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

C. N. Howard, M. F. Kocher, R. M. Hoy, E. E. Blankenship

ABSTRACT. *A John Deere 8295R IVT tractor with a continuously variable transmission (CVT) and a John Deere 8295R PowerShift (PST) tractor 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. For each travel speed with each transmission mode, fuel consumption was determined to be linearly related to drawbar power. Linear regression analyses were performed, and the 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, as travel speed increased, the percent of maximum power above which the GT at SUTB was more fuel efficient than the CVT decreased. Some significant differences existed between fuel consumption at different travel speeds within each transmission operating mode. In order to determine differences in fuel consumption between the transmission operating modes and at different travel speeds, testing with at least three loads and at least three travel speeds is recommended. Additional testing is needed on other tractor models, including models from other manufacturers, to determine whether the differences detected in this study pertain to all CVT-equipped tractors or if they are specific to this tractor model and manufacturer.*

Keywords. *Continuously variable transmission, CVT, Fuel consumption, Fuel efficiency, Geared transmission, Tractor testing.*

Testing tractors to ensure that they meet their advertised performance claims has been a central focus of the Nebraska Tractor Test Lab since the Nebraska Tractor Test Law was passed in 1919. In addition to 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.

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 types of 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 can adjust the transmission ratio and engine speed to operate at the point of maximum fuel efficiency for the given conditions, as described by Renius and Resch (2005). This approach is based on the "shift up and throttle back" (SUTB) or "gear up and throttle down" approach to driving a conventional geared transmission, as described by Grisso et al. (2011). 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. Ideally, a CVT 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 combination of gear and throttle position.

CURRENT TESTING PRACTICES

Currently, only a minimal standardized test protocol is in place that allows comparison of the fuel efficiency between tractor models that are available with both CVTs and GTs at settings other than full throttle. The Organi-

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zation for Economic Cooperation and Development (OECD) oversees the development and maintenance of worldwide tractor testing standards. Currently, the OECD Code 2 standard for official testing of agricultural and forestry tractors (OECD, 2010) is used globally 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 the fuel efficiency of these two types of tractor transmission over a range of loads would provide useful information both for the consumer looking to buy a new tractor and for the manufacturer looking to advertise the benefits of the different transmission options.

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. In addition, 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 fuel consumption of a CVT transmission operating in manual mode may not be the same as the fuel consumption 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 eight different load cycles to simulate the entire range of uses for an agricultural 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. In addition, 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% 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. The power required to pull these implements ranged from 26.4% to 81.4% of available tractor power, with an average value of 51.5%.

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. It was found that 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 ranges in the legends of tillage energy maps 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. The results reported by Chung et al. (2008) showed minimum-to-maximum soil resistance ratios of 0.57 and 0.64. Siefken et al. (2005) showed a minimum-to-maximum soil resistance value of 0.50 in fields that had previously been no-till. Likewise, 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 the ranges in the legends of soil mechanical resistance maps 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 a tractor equipped with a CVT and the same tractor model equipped with a GT operated at full throttle (FT), (2) to determine the partial load level at which statistically significant fuel consumption differences occur between the CVT-equipped tractor and the GT-equipped tractor operated under “shift up and 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), two large row-crop tractors were tested on the concrete test track of the NTTL (fig. 1), located in Lincoln, Nebraska (40° 49′ N, 96° 40′ W), at an elevation of 355 m.

The two tractors tested were the John Deere 8295R PowerShift Transmission (PST) and the John Deere 8295R Infinitely Variable Transmission (IVT) tractors. Deere uses the term PST to describe its version of a geared transmission (GT) and the term IVT to describe its version of a continuously variable transmission (CVT). The tractors were ballasted to a common ballast configuration of 75 kg per 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 4 June to 8 June in 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 then operate at full throttle. The other method is to select a gear that will give the desired travel speed 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 gear ratio to produce the desired travel speed through the field. Therefore, it was decided to compare the two tractors in three different modes of tractor operation:

1. The standard geared transmission with the engine at full throttle (GT at FT).
2. The standard geared transmission shifted up two gears (the OECD Code 2 test procedure was followed, which allowed the manufacturer to choose the number of upshifts) and with the engine throttled back (GT at SUTB) to achieve the same forward speed as GT at FT.
3. The CVT in automatic mode, i.e., the controller set to allow engine speed to vary between 1200 rpm and full throttle depending on the loading conditions (Deere, 2009), with the travel speeds set to achieve the same speeds as GT at FT.

