ENERGY USE AND PRODUCTIVITY OF AN ACTIVE PASSIVE-TILLAGE MACHINE

Kevin J. Shinners, Associate Professor
Agricultural Engineering Department, University of Wisconsin, Madison, WI, USA

J. Malcolm Wilkes, Lecturer
Wye College, University of London, Wye, Ashford, Kent, UK

Todd D. England, Development Engineer
Oshkosh Truck Corporation, Oshkosh, WI, USA

ABSTRACT

An experimental tillage tool that combines active, rotary powered tillage elements with conventional passive chisel shanks was field tested. Negative draft created by the active elements was used to contribute to the passive shank draft requirements. Configuring a tillage machine in this manner could improve power transmission efficiency for tillage, reduce wheel slip, allow use of lighter tractors and allow tillage to take place in more difficult tractive conditions.

Field tests of the experimental active-passive tillage machine were conducted to investigate the effects of velocity ratio, ground speed and active:passive depth ratio on machine performance. The active-passive tillage machine was able to reduce the draft by at least 50% in all cases compared to a similarly configured purely passive tillage tool. The active-passive tillage machine reduced the gross specific fuel consumption by a maximum of 27% and net fuel consumption by a maximum of 36%. Wheel slip was reduced by at least 50% with the active-passive tillage machine. The active-passive tillage machine was able to reduce the draft, fuel consumption, wheel slip and engine flywheel power requirements compared to a similarly configured passive tillage tool. The velocity ratio was the most influential in affecting the operating characteristics.
INTRODUCTION

Approximately 20 percent of energy for production agriculture is used for field operations, with a majority of this energy applied toward tillage operations (Stout et al., 1984). Nearly all tillage tools utilize passive tillage elements. A draft force is applied to the tool, which causes the elements to move through the soil. The power required is developed by the tractor engine and is transmitted through the soil-tire interface and the tractor drawbar. Because of the poor efficiency of power transmission at the soil-tire interface, tillage energy efficiencies are low. Also, because tractors require considerable weight to provide necessary traction, soil compaction may occur and increased power is required to overcome the wheel slip and rolling resistance of the tractor tires.

One way to bypass the inefficient soil-tire interface, is through active or rotary powered tillage elements. These active elements are usually powered by the tractor pto drive. Tillage tools which utilize active elements transmit power directly to the soil rather than having the tool drawn through the soil. These machines generally till a greater volume of soil than is required in most field crop systems and therefore require considerable power per unit width. The forward thrust which is generated by the active elements can be harmful to the drive train of the tractor (Wismer et al., 1968). Purely active tillage machines return power to the tractor's drawbar by pushing the tractor. This results in an overload of the pto drive line and a reverse torque on the drive axle transmission. Additionally, because of the geometry of the line of action of the negative draft, weight transfer takes place from the rear to the front axle. This increases the rolling resistance requirements of the tractor.

A way to control this detrimental forward thrust is to combine active and passive elements, such that the rotary powered elements produce a forward thrust towards the draft requirements of the passive elements. The potential benefits of combining active and passive tillage elements are: (1) power for tilling the soil can be transmitted to the tillage elements through a mechanical power train more efficiently than through the soil-tire interface, (2) the negative draft of the active elements can be used to provide some or all of the draft of the passive elements and contribute to the rolling resistance of the tractor.
tractor, (3) reduced draft of tillage machines will result in less wheel slip, improving field productivity, 
(4) reduced draft of tillage machines will allow use of lighter tractors, reducing soil compaction and 
possibly reducing tractor cost, and (5) reduced draft of tillage machines will allow operations to be 
performed in more difficult traction conditions.

A drawback to tillage tools with both active and passive elements is that the tillage tools 
becomes more complex. Tillage tools will go from a machine with virtually no moving parts to a 
machine with gear boxes, chains, bearing, shafts, etc. With greater complexity of parts comes higher 
machine costs and lower reliability. However, the combination tillage tool will not have drive 
components different from those successfully maintained on many other agricultural machines. 
Additionally, purely active tillage machines with similar drive train components have been successfully 
marketed for years.
PREVIOUS RESEARCH

There have been a number of research projects which utilized the concept of combining active and passive elements on one tillage machine. Charmen et al. (1979) built and tested a high output combination tillage tool especially designed for working heavy soils. This tillage tool consisted of a 2.43 m wide by 38 cm diameter rotor. Bolted to this rotor were L-shaped blades used to cut the soil. To balance the forward thrust from the rotating blades, deep chisel tines were mounted behind the rotary unit. The tool drew 80-100% of its power from the pto. Net specific energy requirements for the machine were 50% less than a moldboard plow operating at the same depth as the passive elements.

