J. agric. Engng Res. (1993) **55**, 277–297

Performance Characteristics of a Tillage Machine with Active–Passive Components

K. J. Shinners;* J. M. Wilkes;† T. D. England‡

* Agricultural Engineering Department, University of Wisconsin, Madison, Wisconsin, USA

† Wye College, University of London, Wye, Ashford, Kent, UK

‡ J-Star Industries, Inc., Fort Atkinson, Wisconsin, USA

(Received 4 August 1992; accepted in revised form 10 December 1992)

An experimental tillage machine that combines active, rotary-powered tillage elements with conventional passive chisel tines was field tested. Negative draught created by the active elements was used to contribute to the passive tine draught requirements. A tillage machine of this configuration 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 (ratio of rotor tip velocity to forward velocity), forward speed and active/passive depth ratio (ratio of depth of active elements to depth of passive elements) on machine performance. The draught of the active–passive tillage machine was at least 50% lower in all cases than a similar machine using purely passive tillage tools. 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 55% with the active–passive tillage machine. The active–passive tillage machine was able to reduce the draught, fuel consumption, wheel slip and engine flywheel power requirements compared to a similar passive tillage tool when the velocity ratio was less than 2·0. The velocity ratio was the most influential factor affecting the operating characteristics.

1. Introduction

Approximately 20% of the energy for production agriculture is used for field operations, with a majority of this energy applied in tillage operations (Stout *et al.*[1]). Nearly all tillage tools utilize passive tillage elements. A draught 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–tyre interface and the tractor drawbar. Because of the poor efficiency of power transmission at the soil–tyre interface, tillage energy efficiencies are low. Also, because tractors require considerable weight to provide necessary traction, detrimental soil compaction may occur, and increased power is required to overcome the wheel slip and rolling resistance of the tractor tyres.

One way to bypass the inefficient soil–tyre interface is through active or rotary-powered tillage elements. Active elements are usually powered by the tractor pto drive. Tillage tools which utilize active elements transmit power directly to the soil rather than being

Paper presented at AG ENG 92, Uppsala, Sweden, 1–4 June 1992.

0021-8634/93/080277 + 21 $08.00/0

Notation

AP	Active to passive depth ratio.
b	Tyre section width, front or rear, mm.
BL	Bite length, mm.
CI	Cone penetrometer index for the first 150 mm, kPa.
CN_f	Wheel numeric front wheel.
CN_r	Wheel numeric rear wheel.
d	Tyre diameter, $0{\cdot}8[(b \times 2) + d_{rim}]$, mm.
d_{rim}	Tyre rim diameter, mm.
D_a	Depth of active elements, mm.
D_p	Depth of passive elements, mm.
FU_n	Net fuel use $= FU_g - FU_{rr}$, l/h.
FU_g	Gross fuel use, l/h.
FU_{rr}	Rolling resistance fuel use, l/h.
MC	Moisture content on dry basis, decimal.
N	Number of active elements per rotor.
P	Horizontal draught force, kN.
$P_{a/p}$	Engine flywheel power, active–passive machine, kW.
P_{db}	Drawbar power, kW.
P_p	Engine flywheel power, passive tool, kW.
P_{pto}	Pto power, kW.
P_{rr}	Rolling resistance power, kW.
P_t	Total power $= P_{pto} + P_{db}$, kW.
R_f	Dynamic vertical load on front tyres, kN.
R_r	Dynamic vertical load on rear tyres, kN.
S	Wheel slip, decimal.
SFC_g	Gross specific fuel consumption, l/ha.
SFC_n	Net specific fuel consumption, l/ha.
SFU	Specific fuel use, kWh/l.
V_p	Rotor tip velocity, km/h.
V_f	True ground speed of tillage machine, km/h.
V_t	Tangential tyre velocity, km/h.
W	Tillage width, m.
W_{df}	Dynamic vertical load on front tyres, $W_{stf} + W_{tf}$, kN.
W_{dr}	Dynamic vertical load on rear tyres, $W_{str} + W_{tr}$, kN.
W_{stf}	Static front axle load, kN.
W_{str}	Static rear axle load, kN.
W_{tf}	Weight transfer to rear axle, $0{\cdot}45(P)$, kN.
W_{tr}	Weight transfer from front axle, $0{\cdot}29(P)$, kN.
WT	Total tractor weight, kN.
η_{tn}	Tractor transmission efficiency, 0·885.
η_{tr}	Tractive efficiency, decimal.
λ	Velocity ratio.
μ_f	Coefficent of rolling resistance of front tyres.
μ_r	Coefficient of rolling resistance of rear tyres.
μ_t	Gross traction coefficient.
ω_t	Angular velocity of the rotors, rev/min.

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.*[2]). Purely active tillage machines return power to the tractor's drawbar by pushing the tractor. This may result 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 draught, weight transfer takes place from the rear to the front axle. This increases the rolling resistance requirements of the tractor (Wismer *et al.*[2]).

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 contributing towards the draught 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–tyre interface, (2) the negative draught of the active elements can be used to provide some or all of the draught of the passive elements and contribute to the rolling resistance of the tractor, (3) reduced draught of tillage machines will result in less wheel slip, thus improving field productivity, (4) reduced draught of tillage machines will allow use of lighter tractors, reducing soil compaction and possibly reducing tractor cost, and (5) reduced draught of tillage machines will allow operations to be performed in more difficult traction conditions that currently require the use of extra ballast, dual tyres or assistance from the front wheels.

A drawback to tillage tools with both active and passive elements is that the tillage tools become more complex. Tillage tools will change from a machine with virtually no moving parts to a machine with gearboxes, chains, bearing, and shafts. 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.

The general objectives of this research were to investigate the performance characteristics of a tillage machine similar to a chisel plough that uses active–passive components. The specific objectives were to compare engine power, specific fuel use, draught and tractor wheel slip when operating an active–passive tillage machine and a purely passive tillage tool at different velocity ratios, forward speeds and active/passive depth ratios.

2. Previous research

Several research projects demonstrated the concept of combining active and passive elements on one tillage machine. Chamen *et al.*[3] 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 760 mm diameter rotor. L-shaped blades were bolted to this rotor to cut the soil. To balance the forward thrust from the rotating blades, deep chisel tines were mounted behind the rotary unit. The tool obtained 80–100% of its power from the pto. Net specific energy requirements for the machine were 50% less than for a mouldboard plough operating at the same depth as the passive elements.

Wilkes and Addai[4] 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 mouldboard bottom, which 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 draught for the mouldboard bottom. When compared to a mouldboard plough operating at the same depth, the "Wye Double Digger" was found to reduce the draught power, wheel slip and specific energy under some experimental conditions.