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 to 11 km 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 the 6th, 8th, and 10th gears for the John Deere 8295R PST tractor. The maximum speeds corresponding to these gears were 5.94, 7.97, and 10.64 km 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, 80.02, 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

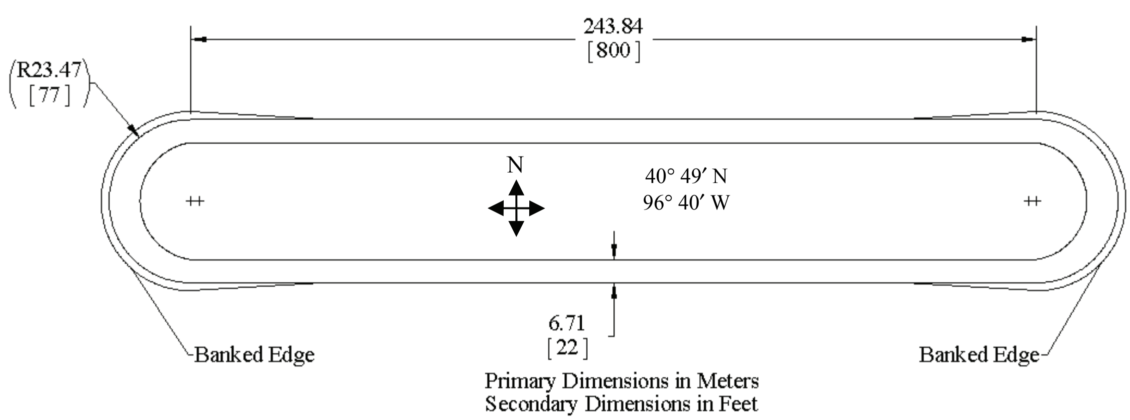

Figure 1. Test track at the Nebraska Tractor Test Laboratory, Lincoln, Nebraska.

once the hydraulic fluid 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 lbs), 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.

If comparable results were achieved on both the north and south sides 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 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 4 June 2010, the GT tractor at SUTB was tested on 5 June 2010, and the CVT tractor was tested on 8 June 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 (ver. 8.6, National Instruments Corp., Austin, Tex.). 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 that 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 flowmeter, 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 measured 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 hydraulic oil temperature. The data acquisition system operated at 1 kHz for the load, pressure, temperature, and fuel flow measurements, so the numbers of data points represented in each of those averages reported were approximately 36,900, 27,500, and 20,600 for the 5.94, 7.96, and 10.64 km·h^{-1} travel speeds, respectively. The data acquisition system operated at 1 Hz for the engine, fan, and wheel speed measurements, so the numbers of data points represented in each of those averages reported were approximately 36, 27, and 20 for the 5.94, 7.96, and 10.64 km·h^{-1} travel speeds, respectively. The data cables were properly shielded and grounded, so no hardware or software filtering of the data was necessary.

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 estimated using regression analysis. 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:

$$\begin{aligned} Q_{i,j} &= \beta_{i0} + \beta_{i1}P_{i,j} + \beta_{i2}M_1 + \beta_{i3}M_2 \\ &\quad + \beta_{i4}P_{i,j}M_1 + \beta_{i5}P_{i,j}M_2 + e_{i,j} \end{aligned} \tag{1}$$

where

$Q_{i,j}$ = measured fuel consumption at speed i for the jth

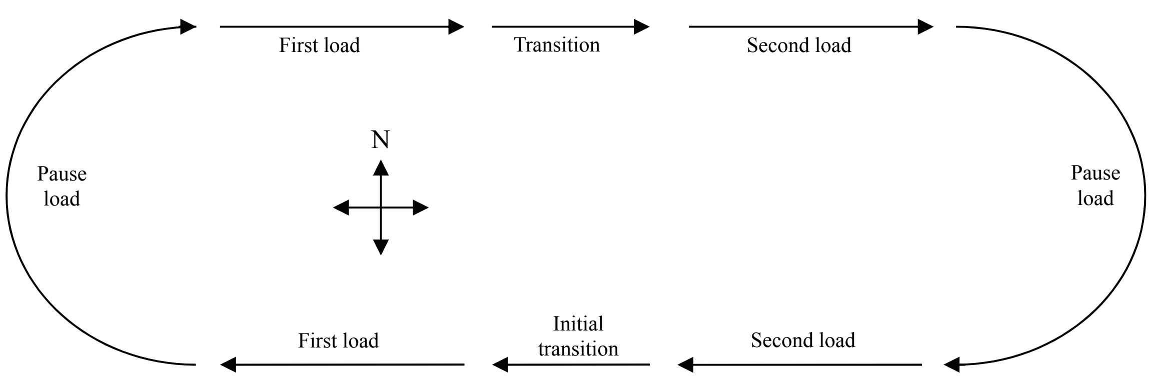

Figure 2. Drawbar load application pattern used (trend continues for the third through sixth loads in randomized sequence of each block) on the test track.

observation (kg h^{-1})

β_{i0} to β_{i5} = intercept (kg h^{-1}) and slope (kg h^{-1} kW^{-1}) terms at speed i

$P_{i,j}$ = calculated drawbar power (drawbar pull multiplied by speed) at speed i for the jth observation (kW)

M = indicator variable denoting transmission mode of operation:

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

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

$e_{i,j}$ = random error at speed i for the jth observation

i = 1, 2, and 3 (corresponding to speeds 1, 2, and 3, respectively)

j = observation number.