Wilkes and Addai (1988) built and tested a combined tillage tool known as the "Wye Double Digger". The digger consisted of a rotary subsoiler which loosened the subsoil in the open furrow and a moldboard bottom, which was located behind and offset by one furrow width. The moldboard bottom turned the next slice of soil onto the loosened subsoil. The active elements performed as both cultivators and tractive devices, producing a forward thrust which provided at least some of the draft for the moldboard bottom. When compared to a moldboard plow operating at the same depth, the "Wye Double Digger" was found to reduce the draft power, wheel slip and specific energy under some experimental conditions.

Hendrick and Gill in the early 1970's wrote a series of papers quantifying design parameters for rotary tillers. Hendrick and Gill (1971a,b) determined that low rotary speed and forward rotor rotation were necessary to reduce the draft and power requirements of rotary tillage tools. However, a rotary speed that was too slow causes excessive variation in tillage depth because of blade kinematics (Hendrick and Gill, 1971c). A fast rotary speed disturbed more soil than necessary and required excess power (Hendrick and Gill, 1971b).

Hendrick (1980) suggested a practical lower limit of the ratio of rotor tip speed to forward speed, i.e. velocity ratio, to be 2.5:1. Hendrick (1980) used this lower limit velocity ratio to design the shape of the soil engaging tools of the machine used in the research reported in this paper. At velocity
ratios below 2.5:1, the back side of the tillage elements of each rotor would contact uncut soil.

Hendrick (1980) built and tested a single rotor powered active tillage machine. He compared the power requirements of a similar passive chisel shank with that of the powered rotor operating at the same depth in three different soil types. The powered rotor disturbed a larger volume of soil and broke the soil more finely than did a passive chisel shank. It was also noticed that the forward rotating powered rotor provided its own thrust through the soils and returned a proportion of its power demand to the system as negative thrust. If the negative draft force was utilized, the specific energy requirements for the powered rotor would have been 51% less than that for the passive chisel.

Araya and Wu (1987) built a single rotor active tillage machine similar to Hendrick (1980). The goal of the project was to develop a powered rotary subsoiler which would inject sewage sludge into the soil. It was determined that the power requirements for the system was reduced by as much as 50% by using rotary elements and by injecting liquid sludge under pressure into the soil during tillage operations.

Shinners et al. (1990) performed preliminary field testing of an active-passive tillage machine. Early tillage machines which combined active and passive tillage elements contained active elements which tilled the whole width of the machine. The passive elements of a few machines were not strategically positioned because they were usually located in the tilled soil after the active elements. The active-passive tillage machine used by Shinners et al. (1990), had active elements that did not till the whole width of the machine but rather only tilled narrow zones very similar to a conventional chisel plow. The passive elements on this machine were positioned to till the soil left untilled by the active elements. With this machine, soil was not completely inverted and surface residue was left.

Four experiments were conducted dealing with depth of tillage, bite length, and active:passive depth ratio. The combination tillage machine with 2 active - 2 passive elements required 87% less draft and draft power than a tool with 4 passive elements. Based on assumed power transmission efficiencies, the combination tillage tool was predicted to be 34% more energy efficient than a similar passive tillage
tool. Wheel slip was 57% less for a 2 active - 2 passive element tool compared to a 4 passive element configuration.
MACHINE DESCRIPTION

The experimental machine (Figures 1 and 2) used in this research was designed and built at the National Soil Dynamics Laboratory, USDA-ARS in Auburn, AL. However, it was extensively redesigned at the Agricultural Engineering Department of the University of Wisconsin. The combination tillage machine was semi-mounted and utilized a castered trailing wheel to assist in raising and lowering the rear of the machine. The active elements consisted of two rotors which were approximately 1.5 m diameter. Six tillage elements of 2.5 cm width were rigidly attached to each rotor. To offset the peak torque loads, the elements on the second rotor had an angular offset of 30° from those on the first rotor. The rotors were powered by the tractor's pto shaft through a torque-sensing variable speed belt drive, a right angle gear box and finally a double chain speed reduction drive. With this drive system the rotor speed could be readily adjusted between 22 and 103 rpm, thus minimizing the downtime needed for speed adjustments during field tests.

Two passive elements were mounted toward the rear of the machine between the two active rotors. Two other passive elements could be mounted at the front of the machine outboard from the active elements. The mounting of the passive elements was designed such that changing depth and removal of the shanks could be easily accomplished without tools. This allowed the active and passive tillage elements to be operated at different relative depths if so desired. Horizontal spacing between all tillage elements was 38 cm. The passive elements were standard spring-cushioned chisel shanks with straight shovel soil engaging tools with a 5 cm width. The soil engaging portion of the active elements were 2.5 cm wide. The six tillage elements on each rotor were designed such that they would not contact uncut soil at velocity ratios greater than 2.5:1. In order to prevent residue from accumulating on either the active or passive elements, plain coulters were located in front of all tillage elements.
QUANTIFYING MACHINE PERFORMANCE

To evaluate the operating performance of the active-passive tillage machine the following parameters were recorded during field tests:

• Tillage depth of active and passive elements.
• Machine configuration relative to combination vs purely passive.
• Rotational speed of active elements.
• True ground speed, axle speed and fuel use of tractor.
• Relative depth between active and passive tillage elements.
  • Pto torque and speed.
  • Draft force.