In the early 1970s, Hendrick and Gill[5–7] wrote a series of papers quantifying design parameters for rotary tillers. Hendrick and Gill[5,6] determined that low rotor speed and forward rotor rotation were necessary to reduce the draught and power requirements of rotary tillage tools. However, a rotor speed that was too slow caused excessive variation in tillage depth because of blade kinematics (Hendrick and Gill[7]). A fast rotor speed disturbed more soil than necessary and required excess power (Hendrick and Gill).[6]

Hendrick[8] suggested a practical lower limit of the ratio of rotor tip speed to forward speed, i.e. velocity ratio, to be 2·5. Hendrick[8] 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, the back side of the tillage elements of each rotor would contact uncut soil.

Hendrick[8] built and tested a single rotor-powered active tillage machine. He compared the power requirements of a similar passive chisel tine 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 tine. 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 draught 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[9] built a single rotor active tillage machine similar to Hendrick.[8] The aim 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.

Early tillage machines which combined active and passive tillage elements generally contained active elements which tilled the whole width of the machine. Also, the passive elements of a few machines were not strategically positioned because they were usually located in the tilled soil after the active elements (Chamen *et al.*;[3] Wilkes and Addai[4]). Because of current interest in leaving some crop residue on the soil surface after tillage to reduce soil erosion, these types of tillage tools might be considered inappropriate because they tend to bury most of the crop residue. The active–passive tillage machine used by Shinners *et al.*[10] 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 plough. 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. Because this active–passive tillage machine was substantially different than others in that it tilled only distinct zones rather than the full width of the machine, further research on its performance characteristics was warranted.

Four experiments were conducted by Shinners *et al.*[10] dealing with depth of tillage, bite length, and active/passive depth ratio. The combination tillage machine with two active and two passive elements required 87% less draught and draught power than a tool with four 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 two active and two passive element tool when compared with a four passive element configuration.

3. Machine description

The experimental machine (*Figs 1* and *2*) used in this research was designed and built at the National Soil Dynamics Laboratory, USDA-ARS in Auburn, Alabama. However, it was extensively redesigned at the Agricultural Engineering Department of the University of Wisconsin-Madison. 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 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 pto shaft through a variable speed belt drive, a right angle gearbox and finally a double chain speed reduction drive. With this drive system the rotor speed could be readily adjusted between 21 and 103 rev/min, 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

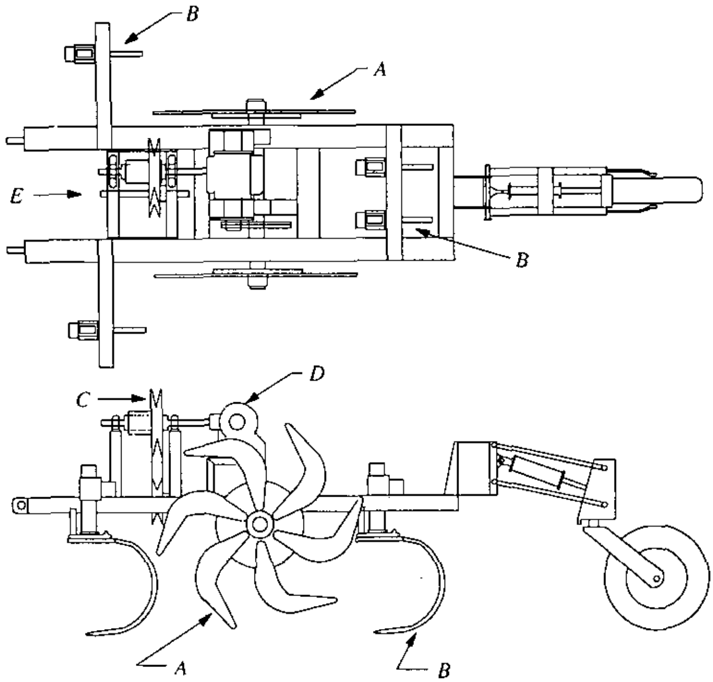

Fig. 1. Scheme of active–passive tillage machine: A, active elements; B, passive elements; C, variable pitch V-belt drive; D, gearbox; E, pto input

Fig. 2. Active–passive tillage machine

outboard from the active elements. The mounting of the passive elements was designed such that changing depth and removal of the tines 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 380 mm. The passive elements were standard spring-cushioned tines with straight shovel soil engaging tools with a 50 mm width. The soil engaging portion of the active elements were 25 mm wide. The six tillage elements on each rotor were designed such that the back side of the blade would not interfere with uncut soil at velocity ratios greater than 2·5. In order to prevent residue from accumulating on either the active or passive elements, plain disc coulters were located in front of all tillage elements.

4. Quantification of machine performance

To evaluate the performance characteristics of the active–passive tillage machine, these measurements were made in field tests.

1. Tillage depth of active and passive elements.
2. Rotational speed of active elements.
3. True ground speed, axle speed and fuel use of tractor.
4. Pto torque and speed.
5. Draught force.

In order to describe the operational characteristics of the active elements, a parameter known as the velocity ratio was used. Hendrick and Gill[5] described the velocity ratio as:

$$\lambda = V_p/V_f \tag{1}$$

The velocity ratio controls the operating performance of the active tillage elements in terms of draught reduction and tillage effectiveness (Hendrick *et al.*[5]). The velocity ratio is also related to the bite length of the active elements. The bite length is the distance between successive cuts measured at the soil surface:

$$\mathrm{BL} = (V_f \times 10^6)/(\omega_r \times N \times 60) \tag{2}$$

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 300 mm during all tests. The passive tines were raised or lowered to change the depth ratio. This is one method that can be used to control the amount of draught required by the tractor. The active/passive depth ratio was defined as:

$$\mathrm{AP} = D_a/D_p \tag{3}$$

A Massey-Ferguson Model 2675 tractor and the tillage machine were equipped with instruments to measure draught force, pto torque, pto rotational speed, rotor rotational speed, true ground speed, tractor rear axle speed, and fuel flow rate. All data from the transducers to be described below were 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[11] described the portable three-point linkage dynamometer which was used to measure draught force. The dynamometer was attached to the three-point linkage 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 rate 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 the tractor and measured pto torque and speed. Rotor speed was measured at the main gearbox input shaft with a magnetic pickup.