No differences in fuel consumption were found among the blocks, so they were dropped from the model, which was implemented using SAS PROC MIXED (SAS Institute Inc., Cary, N.C.). 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 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 mode. In this model, instead of representing transmission mode, the M values represented travel speed:

$$M_1 = \begin{cases} 1 \text{ for speed 1} \\ 0 \text{ otherwise} \end{cases}$$

$$M_2 = \begin{cases} 1 \text{ for speed 2} \\ 0 \text{ otherwise} \end{cases}$$

and i = 1, 2, and 3 (corresponding to transmission modes GT at FT, GT at SUTB, and CVT, respectively). As with 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 fuel consumption with speeds 1, 2, and 3, respectively:

$$\begin{aligned}\hat{Q}_1 &= 2.565 + 0.250P + 5.927M_1 - 0.041PM_1 \\ &+2.095M_2 - 0.031PM_2\end{aligned} \tag{2}$$

$$\begin{aligned}\hat{Q}_2 &= 4.141 + 0.239P + 5.236M_1 - 0.035PM_1 \\ &+1.051M_2 - 0.024PM_2\end{aligned} \tag{3}$$

$$\begin{aligned}\hat{Q}_3 &= 5.205 + 0.240P + 4.801M_1 - 0.034PM_1 \\ &+0.845M_2 - 0.024PM_2\end{aligned} \tag{4}$$

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

$$\hat{Q}_1 = \begin{cases} 8.49 + 0.209P \text{ for GT at FT} \\ 4.66 + 0.219P \text{ for GT at SUTB} \\ 2.56 + 0.250P \text{ for CVT} \end{cases} \tag{5}$$

$$\hat{Q}_2 = \begin{cases} 9.38 + 0.204P \text{ for GT at FT} \\ 5.19 + 0.215P \text{ for GT at SUTB} \\ 4.14 + 0.239P \text{ for CVT} \end{cases} \tag{6}$$

$$\hat{Q}_3 = \begin{cases} 10.01 + 0.206P \text{ for GT at FT} \\ 6.05 + 0.216P \text{ for GT at SUTB} \\ 5.20 + 0.240P \text{ for CVT} \end{cases} \tag{7}$$

The measured fuel consumption data and the predicted models are shown in figure 3. The fuel consumption values for the GT at FT and the 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) for these lines were found to be 0.993 for speed 1 and 0.995 for speeds 2 and 3. There were no discernible trends with respect to drawbar power in the analysis of the fuel consumption prediction errors (Howard, 2010).

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 (fig. 4). Based on the analysis of fuel consumption difference between the GT at FT and the CVT in automatic mode, shown in figures 4a, 4c, and 4e, the fuel savings of using the CVT in automatic mode increased as the power level decreased, but the fuel consumptions were 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. In this experiment, the CVT was more fuel efficient below 128 kW for speed 1, below 131 kW for speed 2, and below 124 kW for speed 3, which correspond 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.