In order to describe the operational characteristics of the active elements, an parameter known as the velocity ratio was used. Hendrick et al. (1980) described the velocity ratio as:

\[
\lambda = \frac{V_p}{V_f} : 1
\]  

[1]

The velocity ratio controls the operating performance of the active-passive tillage machine in terms of draft reduction and tillage effectiveness (Hendrick et al., 1980). The velocity ratio is also related to the bite length of the active elements. The bite length is the length of soil segment tilled by each individual tillage element on a rotor:

\[
BL = \frac{(V_f \times 10^{6})}{(\omega \times N \times 60)}
\]  

[2]

The active:passive depth ratio was used to relate the relative tillage depth between the active elements and passive elements. The active elements were kept at a constant tillage depth of approximately 30 cm during all tests. The passive shanks were raised or lowered to change the depth ratio. This is one method that can be used to control the amount of draft required by the tractor. The active:passive depth ratio was defined as:

\[
AP = \frac{D_a}{D_p} : 1
\]  

[3]
A Massey-Ferguson Model 2675 tractor and the tillage machine were equipped with instruments to measure draft, pto torque, pto rotational speed, rotor rotational speed, true ground speed, tractor rear axle speed, and fuel flow. All data from the transducers to be described below was conditioned and collected with a portable, programmable Campbell Scientific 21X datalogger. The datalogger was capable of excitation of strain gauge transducers, measuring voltage signals from transducers and also measuring pulse signals.

Thompson and Shinners (1989) described the portable three-point hitch dynamometer which was used to measure draft. The dynamometer was attached to the three-point hitch of the tractor. Load cells were located at each of three hitch points which measured the vertical and horizontal reaction forces of the implement.

The tractor was also equipped with a Magnavox Model RSS 101 radar speed sensor to measure the true ground speed. Rear wheel speed was measured by BEI Model M15 rotary encoders driven from axle mounted sprockets through a chain drive. Gross fuel flow was measured by a Pierburg Luftfahrtgerate Union Model PLU116H fuel flowmeter which was incorporated into the fuel system just before the tractor injector pump. In addition to the instrumentation on the tractor, the active-passive tillage machine was equipped with transducers to measure rotor speed, pto torque and speed. A Himmelstein MCRT Model 6-02T torque transducer was placed at the pto shaft of tractor and measured pto torque and speed. Rotor speed was measured at the main gearbox input shaft with a magnetic pickup.

All the data from the instruments were recorded at a frequency of 5 Hz, except for the pto torque and speed which was recorded at 10 Hz. After each test the datalogger would automatically download the average values for each transducer into a Campbell Scientific Storage Module SM192. This data was then downloaded to a microcomputer, converted into an ASCII file and manipulated as required. From the recorded parameters, machine performance variables such as total draft, specific fuel use, wheel slip, pto power and draft power were calculated.
The power to till the soil comes strictly from the tractor drawbar with a passive tillage tool and from a combination of the tractor's pto and drawbar with the active-passive tillage machine. When comparing the required power for tillage between the two tillage machines, simply comparing the drawbar power to the sum of the pto and drawbar power is incorrect because of very different transmission efficiencies between pto and drawbar power. Therefore, it was necessary to estimate the required engine flywheel power for the two tillage machines. This was accomplished by estimating the power transmission efficiencies for the tractor's transmission and tires:

\[
P_{fly\ a/p} = \frac{P_{pto}}{\eta_{tn}} + \frac{P_{db} + P_{rr}}{(\eta_{tr} \times \eta_{tn})}
\]

\[
P_{fly\ p} = \frac{P_{db} + P_{rr}}{(\eta_{tr} \times \eta_{tn})}
\]

The tractor transmission efficiency was assumed to be 0.885. This represents an average value taken from ASAE Engineering Practice EP 496 (ASAE, 1991). Two other variables needed to be estimated. One was the rolling resistance power and the other was the tractive efficiency. Wismer and Luth (1974) estimated the latter parameter through empirical modeling:

\[
\eta_{tr} = \left(1 - \frac{(\mu_r + (R_f / R_d)(\mu_f))}{\mu_t}\right)(1 - S)
\]

\[
\mu_r = (1.2/CN_r + 0.04)
\]

\[
\mu_f = (1.2/CN_f + 0.04)
\]

\[
\mu_t = 0.75(1 - e^{-0.3S}(CN_r))
\]

Wismer and Luth (1974) developed a term called wheel numeric which takes into account the geometry of the tires and also soil strength:

\[
CN_r = \frac{(CI \times b \times d \times 10^6)}{(0.5 \times W_{\text{dr}})}
\]

\[
CN_f = \frac{(CI \times b \times d \times 10^6)}{(0.5 \times W_{\text{df}})}
\]

It was assumed that for a semimounted implement, the weight transfer to the rear wheel and from the front wheel was 0.45 and 0.29 of the measured draft, respectively (Hunt, 1986). Appendix B contains the measured values of the front and rear axle loads of the MF 2675 tractor when the rear
wheels were ballasted and unballasted. Also, provided is a table of the measured diameter and width of the both the front and rear tires.