All 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 automatically transferred the average values for each transducer into a Campbell Scientific Storage Module SM192. These data were then transferred to a microcomputer, converted into an ASCII file and manipulated as required. From the recorded parameters, machine performance variables such as total draught, specific fuel use, wheel slip, pto power and draught 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 pto and drawbar with the active–passive tillage machine. When comparing the power required 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 tyres:

$$P_{a/p} = P_{pto}/\eta_{tn} + (P_{db} + P_{rr})/(\eta_{tr} \times \eta_{tn})$$

$$P_{p} = (P_{db} + P_{rr})/(\eta_{tr} \times \eta_{tn}) \tag{4}$$

The tractor transmission efficiency was assumed to be 0·885. This represents an average value taken from ASAE Engineering Practice EP 496 (ASAE[12]). Two other variables needed to be estimated. One was rolling resistance power and the other was tractive efficiency. Wismer and Luth[13] estimated the latter parameter through empirical modelling as:

$$\eta_{tr} = [1 - \{\mu_r + [(R_f + R_r)(\mu_f)]/\mu_t\}](1 - S) \tag{5}$$

$$\mu_r = (1{\cdot}2/CN_r) + 0{\cdot}04 \tag{6}$$

$$\mu_f = (1{\cdot}2/CN_f) + 0{\cdot}04 \tag{7}$$

$$\mu_t = 0{\cdot}75(1 - e^{-0{\cdot}3 \times S \times CN_r}) \tag{8}$$

Wismer and Luth[13] developed a term called wheel numeric which takes into account the geometry of the tyres and also soil strength:

$$CN_r = (CI \times b \times d \times 10^{-6})/(0{\cdot}5 \times W_{dr}) \tag{9}$$

$$CN_f = (CI \times b \times d \times 10^{-6})/(0{\cdot}5 \times W_{df}) \tag{10}$$

It was assumed that, for a semi-mounted implement, the weight transfer to the rear axle and from the front axle was 0·45 and 0·29 of the measured draught, respectively (Hunt[14]). Appendix A 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 both the front and rear tyres.

Wismer and Luth[13] also estimated the rolling resistance power:

$$P_{rr} = [(R_f \times \mu_f) + (R_r \times \mu_r)] \times V_f \times (10^3/3600) \tag{11}$$

Wheel slip was given by Bekker[15] which quantifies the efficiencies of the driven tyres:

$$S = 1 - (V_f/V_t) \tag{12}$$

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

$$SFC_g = (FU_g \times 10)/(V_f \times W) \tag{13}$$

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

$$SFC_n = (FU_n \times 10)/(V_f \times W) \tag{14}$$

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 at the pto and drawbar:

$$SFU = P_t/FU \tag{15}$$

The average moisture content for the top 0·3 m of soil was determined for each field test on a dry basis using oven drying at 103°C for 72 h. Penetration resistance forces were measured with a hand operated penetrometer at 30 mm intervals to a depth of 0·3 m. The cone diameter was 13 mm, the cone angle was 30°, and the tests were conducted in accordance with ASAE Standard S313.2 (ASAE[16]).

5. 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 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 draught 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, namely, 4·8, 6·4 and 8·0 km/h. Four velocity ratios were used: 2·5, 2·2, 1·9 and 1·5. Three active/passive depth ratios were used: 1·6, 1·3 and 1·0. The three active/passive depth ratios were obtained by keeping the active elements at a constant tillage depth of 300 mm and then operating the passive chisels at 190, 220 and 300 mm, respectively. 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 tines tilling at 300 mm depth and the other two tines operating at 190, 220 or 300 mm to correspond to the active/passive depth ratios of 1·6, 1·3 and 1·0. For all configurations, the draught requirements included that required by the four residue cutting coulters.

All field experiments were conducted at the University of Wisconsin-Madison Arlington Agricultural Research Station 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 5 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. Liquid ballast was added to the rear tyres, so an additional 9 kN of wheel loading was added during tests of the purely passive machine.

On August 16, 1991 an experiment was conducted to determine the fuel used in overcoming the rolling resistance of 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 stationary and at 4·8, 6·4 and 8·0 km/h. The above experimental conditions were repeated with the tractor both unballasted and ballasted with about 9 kN of liquid ballast in the rear tyres. The average values of fuel used were recorded for each test.

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

6. Results

The data from the 5 test days were grouped according to similar active/passive depth and velocity ratios (Tables 1–3). A negative horizontal draught force indicates a force pushing on the tractor and a positive vertical force indicates a downward force. Each value in these tables represents the average of from 14 to 15 total replications. 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 percentage reduction in the ten operating parameters. The lines plotted in *Figs 3–6* are based on linear regression of the data presented in Tables 1–3. These lines are presented only to show the trends of the data. To model the data accurately, a multiple regression equation would be necessary. For instance, besides velocity ratio, the gross specific fuel consumption was influenced by soil moisture, cone penetration resistance and tractor weight. The last three variables would be necessary to account for the day-to-day variations (Table 5). Because this type of model contains multiple predictor variables, they tend to model the trends in the data very closely. If these models are used to predict a parameter, such as draught, for soil conditions different from those of this research, the developed equations may prove to be inappropriate. Therefore, the simple linear regression models were chosen to show the trends in the data.

In all experimental conditions, the active–passive tillage machine reduced the draught of the passive machine by at least 55% (Table 4). Often the reduction was greater than 100%, which indicated the active elements were pushing the tractor by creating a negative draught larger than that needed by the passive elements. Generally, higher negative draught occurred at lower ground speeds, and greater velocity and active/passive depth ratios (*Fig. 3*). Greater negative draught was produced at lower ground speeds because the passive-only draught is less at lower speeds. However, the negative draught created by the active elements is a function of velocity ratio, not just ground speed. The draught

Table 1

Summary of the operating characteristics for the active–passive tillage machine operating at 4·8 km/h ground speed