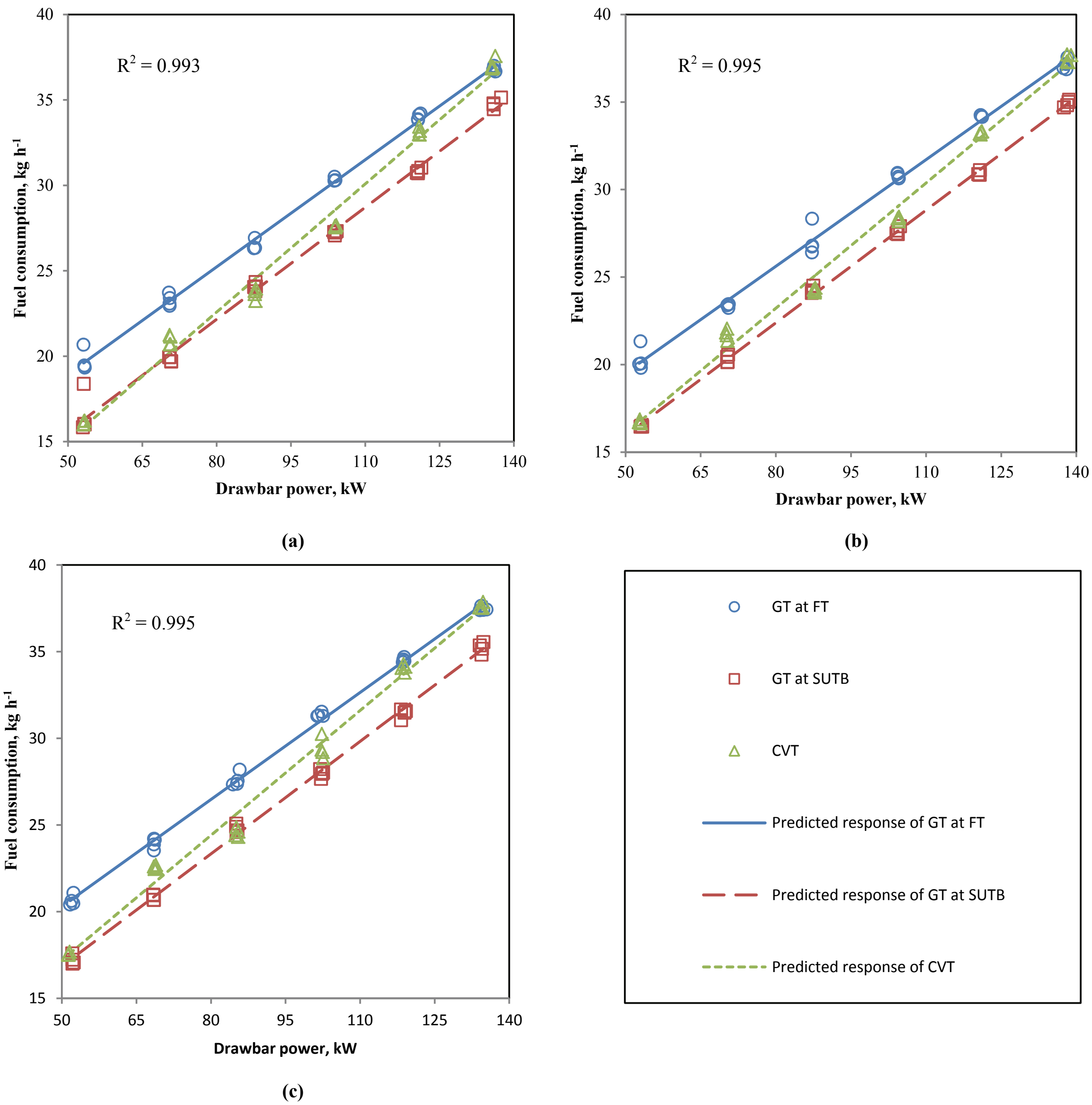

Figure 3. Hourly fuel consumption response to drawbar power for a John Deere 8295R PST and a John Deere 8295R IVT at (a) speed 1, (b) speed 2, and (c) speed 3.

The analysis of differences in 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 consumptions were similar at lower loads, as shown in figures 4b, 4d, and 4f. 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. In this experiment, the GT at SUTB was more fuel efficient above 82 kW for speed 1, above 66.5 kW for speed 2, and above 60 kW for speed 3, which correspond 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. This makes sense since there are inherently higher parasitic losses associated with a CVT than with a standard geared transmission. Note that the specific results (fuel consumption values and 95% confidence intervals) in this experiment for the GT at SUTB are likely dependent on the specific operating conditions (i.e., number of gears shifted up, and reduction in engine speed) and the experimental design (number of data points obtained), so these results should not be considered applicable to all GT at SUTB conditions.

At drawbar power levels less than about 75% of maximum, the CVT was more fuel efficient than the GT at FT. For drawbar power levels above 35% to 50%, depending on speed, the GT at SUTB in this experiment 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

