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Field Evaluation of an Electrically Powered Spiral Mechanization System

Daniel Sherman Humburg

FIELD EVALUATION

OF AN

ELECTRICALLY POWERED SPIRAL MECHANIZATION SYSTEM

BY

DANIEL S. HUMBURG

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This thesis is approved as a creditable and independent investigation by a candidate for the degree, Master of Science, and is acceptable for meeting the thesis requirements for this degree. Acceptance of this thesis does not imply that the conclusions reached by the candidate are necessarily the conclusions of the major department.

Thesis Advisor	Date
Major Advisor	Date
Head of Major Department	Date

ABSTRACT

An alternative method of powering and guiding agricultural implements was described and tested. The system utilized a center pivot irrigation gantry as a position reference device for an electrically powered tool-frame. The system induced the tool-frame to follow a reference point on the irrigation gantry, while that point was incremented outward by a ground driven mechanism. The path followed by the tool-frame was that of an Archimidean spiral.

A crop was planted and cultivated with the system, while evaluating the path for accuracy within operations, and repeatability between operations. A comparison of the overall energy efficiency and energy costs was made between the electric system and a conventional system.

Alterations to the existing spiral mechanization system were recommended and an alternate system using a linear move irrigation gantry as a position reference device and power source was suggested.

An analytical draft model was evaluated for use in predicting draft of low speed tillage implements. Comparisons were made between the draft predicted by the model for the cultivator, to the draft measured during the cultivation operation at different depths.

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Introduction

The mechanization of the production of agricultural crops has dramatically changed man's civilization over the last one hundred years. The introduction of machines, such as the tractor and the combine harvester, has helped to increase the productivity of an individual farmer many times over. Between the years of 1870 and 1985 the percentage of the United States populus employed in primary agriculture dropped from about 50% to 2.24% while the industry continued to supply this country and a portion of the world with an abundant and relatively inexpensive supply of food and fiber (Cooper, et al., 1947, and USDA 1985). Advances in mechanization continue to contribute to increased productivity while eliminating drudgerous, repetitive, tasks.

The use of sprinkler irrigation has improved the suitability of much agricultural land for production of high value cash crops. In 1985 in the United States, 10,957,098 hectares of land were irrigated with the use of sprinkler irrigation machines (Irrigation Journal,1987). Of this area, 5,425,621 hectares were irrigated with the use of center pivot machines. Crops produced under such irrigation systems range from coarse feedgrains and oilseeds to fruit and vegetable crops. The cost of purchasing, operating, and maintaining an irrigation system often represents a significant portion of the production costs of the crop being grown. Werner (1986) found the combination of ownership costs and operating costs of a center pivot irrigation system to represent approximately 25% of the production costs of a corn crop in eastern South Dakota. The use of the irrigation gantry to perform some second task, such as providing a mobile reference point for an implement steering system, would increase it's utility and practicality.

Approaches to automatically guiding agricultural implements have been researched since the early 1900s (Le Pori, et al., 1986). Many of these attempts made use of a field structure, such as the sidewall of a plow furrow from a previous pass, as the reference and guide for successive passes of the tractor and implement. Other systems have made use of buried cables carrying an electric current. A coil on board the tractor or implement was used to sense the magnetic field generated by the current in the cable and was able to provide a position signal to the steering control system.

Problems encountered by such automated systems have not been so much technical as they have been economic. Other advances in the size and nature of agricultural mechanization have increased the productivity of an individual operator to the point where the labor involved in field operations for production of the major feed and grain crops is only a very small portion of those crops' total production costs. The savings in labor, and any other advantages attributable to an automated system, must, in time, pay for any guidance system if it is to be accepted. Also, any system that is to rely upon replacement of the manual operator to justify it's use must be reliable enough to operate with little or no supervision. Systems requiring supervision will lose all, or a part, of their labor advantage. The fact that this labor savings has been insufficient to justify the installation of automated systems is evidenced by the fact that few, if any, automatic guidance systems for agriculture have progressed beyond the research stage.

Applications for which automatic guidance systems do merit consideration are those in which: 1) the machine is able to perform more accurately under automatic control than under manual control, and 2) the operation is one that is frequently repeated so that the labor involved in machine guidance is a significant portion of the crop production costs.

An example of the former is a hydraulic rowfinder on a sugarbeet harvester. This mechanism does not, in fact, replace a human operator, but is able to sense the crowns of sugarbeets in one target row and adjust the harvester position relative to the towing tractor to more accurately maintain the implement on the crop row. The benefit of this innovation is realized, not in lower labor costs, but in a reduction in harvest losses attributable to deviations from the ideal path, and consequently, a higher yield.