Depth ratio	Velocity ratio	Draught force, kN	Vertical force, kN	Fuel flow, l/h	Gross fuel consumption, l/ha	Specific fuel use, kWh/l	Net fuel flow, l/h	Net fuel consumption, l/ha	Pto power, kW	Draught power, kW	Estimated rolling power, kW	Estimated flywheel power, kW	Modelled tractive efficiency, %	Wheel slip, %
1/1	2·5	−1·15	5·76	16·12	22·26	2·02	9·12	12·63	34·0	−1·5	5·7	46·7	59·7	3·09
	2·2	−0·94	5·72	15·86	22·08	1·83	8·82	12·25	30·2	−1·2	5·7	42·4	63·8	3·69
	1·9	−0·49	5·37	14·23	19·83	1·60	7·23	10·10	23·4	−0·6	5·7	36·2	61·2	3·27
	1·5	1·11	5·06	13·82	19·36	1·57	6·85	9·54	27·7	1·4	5·7	44·1	65·8	4·07
	0	12·19	8·75	15·56	21·42	1·00	8·06	11·04	0·0	16·1	9·2	38·2	71·3	9·99
LSD		1·19			1·22									0·75
1·3/1	2·5	−2·44	5·42	14·57	20·39	1·87	7·53	10·57	30·4	−3·2	5·7	39·5	57·1	2·82
	2·2	−2·52	5·13	13·78	18·90	1·80	6·85	9·35	28·1	−3·4	5·7	37·4	48·7	2·19
	1·9	−2·04	5·44	13·10	18·05	1·61	6·06	8·37	23·7	−2·7	5·7	33·4	53·3	2·49
	1·5	−1·06	4·53	12·64	17·56	1·47	5·79	7·95	20·0	−1·4	5·7	31·3	57·9	2·89
	0	10·12	8·51	14·80	19·83	0·92	7·34	10·20	0·0	13·7	6·9	34·2	71·0	8·6
LSD		0·73			1·12									1·00
1·6/1	2·5	−0·97	5·04	13·51	18·61	1·76	6·55	9·26	28·3	−4·7	5·7	34·8	42·1	1·87
	2·2	−2·55	5·20	12·91	18·05	1·80	5·87	8·04	26·5	−3·3	5·7	35·7	48·0	2·15
	1·9	−2·20	5·26	12·98	17·96	1·55	5·94	8·04	22·8	−2·9	5·7	31·8	54·0	2·54
	1·5	−1·75	3·77	12·41	17·12	1·31	5·60	7·58	18·5	−2·3	5·7	29·0	48·7	2·19
	0	8·83	8·22	13·78	18·80	0·84	6·36	8·79	0·0	11·7	6·9	30·9	71·0	8·67
LSD		0·72			0·84									0·62

Table 2

Summary of the operating characteristics for the active–passive tillage machine operating at 6·4 km/h ground speed

Depth ratio	Velocity ratio	Draught force, kN	Vertical force, kN	Fuel flow, l/h	Gross fuel consumption, l/ha	Specific fuel use, kWh/l	Net fuel flow, l/h	Net fuel consumption, l/ha	Pto power, kW	Draught power, kW	Estimated rolling power, kW	Estimated flywheel power, kW	Modelled tractive efficiency, %	Wheel slip, %
1/1	2·5	−0·64	5·64	19·07	19·55	2·18	12·38	12·53	43·0	−1·2	7·5	61·4	58·9	2·99
	2·2	−0·85	5·35	18·43	18·61	2·12	11·77	11·88	40·6	−1·5	7·5	58·2	57·9	2·89
	1·9	0·99	5·22	17·87	18·33	1·83	10·98	11·22	35·6	1·7	7·5	57·9	62·5	3·46
	1·5	5·87	3·85	14·88	15·34	1·50	8·21	8·51	11·9	10·3	7·5	43·6	70·0	5·41
	0	13·23	9·00	19·04	19·27	1·24	11·28	11·50	0·0	23·6	9·2	54·6	71·2	10·94
LSD		0·74			0·16									0·89
1·3/1	2·5	−2·25	5·23	16·54	16·56	2·14	8·93	8·95	39·6	−4·1	7·5	56·0	36·2	1·66
	2·2	−2·32	4·98	16·43	16·65	2·12	9·77	9·91	38·9	−4·2	7·5	52·1	49·6	2·24
	1·9	−1·54	5·01	15·37	15·62	1·93	8·48	8·60	32·3	−2·7	7·5	47·0	54·1	2·55
	1·5	2·61	3·49	13·66	13·94	1·33	7·08	7·20	13·5	4·7	7·5	37·1	66·2	4·15
	0	10·72	8·50	16·92	16·93	1·15	9·24	9·26	0·0	19·3	9·2	47·4	71·2	9·37
LSD		1·00			0·19									0·93
1·6/1	2·5	−1·41	22·91	16·50	16·65	1·88	9·84	9·91	36·6	−5·9	7·5	46·6	38·7	1·74
	2·2	−3·27	4·83	15·14	15·34	1·88	8·48	8·60	36·8	−5·8	7·5	49·3	25·9	1·39
	1·9	−2·16	4·90	14·84	15·16	1·90	7·83	8·04	31·8	−3·8	7·5	45·0	48·5	2·18
	1·5	1·85	3·16	12·91	13·00	1·13	6·36	6·45	11·2	3·3	7·5	33·1	63·2	3·55
	0	10·14	7·68	15·82	15·99	1·15	8·29	8·32	0·0	18·1	9·2	45·6	70·8	8·05
LSD		0·90			1·31									1·78

Table 3

Summary of the operating characteristics for the active–passive tillage machine operating at 8·0 km/h ground speed

Depth ratio	*Velocity ratio*	*Draught force, kN*	*Vertical force, kN*	*Fuel flow, l/h*	*Gross fuel consumption, l/ha*	*Specific fuel use, kWh/l*	*Net fuel flow, l/h*	*Net fuel consumption, l/ha*	*Pto power, kW*	*Draught power, kW*	*Estimated rolling power, kW*	*Estimated flywheel power, kW*	*Modelled tractive efficiency, %*	*Wheel slip, %*
1/1	2·5	−0·40	4·90	23·69	19·27	2·22	16·54	13·47	53·4	−0·9	9·4	78·2	56·6	2·74
	2·2	0·29	4·36	23·13	19·08	2·26	16·12	13·00	51·0	0·6	9·4	78·1	58·3	2·92
	1·9	2·19	4·25	19·64	16·74	1·89	13·36	10·94	33·7	4·8	9·4	63·2	67·1	4·40
	1·5	5·06	3·63	18·21	14·78	1·73	11·09	9·07	20·1	11·3	9·4	58·1	69·6	5·20
	0	15·15	8·70	25·02	20·49	1·35	17·18	14·12	0·0	33·7	11·6	75·3	71·2	11·14
LSD		1·13			1·50									0·81
1·3/1	2·5	−1·88	4·96	19·19	15·43	2·32	11·66	9·35	48·0	−4·2	9·4	68·0	44·8	1·99
	2·2	−1·95	4·56	18·51	14·87	2·11	11·36	9·07	43·9	−4·4	9·4	62·4	46·5	2·07
	1·9	−0·22	4·24	17·56	14·31	1·86	10·41	8·42	33·2	−0·5	9·4	54·8	61·0	3·25
	1·5	2·91	2·90	15·22	12·44	1·64	8·21	6·69	18·5	6·5	9·4	49·9	65·0	3·87
	0	11·32	8·48	20·66	16·28	1·28	12·38	10·01	0·0	25·5	11·6	61·5	71·2	9·30
LSD		0·84			1·22									0·82
1·6/1	2·5	−2·51	4·53	14·69	14·78	2·13	11·32	9·07	45·0	−5·7	9·4	62·7	36·8	1·68
	2·2	−1·62	4·31	16·88	14·12	2·44	10·14	8·23	45·9	−3·7	9·4	69·5	38·9	1·75
	1·9	−1·02	4·14	16·81	13·75	1·86	9·42	7·67	34·8	−2·5	9·4	53·9	56·4	2·74
	1·5	2·40	2·33	14·65	11·69	1·52	7·68	6·17	16·9	5·5	9·4	47·5	62·2	3·38
	0	9·50	7·78	18·28	14·78	1·18	10·64	8·51	0·0	21·5	11·6	54·9	71·0	8·49
LSD		0·72			0·84									0·72