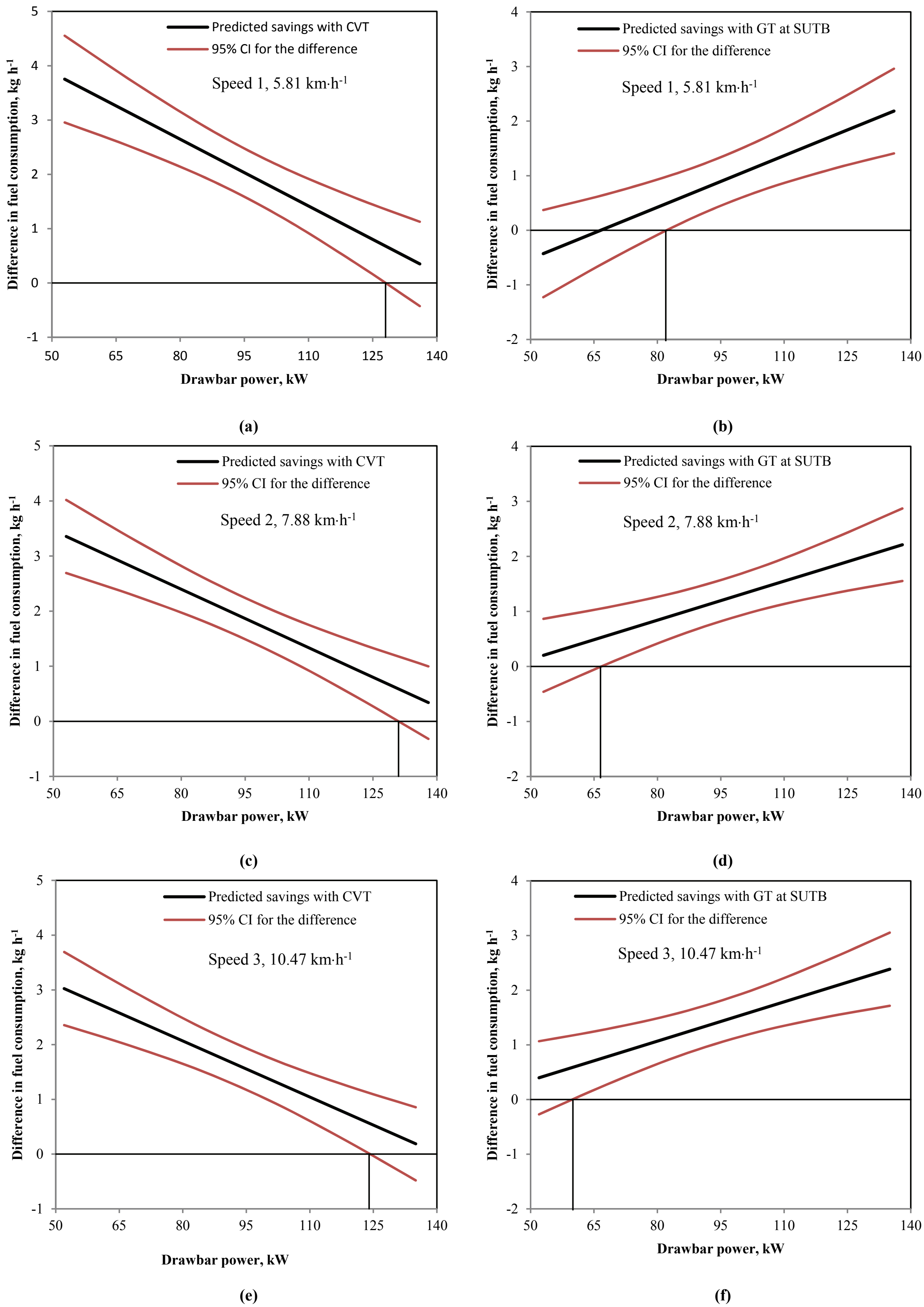

Figure 4. Difference in hourly predicted fuel consumption response to drawbar power between the GT at FT and the CVT (GT at FT – CVT) for (a) speed 1 (5.81 km·h^{-1}), (c) speed 2 (7.88 km·h^{-1}), and (e) speed 3 (10.47 km·h^{-1}) and between the CVT and the GT at SUTB (CVT – GT at SUTB) for (b) speed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

			CVT more efficient than GT at FT		GT at SUTB more efficient than CVT	
Speed Designation	Average forward travel speed from CVT fuel efficiency test ($km\ h^{-1}$)	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

SUTB in this experiment consumed fuel at the lowest rate. If the instantaneous fuel use data were available to the operator in the cab, the operator may choose to actively engage in changing gears and throttle settings to achieve the lowest fuel use rate. However, during realistic field operations, the soil and terrain conditions often vary, causing the drawbar load to vary dynamically across the field. Depending on transmission characteristics and the variability of soil and terrain conditions in the field, some operators may prefer to minimize shifting gears with the tractor under load, either for ride comfort or from a concern that the shock loading that occurs with gear shifts may damage the tractor. When operating with varying load, the CVT tractor has the advantage of automatically and smoothly adjusting the engine speed and transmission speed ratio to reduce fuel consumption.

The patterns 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 as functions of speed appeared to be approximately linear, and decreasing. 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 speeds 1 and 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 (figs. 5a, 5c, and 5e). 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.