Wismer and Luth (1974) also estimated the rolling resistance power:

$$P_{rr} = \left[ (R_f \times \mu_f) + (R_r \times \mu_r) \right] \times (V_i) \times (10^3 / 3600) \quad [11]$$

Wheel slip was given by Bekker (1956) which quantifies the efficiencies of the driven tires:

$$S = 1 - \left( \frac{V_f}{V_t} \right) \quad [13]$$

Gross specific fuel consumption provides a measure of the efficiency with which fuel is being converted into useful work:

$$SFC_{gross} = \frac{(FU_{gross} \times 10)}{(V_t \times W)} \quad [14]$$

Net specific fuel consumption is similar to gross specific fuel consumption but the fuel use due to the rolling resistance of the tires is subtracted away from the gross fuel use of the tractor. By doing this the fuel use of only the tillage tool or machine is estimated:

$$SFC_{net} = \frac{(FU_{net} \times 10)}{(V_t \times W)} \quad [15]$$

Specific fuel use defines a ratio of the amount of power that is produced from a set flow rate of fuel. To calculate specific fuel use, a total power term was used which was calculated by adding together the power at the pto and drawbar:

$$SFU = \frac{P_{tot}}{FU} \quad [15]$$

During each field test two soil parameters were recorded. One parameter was soil moisture, expressed in terms of dry basis. The average moisture content for the top 30 cm of soil was determined by taking a sample with a soil probe and determining sample weight before and after oven drying at 103°C for 72 hours.

The second soil property measured was the cone index. The cone index was determined by using a hand operated penetrometer. The penetrometer was inserted into the ground to a depth of 30 cm. The penetrometer was able to record and store the average penetration force in kgf in 3 cm intervals for the first 33 cm. The penetrometer tests were conducted in accordance with ASAE.
DESCRIPTION OF FIELD TESTS

Preliminary field tests were conducted during the summer and fall of 1990. Based on the knowledge gained, extensive field tests were then conducted during the summer 1991.

Results from the preliminary field tests indicated that velocity ratio, active:passive depth ratio and ground speed were the most important variables in terms of reducing draft and fuel use. Therefore, the tests conducted in 1991 concentrated on the effects of these three variables on machine performance. Three ground speeds were used; 4.8, 6.4 and 8.0 km/h. Four velocity ratios were used; 2.5, 2.2, 1.9 and 1.5:1. Three active:passive depth ratios were used; 1.6, 1.3 and 1:1. The three active:passive depth ratios were obtained by keeping the active elements at a constant tillage depth of 30 cm and then raising the passive chisels 8 and 11 cm. The performance of the active-passive tillage machine was compared to that of a purely passive tillage tool. The passive tillage tool used for comparison purposes was the experimental machine with the rotors removed and four passive elements in place. In order to obtain a fair comparison, the passive tool was configured with two shanks tilling at 30 cm depth and the other two shanks operating at 19, 22 or 30 cm to correspond to the active:passive depth ratios of 1.6, 1.3 or 1:1. For all configurations, the draft requirements included that required by the four residue cutting coulters.

All field experiments were conducted at the University of Wisconsin Arlington Experimental Farm on Plano silt loam soils with 0 to 6% slope. Previous crops grown on the test fields included corn, oats and snap beans. Field experiments were conducted on May 20 and 29, June 26, and August 5 and 19, 1991. On these five days each experimental condition was replicated at least twice and more usually four times per day. The length of each replicate ranged from 88 to 152 m. In most cases, the tractor was unballasted during testing of the active-passive tillage machine. Approximately 9 kN of liquid ballast was added to the rear tires during tests of the purely passive machine.

On August 16, 1991 an experiment was conducted to determine the rolling resistance fuel use for the MF 2675 tractor. The goal was to develop expressions for specific fuel use as a function of
ground speed and vertical three-point hitch load. Five three-point hitch vertical loads were used; 0.0, 2.7, 4.5, 6.2, and 8.0 kN. For each vertical load, the tractor was operated at idle and at 4.8, 6.4, 8.0 km/h. The above experimental conditions were repeated with tractor both unballasted and ballasted with about 9 kN liquid ballast. The average fuel use values were recorded for each test.