Table 4

Percentage reduction in values of the operating characteristics of the active–passive tillage machine compared to a similarly configured passive tillage machine

Depth ratio	*Velocity ratio*	*Draught force*	*Gross fuel flow*	*Gross fuel consumption*	*Specific fuel use*	*Net fuel flow*	*Net fuel consumption*	*Draught power*	*Engine flywheel power*	*Wheel slip*
8·0 km/h										
1/1	2·5	102·7	5·3	5·9	−64·0	3·7	4·6	102·7	−3·9	75·4
	2·2	98·1	7·6	6·8	−67·6	6·2	8·0	98·1	−3·7	73·8
	1·9	85·5	21·5	18·3	−39·7	22·2	22·5	85·7	16·2	60·5
	1·5	66·6	27·2	27·9	−27·8	35·5	35·8	66·5	22·9	53·3
1·3/1	2·5	116·6	4·3	5·2	−80·8	5·8	6·5	116·4	−10·7	78·6
	2·2	117·2	7·7	8·6	−64·6	8·3	9·4	117·3	−1·5	77·7
	1·9	101·9	12·5	12·1	−45·0	15·9	15·9	102·0	10·8	65·1
	1·5	74·3	24·2	23·6	−27·8	−33·6	33·2	74·6	18·8	58·4
1·6/1	2·5	126·5	19·7	0·0	−80·2	−6·4	−6·6	126·6	−14·3	80·2
	2·2	117·1	7·7	4·4	−106·2	4·6	3·3	117·0	−26·5	79·4
	1·9	110·8	8·1	7·0	−57·3	11·4	9·9	111·6	1·8	67·7
	1·5	74·8	19·9	20·9	−29·0	27·8	27·5	74·6	13·6	60·2
6·4 km/h										
1/1	2·5	104·9	−0·2	−1·5	−76·0	−9·7	−8·9	105·0	−12·5	72·7
	2·2	106·4	3·2	3·5	−71·0	−4·4	−3·3	106·5	−6·6	73·6
	1·9	92·6	6·2	4·9	−47·3	2·7	2·4	92·6	−6·0	68·4
	1·5	55·6	21·9	20·4	−20·6	27·2	26·0	56·4	20·1	50·6
1·3/1	2·5	121·0	2·2	2·2	−86·3	3·3	3·3	121·2	−18·3	82·3
	2·2	121·6	2·9	1·7	−84·4	−5·7	−7·1	121·5	−10·0	76·1
	1·9	114·4	9·2	7·7	−67·8	8·2	7·1	114·2	0·8	72·8
	1·5	75·6	19·2	17·7	−16·0	23·4	22·2	75·9	21·7	55·7
1·6/1	2·5	113·9	−4·3	−4·1	−64·2	−18·7	−19·1	132·6	−1·7	78·4
	2·2	132·2	4·3	4·1	−63·5	−2·3	−3·4	132·3	−8·1	82·7
	1·9	121·3	6·2	5·3	−65·5	5·5	3·4	120·9	1·3	72·9
	1·5	81·8	18·4	18·7	1·2	23·3	22·5	81·6	27·6	55·9
4·8 km/h										
1/1	2·5	109·4	−3·7	−3·9	−103·0	−13·2	−14·4	109·4	−22·3	69·1
	2·2	107·7	−1·9	−3·1	−83·4	−9·4	−11·0	107·6	−11·0	63·1
	1·9	104·0	8·5	7·4	−60·7	10·3	8·5	104·0	5·2	67·3
	1·5	90·8	11·2	9·6	−57·5	15·0	13·6	91·1	−15·7	59·3
1·3/1	2·5	124·1	1·5	−2·8	−103·6	−2·6	−3·7	123·2	−15·6	67·2
	2·2	124·9	6·9	4·7	−95·0	6·7	8·3	124·5	−9·3	74·5
	1·9	120·1	11·5	9·0	−75·0	17·5	17·9	119·6	2·3	71·1
	1·5	110·5	14·6	11·3	−59·2	21·1	22·0	110·1	8·4	66·4
1·6/1	2·5	111·0	1·9	1·0	−108·9	−3·0	−5·3	139·9	−12·4	78·4
	2·2	128·9	6·3	4·0	−113·1	7·7	8·5	128·2	−15·4	75·2
	1·9	124·9	5·8	4·5	−83·4	6·6	8·5	124·6	−2·3	70·7
	1·5	119·8	9·9	9·0	−55·6	11·9	13·8	119·9	6·4	74·7

versus velocity ratio plots suggest that higher velocity ratios result in larger negative draught. It is believed that this trend was due to the active 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 (Hendrick[8]). In this research however, velocity ratios well below 2·5 were required to obtain favourable fuel use results. When the back side of the blades contact uncut soil, less negative draught was produced because this interference caused a reverse torque to be applied to the rotor.

In all configurations, wheel slip was reduced by at least 50% (Table 4). The majority of