A comparison of the values of the predicted fuel consumption difference between speeds 2 and 3 ($\hat{Q}$ for speed 3 minus $\hat{Q}$ 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 (figs. 5b, 5d, and 5f). 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 between speeds 3 and 2 was 2.59 $km\ h^{-1}$, while the average difference between speeds 2 and 1 was 2.07 $km\ h^{-1}$. 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 lines for engine speed as a function of drawbar power for the GT at FT and the GT at SUTB seemed parallel, as did the fuel consumption lines. However, there were noticeable differences in the slope of the engine speed lines 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 was 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 this experiment, the CVT-equipped tractor had significant fuel savings when the engine speed was more than approximately 400 to 450 rpm below that of the GT-equipped tractor at FT when operated at the same power level. On the other hand, the GT-equipped tractor operating under the SUTB conditions of this experiment had significant fuel savings when the GT at SUTB engine

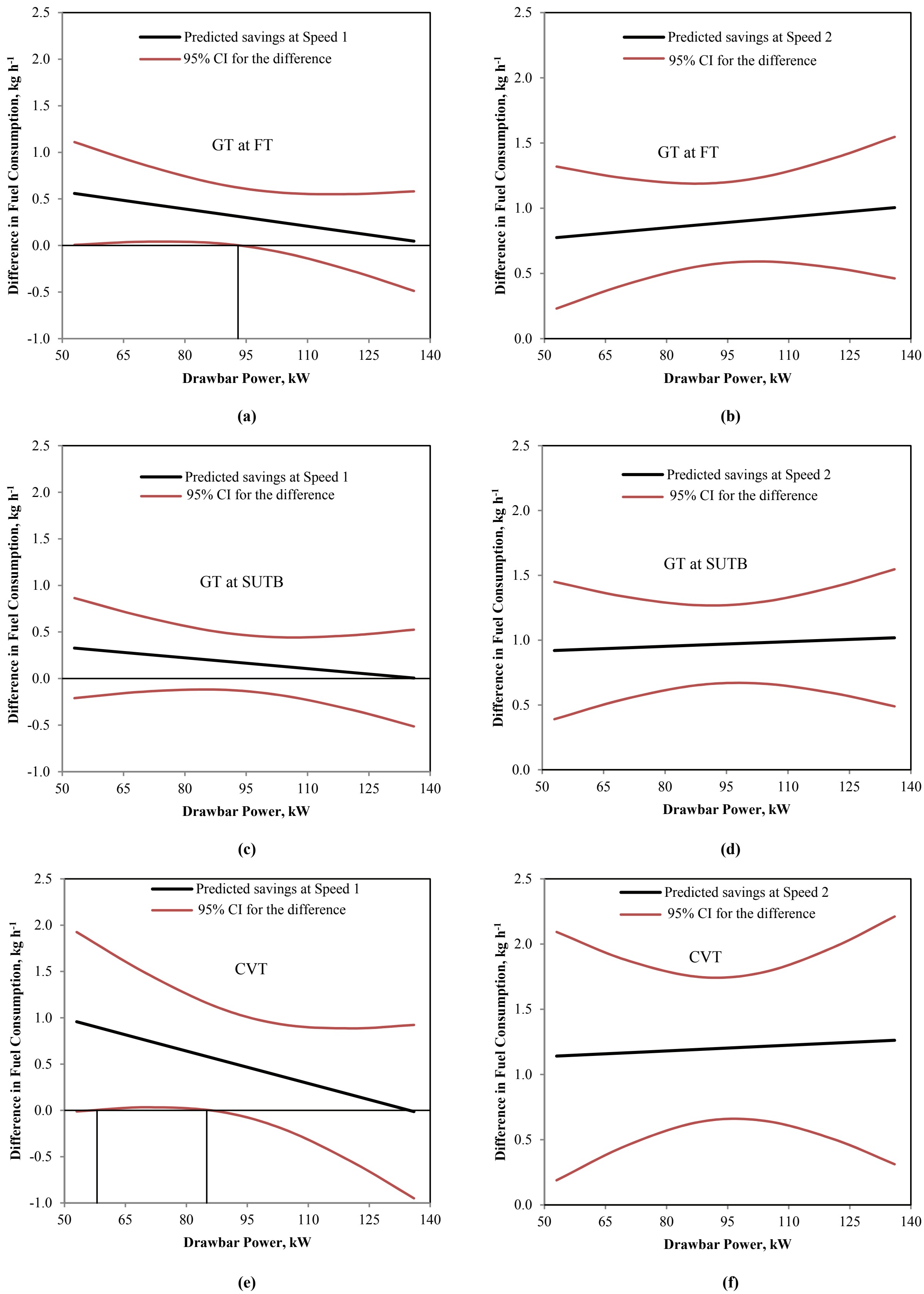

Figure 5. Difference in hourly predicted fuel consumption response to drawbar power between speeds 1 and 2 (speed 2 – speed 1) for (a) GT at FT, (c) GT at SUTB and (e) CVT and between speeds 2 and 3 (speed 3 – speed 2) for (b) GT at FT, (d) GT at SUTB, and (f) CVT.