Statistical analysis was carried out on the draft, gross fuel consumption and wheel slip data to quantify differences between experimental conditions. These three parameters were chosen because it was felt that they represented the most important parameters and other parameters could be derived from these three. Two-way analysis of variance was used where the day effects were removed by blocking. The least square difference (LSD) reported in the results indicate no statistical difference at a probability of 5%.
RESULTS

The data from the five test days was grouped according to similar active:passive depth and velocity ratios (Tables 1 - 3). Each value in these tables represents the average of from 14 to 15 total replications for tests conducted on May 20 and 29, June 26, and August 5 and 19, 1991. By averaging the data, the general trends of the active-passive tillage machine were determined. Table 4 compares the active-passive tillage machine to the purely passive tillage tool in terms of percent reduction of ten operating parameters.

In all experimental conditions the active-passive tillage machine reduced the draft of the passive machine by at least 55%. Often the reduction was greater than 100%, which meant the active elements were pushing the tractor by creating a negative draft larger than that needed by the passive elements. Generally, higher negative draft occurred at greater velocity and active:passive depth ratios.

The active-passive tillage machine reduced net fuel consumption for tillage by 36% at 8.0 km/h, an active:passive depth ratio of 1:1 and a velocity ratio of 1.5:1. This represented the maximum percent reduction. In general the largest percent reduction in gross specific fuel consumption occurred at 8.0 km/h and lower reductions occurred at the slower ground speeds. Lower velocity ratios produced greater reductions in all fuel use parameters.

At the slower ground speeds and higher velocity ratios, gross specific fuel consumption was greater for the active-passive tillage machine than the purely passive tool. In order to achieve a reduced gross specific fuel consumption, the active-passive tillage machine should be operated at velocity ratios below 2.0:1. At velocity ratios above 2.0:1, excessive energy was probably consumed by throwing and pulverizing the soil.

The power transmission efficiency of the active-passive tillage machine can be represented in the specific fuel use parameter. Most test combinations produced a negative percent reduction in terms of specific fuel use. In other words, the tractor engine produced on the average more power per unit of fuel flow when operating the active-passive tillage machine than when operating the purely passive tillage
tool. This can be attributed to the more efficient mechanical drive system used to transfer power for the former machine.

With net fuel use, the rolling resistance of the tractor was removed of the analysis. The net fuel use parameter helps describes the fuel use if the tractor and tillage tool combination were more closely matched in terms of size and weight. The MF 2675 tractor was larger in weight than required to produce the draft power for four passive chisel shanks. With no ballast in the rear tires and pulling 4 passive shanks, the maximum wheel slip was approximately 13%. When the tractor was pulling the four shank passive tool, wheel slip was in the 5-10% range. The low wheel slip was caused by the large unballasted weight of the tractor. Because of the excessive weight, a relatively large percentage of the gross fuel use went into rolling resistance. To see the full benefits of the active-passive tillage machine, a better match between tillage machine size and tractor weight should be used in future experiments.

The reduction in engine flywheel power quantifies the difference in engine power needed to operate the active-passive tillage machine as compared to the passive tillage tool. Engine flywheel power reductions were found at the lower velocity ratio range (1.5 and 1.9:1). At 8.0 km/h, an active:passive depth ratio of 1:1 and velocity ratio of 1.5:1, the engine flywheel power was reduced by 23% compared to the passive tool requirements. As the velocity ratio increased above 2.0:1, the engine flywheel power for the active-passive tillage machine was greater than that required by the passive tool. The largest difference occurred at 8.0 km/h, an active:passive depth ratio of 1.6:1 and a velocity ratio of 2.2:1. At this configuration, the active-passive tillage machine required 26% more engine flywheel power. Even though engine flywheel power increased, gross specific fuel consumption was still reduced in this situation. This again was related to the higher power transmission efficiency when operating the active-passive tillage machine.

In all configurations, wheel slip was reduced by at least 50%. The majority of the time reduction was in the 70-80% range. Because of the low wheel slips which developed with the active-passive tillage machine, it appears lighter tractors could be used to operate the active-passive tillage machine, which
could lead to lower soil compaction and increased field productivity.

Through linear regression, draft and gross specific fuel consumption for the active-passive tillage tool were modeled. To model the gross specific fuel consumption data a multiple regression equation was necessary. Besides velocity ratio, the gross specific fuel consumption was influenced by soil moisture, cone penetrometer and tractor weight. The last three variables were necessary to account for the day to day variations. Below are the linear regression equations for each ground speed. Table 5 contains the moisture content and cone penetrometer readings for the five experiments conducted in 1991.