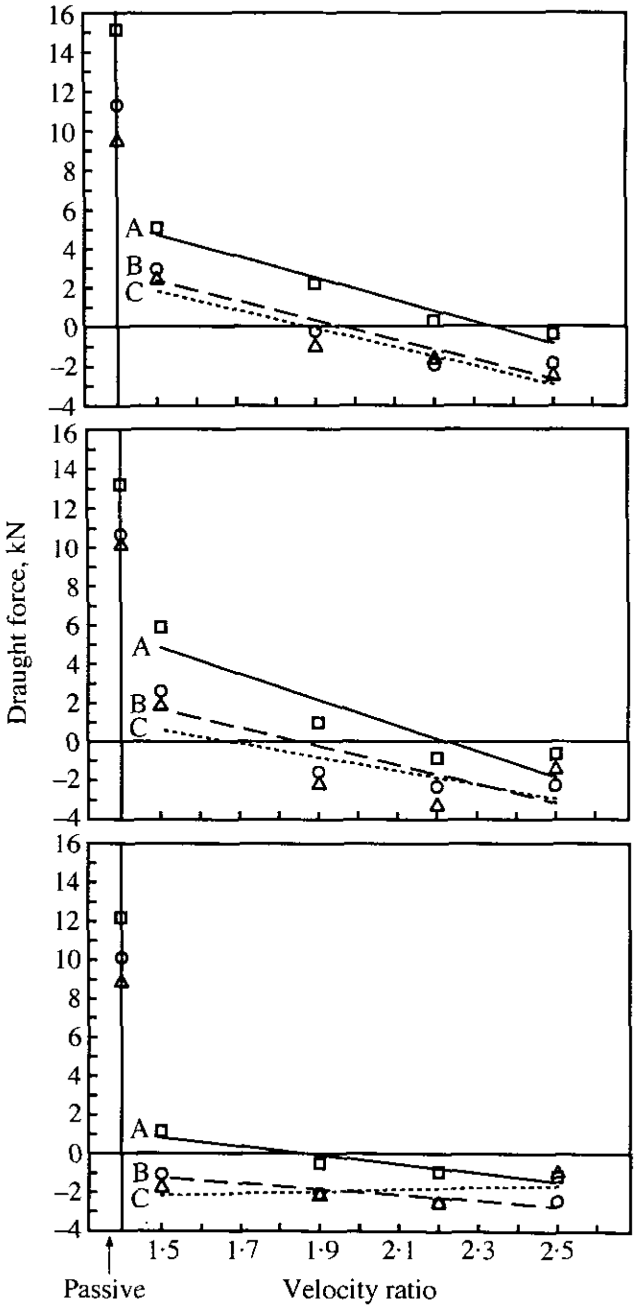

Fig. 3. Draught force versus velocity ratio: A, 1/1 depth ratio (□); B, 1·3/1 depth ratio (○); C, 1·6/1 depth ratio (△). (Top) 8·0 km/h; (middle) 6·4 km/h; (bottom) 4·8 km/h

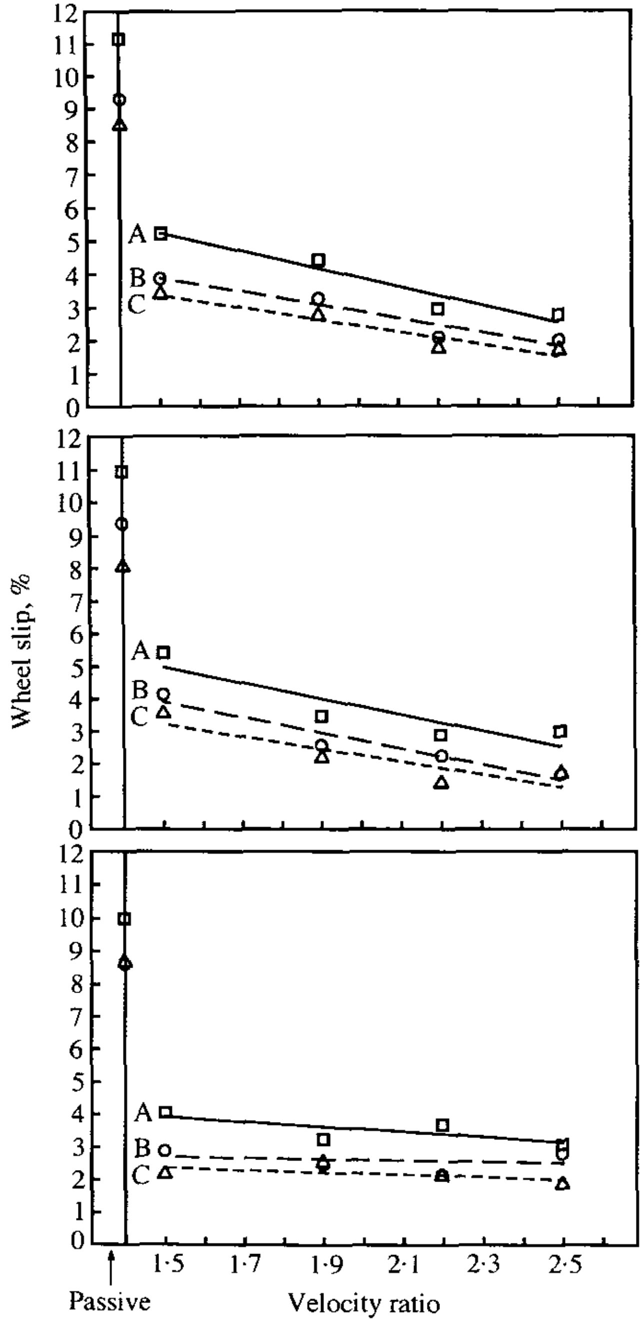

Fig. 4. Wheel slip versus velocity ratio: A, 1/1 depth ratio (□); B, 1·3/1 depth ratio (○); C, 1·6/1 depth ratio (△). (Top) 8·0 km/h; (middle) 6·4 km/h; (bottom) 4·8 km/h

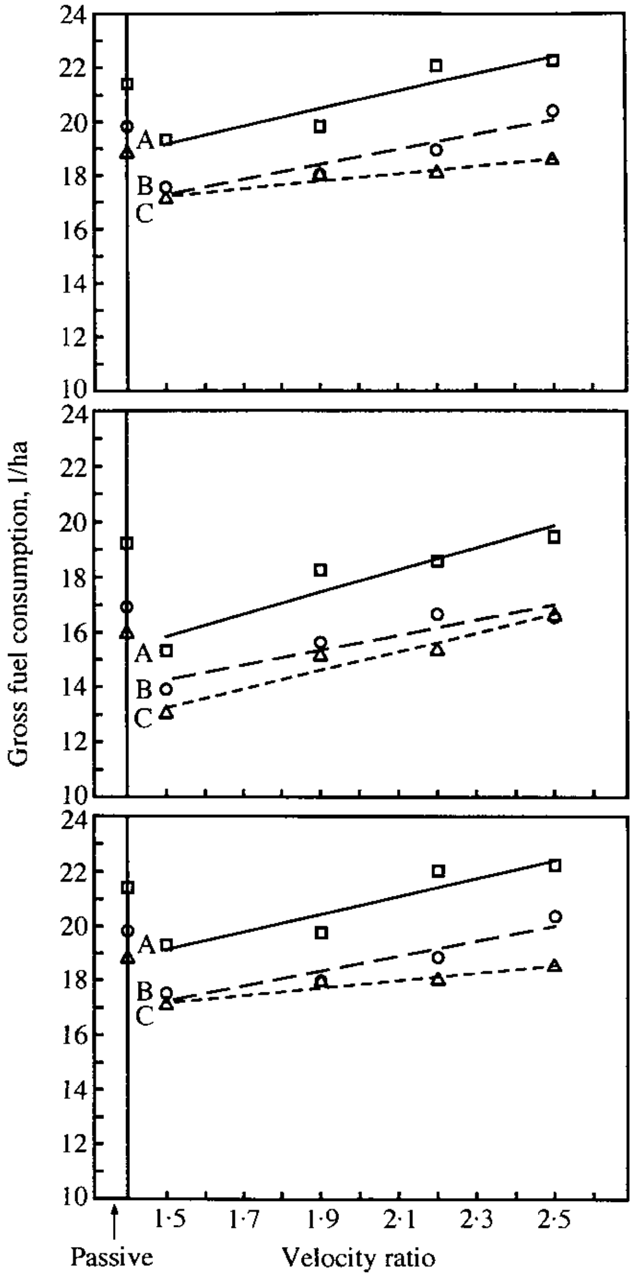

Fig. 5. Gross fuel consumption versus velocity ratio: A, 1/1 depth ratio (□); B, 1·3/1 depth ratio (○); C, 1·6/1 depth ratio (△). (Top) 8·0 km/h; (middle) 6·4 km/h; (bottom) 4·8 km/h