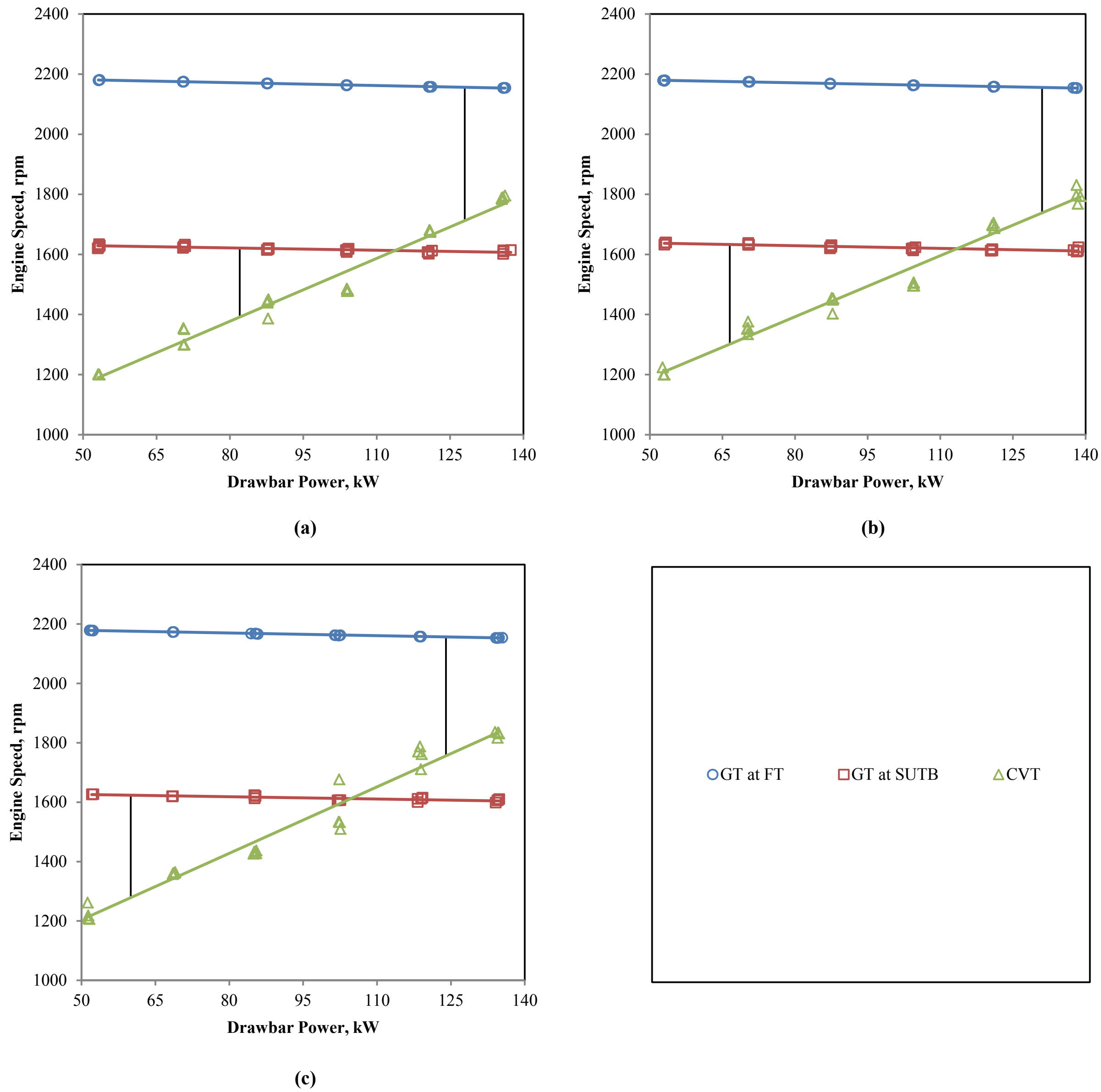

Figure 6. Engine speed as a function of drawbar power for all three transmission modes at (a) speed 1, (b) speed 2, and (c) speed 3. The differences between predicted engine speeds at the points where the two transmissions produce significantly different fuel consumption values are marked with vertical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

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

Speed Designation	Average forward travel speed (km 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

speed was less than about 230 to 340 rpm higher than the engine speed of the CVT-equipped tractor operating at the same power level. These results showed that, in this experiment, to compensate for the higher parasitic losses associated with the CVT, the engine speed of the CVT-equipped tractor had to be significantly lower than that of the GT-equipped tractor at FT to achieve a reduction in fuel consumption.

