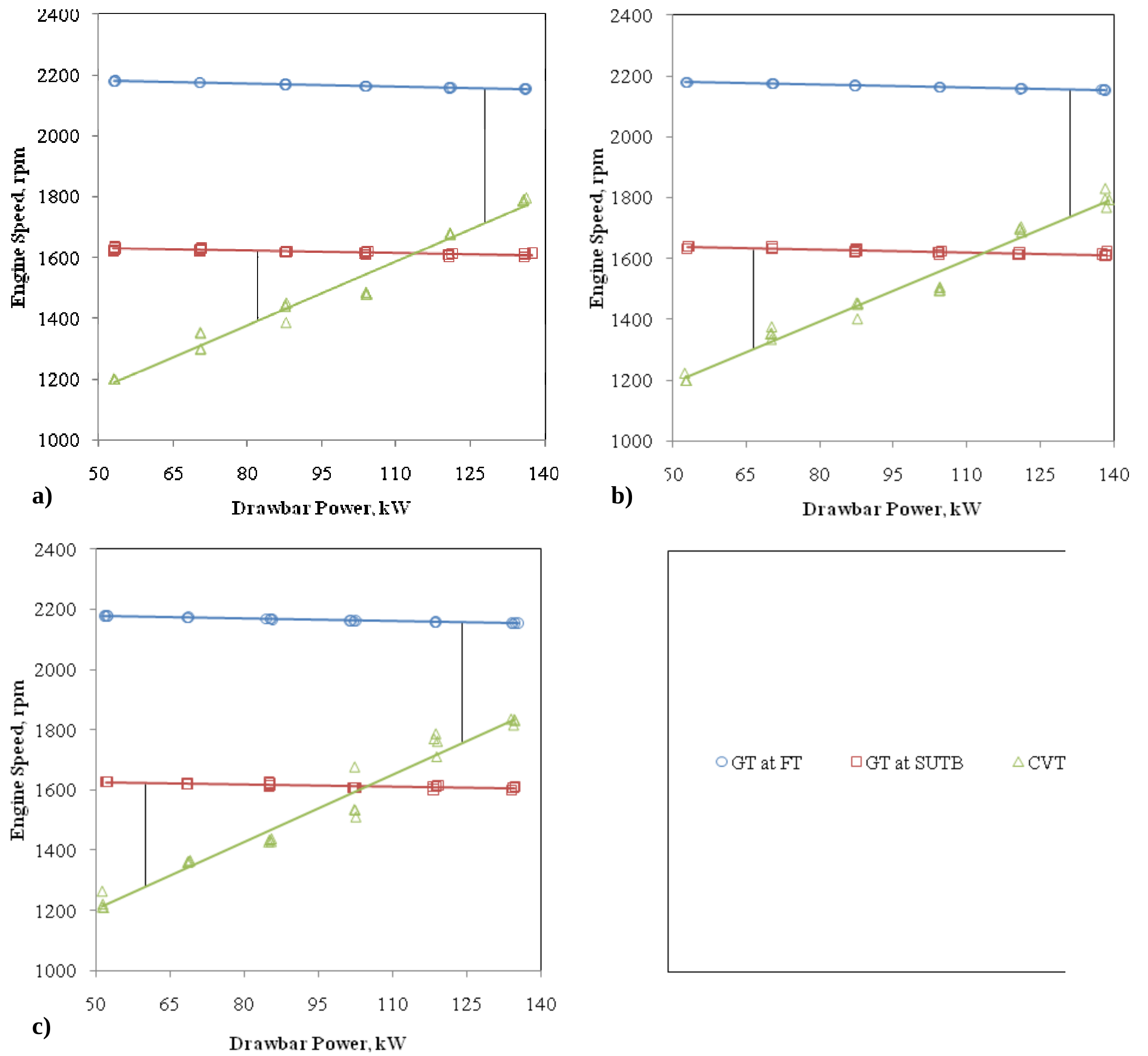

Figure 6. Engine speed as a function of drawbar power for all three transmission modes at (a) Speed 1, (b) Speed 2 and (c) Speed 3. The differences between predicted engine speeds at the points where the two transmissions produce significantly different fuel consumption values are marked with vertical lines.

With steady state loads and drawbar power levels above 35 to 50 percent of maximum depending on speed, and up to 80 percent of maximum, the GT tractor operated in SUTB mode consumed fuel at a lower rate than the CVT tractor. However, during realistic field conditions, the soil and terrain conditions often vary causing the drawbar load to vary dynamically through the field. In working to minimize fuel consumption with varying load, the operator of a GT at SUTB tractor would have to continually watch the fuel use display and adjust the throttle and gear settings to keep fuel use low while maintaining a reasonable travel speed. Some operators may prefer to minimize shifting gears with the tractor under load as they believe the shock loading that occurs with these gear shifts to be damaging to the tractor. When operating with varying load, the CVT tractor has advantages of: automatically and smoothly adjusting the engine speed and transmission speed ratio to reduce fuel consumption, and the capability to operate at any travel speed within its range.

Additional testing is needed on other models of tractors from other manufacturers to determine whether the trends found in this study pertain to all CVT equipped tractors or if they are specific to this tractor model from this manufacturer. It might also be worthwhile to test at other speeds to determine whether the trends found in this study still apply.

Conclusions

The results indicated that the CVT operated in automatic mode was more fuel efficient than the standard geared transmission operated at full engine speed when the drawbar power was less than 76% to 81% of maximum drawbar power. This was expected since the CVT automatically shifted up and throttled back to achieve the same travel speed at a lower engine speed. These results also correlated almost exactly with the results that Coffman et al. (2010) achieved when testing the John Deere 8530 IVT tractor with the transmission in automatic mode (CVT), and manual mode (GT at FT). The results also indicated, however, that the same geared transmission operated at a reduced engine speed and shifted up two gears achieved greater fuel efficiency than the CVT when the drawbar power was greater than 37% to 52% of maximum drawbar power. This makes sense, though, since there are inherently higher parasitic losses associated with a CVT than with a standard geared transmission.

The point at which the fuel consumption was found to be significantly different between transmission operating modes at each of the three forward travel speeds was also determined. Over the range of travel speed tested, (5.81 to 10.47 $km \cdot h^{-1}$), as travel speed increased, the percent of maximum power below which the CVT was significantly more fuel efficient than the GT at FT decreased. Likewise, the percent of maximum power above which the GT at SUTB was more fuel efficient than the CVT decreased as speed increased. This suggests that multiple speeds need to be tested to achieve an accurate comparison between a GT and a CVT. The minimum number required would be two that span the range of working speeds with which the tractor is used, although testing with at least three speeds would be recommended.

For each travel speed with each transmission mode, the relationship between fuel consumption and drawbar power was determined to be linear. Therefore, the minimum number of load levels that need to be tested for each travel speed in order to obtain a minimal evaluation of the linearity of the results is three loads that span the anticipated range of power levels over which the tractor is commonly used (30% to 80% in this study). Testing with more than three load levels is recommended to obtain a reasonable estimate of the linearity of the results.

Limitations to the study existed. Only one model of tractor was tested from one manufacturer, which does not give any information on how other models or tractors from other manufacturers would perform. Also, the test speeds were chosen based on the maximum loaded travel speeds in certain gears for the GT tractor.

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