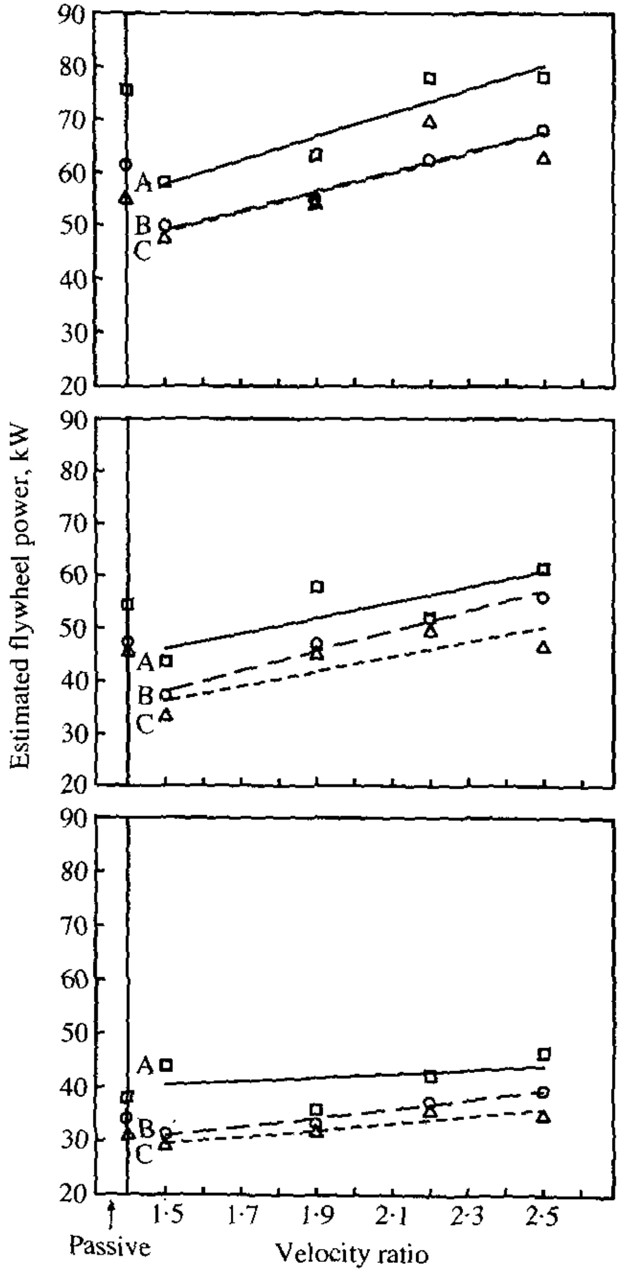

Fig. 6. Estimated flywheel power versus velocity ratio: A, 1/1 depth ratio (□); B, 1·3/1 depth ratio (○); C, 1·6/1 depth ratio (△). (Top) 8·0 km/h; (middle) 6·4 km/h; (bottom) 4·8 km/h

Table 5

Test conditions for May 20 and 29, June 26, August 5 and 19, 1991

Date, 1991	*Moisture content, dry basis decimal*	*Average cone resistance, kPa*	*Tractor weight, KN*
May 20	0·241	1228	60·7
May 29	0·257	1552	60·7
June 26	0·182	2413	69·7
August 5	0·172	2334	69·7
August 19	0·240	1505	60·7

the time, reduction was in the 70–80% range. Generally, lower wheel slip occurred at lower ground speeds, and greater velocity and greater active/passive depth ratios (*Fig. 4*). These are the same configurations that produced the greatest negative draught. Because of the low wheel slip which occurred with the active–passive tillage machine, it appears that lighter tractors could be used to operate the active–passive tillage machine, which could lead to increased field productivity and lower soil compaction.

In general, the largest percentage reduction in gross specific fuel consumption occurred at 8·0 km/h, and lower reductions occurred at the slower ground speeds (Table 4). 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 (*Fig. 5*). In order to achieve a reduced gross specific fuel consumption, the active–passive tillage machine had to be operated at velocity ratios below 2·0. At velocity ratios above 2·0, excessive energy was probably consumed by throwing and pulverizing the soil. The gross specific fuel consumption versus velocity ratio plots suggest that velocity ratio has less effect on fuel use at greater active/passive depth ratios and greater ground speeds, as indicated by the generally decreasing slope of lines in *Fig. 5*.

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 percentage reduction in terms of specific fuel use (Table 4). In other words, the tractor produced on average more output 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 active–passive tillage machine.

The MF 2675 tractor was greater in weight than that required to produce the draught power for four passive chisel tines. With no ballast in the rear tyres and pulling four passive tines, the maximum wheel slip was approximately 13%, while the general range was 5–10% wheel slip. This is generally lower wheel slip than deemed optimum with regard to tractive efficiency. The low wheel slip was caused by the large unballasted weight of the tractor (Table 5). Because of the excessive weight, a relatively large percentage of the gross fuel use went into rolling resistance. With net fuel use, the rolling resistance of the tractor was removed from the analysis. The net fuel use parameter helps describe the fuel use if the tractor and tillage tool combination were more closely matched in terms of size and weight. 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 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 (Table 4). At the lower two ground speeds and 2·5 velocity ratio, net fuel consumption was still generally higher than that of a similar passive tillage tool.

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). At 8·0 km/h, an active/passive depth ratio of 1/1 and velocity ratio of 1·5, the engine flywheel power was reduced by 23% compared to the passive tool requirements (Table 4). As the velocity ratio increased above 2·0, the engine flywheel power for the active–power tillage machine was greater than that required by the passive tool (*Fig. 6*). The largest difference occurred at 8·0 km/h, an active/passive depth ratio of 1·6 and a velocity ratio of 2·2. At this configuration, the active–passive tillage machine required 26% more engine flywheel power (Table 4). 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.