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 differences detected in this study still apply.

CONCLUSIONS

The results indicated that the CVT-equipped tractor operated in automatic mode was more fuel efficient than the standard geared transmission tractor operated at full engine speed (GT at FT) 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 were similar to results obtained by Coffman et al. (2010) when testing the John Deere 8530 IVT tractor with the transmission in automatic mode (CVT) and manual mode (simulating GT at FT). However, the results also indicated that the same geared transmission shifted up two gears and operated at a reduced engine speed (GT at SUTB) achieved greater fuel efficiency than the CVT when the drawbar power was greater than 37% to 52% of maximum drawbar power. This makes sense, 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 speeds 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. Some significant differences existed between fuel consumption at different travel speeds within each transmission operating mode. As an example, the fuel consumption at speed 2 was significantly lower than at speed 3 within each of the three transmission operating modes. 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. In addition, the test speeds were chosen based on the maximum loaded travel speeds in certain gears for the GT tractor. Also, operation of the GT at SUTB was limited to one combination of shifting the transmission up two gears and reducing the engine speed accordingly to maintain the same travel speed as the GT at FT.

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REFERENCES

Adamchuck, V. I., T. J. Ingram, K. A. Sudduth, and S. O. Chung. 2008. On-the-go mapping of soil mechanical resistance using a linear depth effect model. *Trans. ASABE* 51(6): 1885-1894.

Chung, S. O., K. A. Sudduth, C. Plouffe, and N. R. Kitchen. 2008. Soil bin and field tests of an on-the-go soil strength profile sensor. *Trans. ASABE* 5(1): 5-18.

Coffman, B. A., M. F. Kocher, V. I. Adamchuk, R. M. Hoy, and E. E. Blankenship. 2010. Testing fuel efficiency of a tractor with a continuously variable transmission. *Applied Eng. in Agric.* 26(1): 31-36.

Deere. 2009. 8225R, 8245R, 8270R, 8295R 8320R and 8345R tractor operator manuals. Moline, Ill.: Deere & Co. Available at: http://manuals.deere.com.

Degrell, O., and T. Feuerstein. 2005. DLG-PowerMix. Groß-Umstadt, Germany: DLG Test Center for Agricultural Machinery. Available at: www.dlg-test.de/powermix/. Accessed 5 February 2010.

Grisso, R., R. Pitman, J. V. Perumpral, D. Vaughan, G. T. Roberson, and R. M. Hoy. 2011. "Gear up and throttle down" to save fuel. Virginia Cooperative Extension Publication 442-450 (revised). Blacksburg, Va.: Virginia Tech. Available at: http://pubs.ext.vt.edu/442/442-450/442-450_pdf.pdf. Accessed 16 September 2011.

Howard, C. N. 2010. Testing fuel efficiency of tractors with both continuously variable and standard geared transmissions. Unpublished MS thesis. Lincoln, Neb.: University of Nebraska. Available at: http://digitalcommons.unl.edu/biosysengdiss/10/. Accessed on 17 June 2011.

McLaughlin, N. B., and S. D. Burt. 2000. Spatial mapping of tillage energy. In *Proc. 5th Intl. Conf. on Precision Agriculture*. Madison, Wisc.: ASA-CSSA-SSSA.

McLaughlin, N. B., C. F. Drury, W. D. Reynolds, X. M. Yang, Y. X. Li, T. W. Welacky, and G. Stewart. 2008. Energy inputs for conventional primary tillage implements in a clay loam soil. *Trans. ASABE* 51(4): 1153-1163.

NTTL. 2010. Nebraska OECD tractor test 1969: Summary 730, John Deere 8295R diesel, 16 speed. Lincoln, Neb.: University of Nebraska, Nebraska Tractor Test Lab. Available at: http://tractortestlab.unl.edu.

OECD. 2010. Code 2, OECD standard code for the official testing of agricultural and forestry tractor performance. Paris, France: Organization for Economic Cooperation and Development. Available at: www.oecd.org/tad/code/Code%202%20-%20Final.pdf.

Renius, K. T., and R. Reisch. 2005. Continuously variable tractor transmissions. ASAE Distinguished Lecture No. 29. St. Joseph, Mich.: ASAE.

Ricketts, C. J., and J. A. Weber. 1961. Tractor engine loading. *Agric. Eng.* 42(5): 236-239, 250, 252.

Siefken, R. J., V. I. Adamchuk, D. E. Eisenhauer, and L. L. Bashford. 2005. Mapping soil mechanical resistance with a multiple blade system. *Applied Eng. in Agric.* 21(1): 15-23.