<table>
<thead>
<tr>
<th>Depth Ratio</th>
<th>Ground Speed -- 8.0 km/h</th>
<th>Ground Speed -- 6.4 km/h</th>
<th>Ground Speed -- 4.8 km/h</th>
</tr>
</thead>
<tbody>
<tr>
<td>1:1</td>
<td>SFC\text{\textsubscript{gross}} = -211.3 + 8.98(\lambda) + 197.3(MC) - 0.0135(CI) + 3.15(WT) \quad R^2 = 87.4 %</td>
<td>SFC\text{\textsubscript{gross}} = -108.5 + 4.68(\lambda) + 102.9(MC) - 0.0081(CI) + 1.68(WT) \quad R^2 = 72.2 %</td>
<td>SFC\text{\textsubscript{gross}} = -121.6 + 3.93(\lambda) + 138.4(MC) - 0.0068(CI) + 1.68(WT) \quad R^2 = 88.7 %</td>
</tr>
<tr>
<td>1.3:1</td>
<td>SFC\text{\textsubscript{gross}} = -66.2 + 4.86(\lambda) + 47.0(MC) - 0.0135(CI) + 10.51(WT) \quad R^2 = 95.5 %</td>
<td>SFC\text{\textsubscript{gross}} = -91.7 + 2.99(\lambda) + 88.3(MC) - 0.0068(CI) + 1.47(WT) \quad R^2 = 70.8 %</td>
<td></td>
</tr>
<tr>
<td>1.6:1</td>
<td>SFC\text{\textsubscript{gross}} = -22.7 + 4.30(\lambda) + 5.7(MC) - 0.0047(CI) + 0.63(WT) \quad R^2 = 92.1 %</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
1:1 \[ SFC_{\text{gross}} = -182.3 + 3.83(\lambda) + 190.7(MC) - 0.0110(CI) + 2.73(WT) \quad R^2 = 92.5\% . \]

1.3:1 \[ SFC_{\text{gross}} = -163.6 + 2.89(\lambda) + 190.7(MC) - 0.0081(CI) + 2.31(WT) \quad R^2 = 94.6\% . \]

1.6:1 \[ SFC_{\text{gross}} = -115.9 + 2.24(\lambda) + 136.5(MC) - 0.0068(CI) + 1.68(WT) \quad R^2 = 89.4\% . \]

<table>
<thead>
<tr>
<th>Date</th>
<th>Moisture Content (dry basis) (decimal)</th>
<th>Average Cone Penetrometer (kPa)</th>
<th>Tractor Weight (kN)</th>
</tr>
</thead>
<tbody>
<tr>
<td>May 20</td>
<td>0.24</td>
<td>1228</td>
<td>60.7</td>
</tr>
<tr>
<td>May 29</td>
<td>0.26</td>
<td>1552</td>
<td>60.7</td>
</tr>
<tr>
<td>June 26</td>
<td>0.18</td>
<td>2413</td>
<td>69.5</td>
</tr>
<tr>
<td>August 5</td>
<td>0.17</td>
<td>2334</td>
<td>69.5</td>
</tr>
<tr>
<td>August 19</td>
<td>0.24</td>
<td>1505</td>
<td>60.7</td>
</tr>
</tbody>
</table>

**Table 6** -- Test conditions for May 20 and 29, June 26 and August 5,19, 1991.

A similar analysis was performed on the draft of the active-passive tillage machine. To model the data a multiple regression model was used. Draft was influenced by velocity ratio and soil properties.

**Depth Ratio**

**Ground Speed -- 8.0 km/h**

\[ P = 15.92 - 10.29(\lambda) + 21.28(MC) + 6.45 \times 10^{-6}(CI) \quad R^2 = 81.6\% . \]

1.3:1 \[ P = 3.04 - 7.92(\lambda) + 39.47(MC) + 1.60 \times 10^{-3}(CI) \quad R^2 = 82.9\% . \]

1.6:1 \[ P = 5.15 - 7.75(\lambda) + 29.63(MC) + 1.07 \times 10^{-3}(CI) \quad R^2 = 78.4\% . \]

**Ground Speed -- 6.4 km/h**

\[ 1:1 \quad P = 4.49 - 7.41(\lambda) + 35.06(MC) + 1.85 \times 10^{-3}(CI) \quad R^2 = 71.1\% . \]
1.3:1 \[ P = 5.89 - 4.93(\lambda) + 15.43(MC) - 1.74 \times 10^{-3}(CI) \] \[ R^2 = 54.9\% \]

1.6:1 \[ P = 8.97 - 5.62(\lambda) + 0.71(MC) + 1.29 \times 10^{-4}(CI) \] \[ R^2 = 65.9\% \]

Ground Speed -- 4.8 km/h

1:1 \[ P = 2.69 - 2.44(\lambda) + 22.29(MC) + 1.63 \times 10^{-3}(CI) \] \[ R^2 = 46.9\% \]

1.3:1 \[ P = 5.85 - 1.17(\lambda) - 10.03(MC) - 1.67 \times 10^{-3}(CI) \] \[ R^2 = 33.7\% \]

1.6:1 \[ P = 17.79 - 2.33(\lambda) - 49.03(MC) - 2.00 \times 10^{-3}(CI) \] \[ R^2 = 78.9\% \]

The specific fuel use regression equations suggest that the velocity ratio has less effect on fuel use at greater active:passive depth ratios and greater ground speeds. The draft regression equations suggest that higher velocity ratios result in larger negative draft. It is believed that this trend was due to the rotor tillage element design. These tillage elements were designed such that the back side would not contact uncut soil at velocity ratios greater than 2.5:1 (Hendrick, 1980). In this research however, velocity ratios well below 2.5:1 were required to obtain favorable fuel use results. When the back side of the blades contacted uncut soil, less negative draft was produced because this interference caused a reverse torque to be applied to the rotor.