7. Discussion

Today's tillage tools are becoming larger and wider and are operating at faster speeds. These faster operating speeds can lead to higher draught requirements. The active–passive tillage machine has shown greatest benefits when operating at faster speeds and higher passive element draught requirements. The greatest benefits in terms reductions of draught and gross specific fuel consumption occurred at the higher ground speeds tested. At slower speeds the passive element draught requirements were less. In these cases, the benefits of active–passive tillage were not as great.

At the faster speeds and slower velocity ratios, the active–passive tillage machine performed equally as well as the passive system in terms of tillage effectiveness by observation. It was very difficult to differentiate visually 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. When rotor velocity ratios were great, considerable throwing and pulverizing of the soil occurred. In these circumstances, it was observed that the soil appeared to be tilled more by the active elements compared to the tillage performed by the passive elements.

Increasing tillage width results in higher overall draught requirements. The active–passive tillage machine could be used to make wider tillage machines that do not require high draught. The active–passive tillage machine was capable of producing negative draught. This negative draught 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 draught 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 and velocity ratio of 1·9, a draught force of −1·03 kN was produced. The purely passive system at 8·0 km/h and 1·6 depth ratio required a draught force of 9·49 kN or 2·37 kN per passive element. Adding an additional passive element to the above two active–two passive configuration would require only 1·34 kN of draught to be supplied by the tractor. This compares very favourably to the draught requirements of 11·85 kN for a five element purely passive tillage tool.

The tillage machine was able to reduce the draught, 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. Little is known, however, about how velocity ratio affects tillage effectiveness. Research is being conducted to quantify the relationship of velocity ratio on soil properties, such as volume of soil disturbed, 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.

The machine used in this research had rotor tillage elements which were designed to be operated at a velocity ratio above 2·5 (Hendrick[8]). From this research, it was concluded that to minimize fuel consumption, the velocity ratio should be kept below 2·0. In order to eliminate soil interference with the back side of the elements, new active elements have been designed and fabricated to operate at these lower velocity ratios.

An economic analysis is required to determine if the potential benefits of the active–passive tillage machine, such as reduced fuel use, increased productivity due to decreased wheel slip and reduced soil compaction due to lower ballast requirements, can offset the higher initial and operating costs of this machine compared to a conventional chisel plough.

8. 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 draught by at least 55% in all cases compared to a similar passive tillage system. During many instances, especially at 4·8 km/h, a negative draught force was produced which meant the active elements were producing greater draught than the passive elements required.

The active–passive tillage machine was also able to reduce gross specific fuel consumption compared to a purely passive tillage tool when the velocity ratio was less than 2·0. The active–passive tillage machine reduced the gross specific fuel consumption by a maximum of 28%. This maximum percentage reduction occurred at a velocity ratio of 1·5, 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 percentage 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.

Velocity ratio also influenced the estimated engine flywheel power required by the tractor to operate the active–passive tillage machine. At low velocity ratios ($\lambda < 2{\cdot}0$), the engine flywheel power requirement 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. When the velocity ratio was greater than 2·0, 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 draught, 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.

References

1 **Stout, B. A.,** *et al.* Energy use and production in agriculture. Council Agr. Sci. Tech. (CAST). Report No. 99, 1984, Ames, IA, Iowa State University Press

2 **Wismer, R. D.; Wegshied, E. L.; Luth, H. J.; Romig, B. E.** Energy application in tillage and earth moving. SAE Paper No. 68-677, 1968, SAE, Warren, PA

3 **Chamen, W. C.; Cope, R. F.; Patterson, D. E.** Development and performance of high output rotary digger. Journal of Agricultural Engineering Research 1979, **24:** 301–318

[4] **Wilkes, J. M.; Addai, S. H.** The "Wye Double Digger" as an alternative to the plough to reduce energy requirements and soil damage. International Conference on Agricultural Engineering, 1988, Paper No. 88-190

[5] **Hendrick, J.; Gill, W.** Rotary-tiller design parameter part I—direction of rotation. Transactions of the ASAE 1971a, **14**(4): 669–674

[6] **Hendrick, J.; Gill, W.** Rotary-tiller design parameter part II—depth of tillage. Transactions of the ASAE 1971b, **14**(4): 675–678

[7] **Hendrick, J.; Gill, W.** Rotary-tiller design parameter part III—ratio of peripheral and forward velocities. Transactions of the ASAE 1971c, **14**(4): 679–683

[8] **Hendrick, J. G.** A powered rotary chisel. Transactions of the ASAE 1980, **23**(6): 1349–1352

[9] **Araya, K.; Tsunematsu, S.; Wu, L.** A powered rotary subsoiler with pressurized sewage sludge injection. Transactions of the ASAE 1987, **30**(5): 1226–1230

[10] **Shinners, K. J.; Alcock, R.; Wilkes, J. M.** Combining active and passive tillage elements to reduce draught requirements. Transactions of the ASAE 1990, **33**(2): 400–404.

[11] **Thomson, N. P.; Shinners, K. J.** A portable instrumentation system for measuring tillage draught and speed. Applied Engineering in Agriculture 1988, **5**(2): 133–137

[12] **ASAE Standards.** ASAE Engineering Practice EP496—Agricultural machinery management, 1991. ASAE St Joesph, MI, USA

[13] **Wismer, R. D.; Luth, H. J.** Off-road traction prediction for wheeled vehicles. Transactions of the ASAE 1974, **17**(1): 8–11.

[14] **Hunt, D. R.** Engineering Models for Agricultural Production, 1986. Westport, CT, AVI Publishing Co.

[15] **Bekker, M. G.** Theory of Land Locomotion, 1956. Ann Arbor, MI, University of Michigan Press

[16] **ASAE Standards.** ASAE Standard S313.2—Soil cone penetrometer, 1991. ASAE St Joseph, MI

[17] **Shinners, K. J.; Wilkes, J. M.; England, T. D.** Energy use and productivity of an active–passive tillage machine. International Conference on Agricultural Engineering, 1992, Paper No. 9201-10

Appendix A

Table A1 shows the static wheel loads on the front and rear tyres when the rear wheels of the tractor did and did not contain liquid ballast. Table A2 shows the geometry of the front and rear tyres.

Table A1
Weight distribution of the MF 2675 tractor

Axle	*With ballast, kN*	*Without ballast, kN*
Rear axle	47·7	38·7
Front axle	22·0	22·0
Total tractor	69·7	60·7

Table A2
Tyre geometry of the MF 2675 tractor

Tyre	*Width mm*	*Rim diameter, mm*
Front	279	406
Rear	467	965