For all regression equations above, the residual plots were random (England, 1991). Therefore, the gross specific fuel consumption or draft values that were predicted from the models were approximately independent and the models were considered valid because of the scatter in these residual plots. Because these models contain multiple predictor variables, they tend to model the trends in this data very closely. If these models are used to predict gross specific fuel consumption or draft for different soil conditions other than these, the developed equations may prove to be an inappropriate model for predicting specific fuel consumption or draft.
DISCUSSION

Today's tillage tools are becoming larger and wider and operating at faster speeds. These faster operating speeds can lead to higher draft requirements. The active-passive tillage machine has shown greatest benefits when operating at faster speeds and higher passive element draft requirements. The greatest benefits in terms of draft and gross specific fuel consumption reductions occurred at the higher ground speeds tested. At the faster speeds and slower velocity ratios, the active-passive tillage machine performs equal to the passive system in terms of tillage effectiveness by observation. It was very difficult to visually differentiate the soil tilled by the passive or active elements. The tilled soil appeared similar in terms of surface roughness, residue cover and soil clod size. At slower speeds the passive element draft requirements were less. In these cases, the benefits of active-passive tillage were not as great. When rotor velocity ratios were great, considerable throwing and pulverizing of the soil occurred. In these circumstances, it was observed that the soil appeared more well tilled by the active elements compared to the tillage performed by the passive elements.

Increasing tillage tool width results in higher overall draft requirements. The active-passive tillage machine could be used to make wider tillage machines that do not require high draft. The active-passive tillage machine was capable of producing negative draft. This negative draft could be utilized by adding additional passive elements. The passive elements could be set at different tillage depths. For instance, a passive element tillage depth could be found where all the draft requirements of the passive elements could be supplied by the active elements. By minimizing the number of active elements and maximizing the number of passive elements, the cost of the active-passive tillage machine could become more economical. As an example, at 8.0 km/h, active:passive depth ratio of 1.6:1 and velocity ratio of 1.9:1, a draft force of -1.03 kN was produced. The purely passive system at 8.0 km/h and 1.6:1 active:passive depth ratio required a draft force of 9.49 kN or 2.37 kN per passive element. Adding an additional passive element to the above 2 active-2 passive configuration would require only 1.34 kN of draft to be supplied by the tractor. This compares very favorably to the draft requirements of 11.85 kN
for a 5 element purely passive tillage tool.
NOMENCLATURE

\[ \lambda = \text{Velocity ratio.} \]

\[ V_p = \text{Rotor tip velocity, km/h.} \]

\[ V_f = \text{True ground speed of tillage machine, km/h.} \]

\[ \text{BL} = \text{Bite length, mm.} \]

\[ \omega_r = \text{Angular velocity of the rotors, rpm.} \]

\[ N = \text{Number of active elements per rotor.} \]

\[ AP = \text{Active to passive depth ratio.} \]

\[ d_a = \text{Depth of active elements, cm.} \]

\[ d_p = \text{Depth of passive elements, cm.} \]

\[ P_{fly,a/p} = \text{Engine flywheel power, active-passive machine, kW.} \]

\[ P_{fly,p} = \text{Engine flywheel power, passive tool, kW.} \]

\[ P_{pto} = \text{Pto power, kW.} \]

\[ P_{db} = \text{Drawbar power, kW.} \]

\[ P_{rr} = \text{Rolling resistance power, kW.} \]

\[ \eta_{tm} = \text{Tractor transmission efficiency, 0.885.} \]

\[ \eta_{tr} = \text{Traction efficiency, decimal.} \]

\[ R_f = \text{Dynamic vertical load on front tires, kN.} \]

\[ R_r = \text{Dynamic vertical load on rear tires, kN.} \]

\[ S = \text{Wheel slip, decimal.} \]

\[ \text{CN}_r = \text{Wheel numeric rear wheel.} \]

\[ \text{CN}_f = \text{Wheel numeric front wheel.} \]

\[ \text{CI} = \text{Cone penetrometer index for the first 15 cm, kPa.} \]

\[ b = \text{Tire section width, front or rear, cm.} \]

\[ d = \text{Tire diameter, 0.8[(b x 2) + d_{rim}], cm.} \]

\[ d_{rim} = \text{Tire rim diameter, cm.} \]

\[ W_{dr} = \text{Dynamic vertical load on rear tires, } W_{str} + W_{tr}, \text{kN.} \]

\[ W_{df} = \text{Dynamic vertical load on front tires, } W_{sff} + W_{tff}, \text{kN.} \]

\[ W_{str} = \text{Static rear axle load, kN.} \]

\[ W_{sff} = \text{Static front axle load, kN.} \]

\[ W_{tff} = \text{Weight transfer to rear wheel, 0.45(P), kN.} \]

\[ W_{tr} = \text{Weight transfer from front wheels, 0.29(P), kN.} \]

\[ P = \text{Horizontal draft force, kN.} \]

\[ P_{rr} = \text{Rolling resistance power, kW.} \]

\[ \mu_f = \text{Coefficient of rolling resistance of front tires.} \]

\[ \mu_r = \text{Coefficient of rolling resistance of rear tires.} \]

\[ \mu_t = \text{Gross traction coefficient.} \]

\[ V_t = \text{Tangential tire velocity, km/h.} \]

\[ W = \text{Tillage width, m.} \]

\[ P_{tot} = \text{Total power} = P_{PTO} + P_{db}, \text{kW.} \]

\[ SFC_{gross} = \text{Gross specific fuel consumption, L/ha.} \]

\[ SFC_{net} = \text{Net specific fuel consumption, L/ha.} \]
FU_{net} = \text{Net fuel use} = FU_{gross} - FU_{rolling}, \text{L/h.}

FU_{gross} = \text{Gross fuel use, L/h.}

FU_{rolling} = \text{Rolling resistance fuel use, L/h.}

SFU = \text{Specific fuel use, kW-h/L.}

MC = \text{Moisture content in dry basis, decimal.}

WT = \text{Total tractor weight, kN.}
CONCLUSIONS

Field tests of an experimental active-passive tillage machine were conducted to investigate the effects of velocity ratio, ground speed and active:passive depth ratio on machine performance. The active-passive tillage machine was able to reduce the draft by at least 50% in all cases compared to a purely passive tillage system. During many instances, especially at 4.8 km/h, a negative draft force was produced which meant the active elements were producing greater draft than the passive elements required.

The active-passive tillage machine also was able to reduce gross specific fuel consumption compared to a purely passive tillage tool. The active-passive tillage machine reduced the gross specific fuel consumption by a maximum of 28%. This maximum percent reduction occurred at a velocity ratio of 1.5:1, a ground speed of 8.0 km/h and an active:passive depth ratio of 1:1. These same conditions resulted in a 36% reduction in net fuel consumption. In general, the percent reduction in net specific fuel consumption was in the 20 - 30% range. In order to minimize the gross specific fuel consumption parameter, the active-passive tillage machine should be operated at velocity ratios of less than 2.0:1.

Velocity ratio also influence the estimated engine flywheel power of the active-passive tillage machine. At low velocity ratios ($\lambda < 2.0:1$), the engine flywheel power for the active-passive tillage machine was lower than that required by the passive tillage tool by up to 23%. This occurred at 8.0 km/h, an active:passive depth ratio of 1:1 and velocity ratio of 1.5:1. When the velocity ratio was greater than 2.0:1, more engine flywheel power was required for the active-passive tillage machine than for the passive tool.

During all experimental conditions, wheel slip was reduced by at least 50% with the active-passive tillage machine. In general, the active-passive tillage machine was able to reduce the draft, fuel consumption, wheel slip and engine flywheel power requirements compared to a similarly configured passive tillage tool. The velocity ratio was the most influential in affecting the operating parameters.
FUTURE CONSIDERATIONS

The active-passive tillage machine has a bright future. The tillage machine was able to reduce the draft, gross fuel consumption, wheel slip and engine flywheel power requirements compared to a passive tillage system. The velocity ratio was the most influential in affecting the operating parameters. But little is known about how velocity ratio affects tillage effectiveness. Research is being conducted to quantify the relationship of velocity ratio on soil properties, such as, bulk density, clod size, surface residue and roughness. This research will concentrate on whether the active-passive tillage machine has the similar tillage effectiveness as a passive tillage system.

With the present 2 active and 2 passive tillage machine, negative draft was sometimes produced. It would be advantageous to add more passive tillage elements to utilize this negative draft. The active:passive depth ratio could be adjusted, so that low draft requirements can still be obtained with a wider tillage system.

The machine used in this research had rotor tillage elements which were designed to be operated at a velocity ratio above 2.5:1 (Hendrick, 1980). From this research, it was concluded that to minimize fuel consumption, the velocity ratio should be kept below 2.0:1. In order to eliminate soil interference with the back side of the element, new active elements have been designed and fabricated to operate at these lower velocity ratios.
REFERENCES


ASAE Standards 1991. ASAE Standard S313.2 - Soil cone penetrometer. ASAE, St. Joseph, MI.