Examples of applications that require frequent trips over the same path include the mowing of turf in sod production and the frequent cultivation of some row crops. A possible application for automatic guidance is in the area of repetitive harvesting of vegetable and fruit crops. As selective harvesters are developed to replace manual harvesting of crops that must be frequently harvested over their growing season, the labor involved in operating the harvesters will become an increasingly large proportion of the production cost. In effect, the manual labor of harvest will be replaced by labor employed to guide harvesting machines. This labor is path oriented, and, as such, could be partially or completely automated. Such a harvesting system may provide a setting whereby the use of an automated guidance system is economically advantageous.

A difference, and potential advantage, of automated cropping systems that make use of existing irrigation machines, is the possible use of electric power as the energy source for the prime mover in the field. Use of electrical energy would reduce the operator's dependance upon petroleum fuel and could provide cost savings depending upon system efficiency and relative fuel costs.

A system to automatically guide an electrically powered tool frame in a predetermined spiral path was developed at South Dakota State University. Electric motors were used to power the drive wheels of a small tool frame or tractor. Electric power was supplied through a flexible cord from the axis of a center pivot irrigator. A control system was implemented to allow the tractor and implement to follow a moving reference point on the irrigation boom. A mechanical system was used to progressively move the guidance reference point along the boom, causing the tractor to follow a spiral path. This path began at the center of the pivot and terminated at the outer end of the single irrigation gantry span.

Since no practical field tests or demonstrations of this machine had been performed, research was initiated with the following primary objectives:

1) Determine the accuracy and repeatability of the spiral path control system for two different field operations while demonstrating those operations for a row crop.

2) Compare the energy efficiency and energy cost of an automated electric system to that of a conventional system.

3)Make recommendations for design improvements in subsequent automated systems using existing irrigation machines, and suggest possible applications for such systems.

Since measurements of draft using a shallow tillage cultivator were to be required to satisfy objective number two, and since existing models for soil implement draft were insufficiently validated for very low speeds, the following secondary objective was established:

1) To validate an existing soil implement draft model for use in subsequent designs of slow moving automated systems.

Review of Literature

Henry Ford said in his autobiography, "I have followed many a weary mile behind a plough and I know all the drudgery of it. What a waste it is for a human being to spend hours and days behind a slowly moving team of horses when, in the same time, a tractor could do six times the work," and later, speaking of work in general, "We have succeeded, to a very great extent, in relieving men of the heavier and more onerous jobs that used to sap their strength, but even when lightening the heavier labor we have not yet succeeded in removing monotony", (Ford,1922).

Henry Ford contributed to the mechanization of agriculture through the introduction of the Fordson, a small, lightweight tractor. Prior to its introduction, the cost of plowing an acre of land with horses was $1.46 (Ford, 1922). Ford calculated that the cost of plowing an equivalent area with a tractor powered by an internal combustion engine was $.95. Also, the time required to perform the task with the tractor was about one fourth that required by a team of horses and driver. Thus, a farm operator was able to reduce his time spent in the field, or alternatively, increase his farmed acreage while avoiding the necessity of managing draft animals. Today.

using medium sized equipment and average speeds, it currently requires approximately 0.85 hours to plow a hectare (2.43 acres) of land (ASAE 230.3, 1984).

Continued improvements in mechanization have resulted in production techniques that allow an average cash crop farmer to plant, cultivate, and harvest a hectare of corn with less than two hours of time spent in the field performing those operations (ASAE 230.3, 1984). Similarly the time required to mow, condition, and bale one hectare of hay three times during a season has been reduced to approximately three hours. The introduction of the mechanical tomato harvester has dramatically affected the commercial production of tomatoes. Schmitz and Seckler (1970), estimated that 205 manhours per hectare of human labor were displaced by the adoption of this innovation. They calculated that the mechanization of this process reduced a producer's harvest costs by \$5.41 to \$7.47 per ton of tomatoes harvested. Brandt and French (1983), calculated that the tomato harvester reduced from 50% to 20%, the portion of tomato production labor associated with harvesting. They also found that the harvester ultimately resulted in higher wage rates, better working conditions, and significant benefits to consumers. Similarly, the introduction of mechanical cotton harvesters reduced harvest costs for cotton producers by

$.045 per pound, or, $49 per hectare (Martin and Havlicek, 1977). However, many labor intensive operations are still not mechanized. An operation that has been essentially unmechanized is the harvesting of green asparagus. Humburg, et al. (1986) found the capacity of a manual laborer to harvest this crop to be between 1.2 and 1.8 hectares per day. Since this crop is harvested daily for nearly sixty days, the labor committed to harvest of one hectare is between 333 and 500 hours. A machine was proposed by Humburg, et al. (1986) to perform this harvesting operation. This machine would have performed the selection and removal of mature asparagus spears, replacing the manual portion of the actual harvesting. However, since the entire acreage of the crop still must be harvested daily, an estimated 41 hours per hectare would have been required to guide the harvester during the season. Labor intensive operations such as this have provided the stimuli for research to automate all, or a part of, the steering of vehicles for field work.

The ability of some automatic systems to perform operations more accurately, and at higher speeds, than their human counterparts is further reason for the deployment of automated systems.

Automatic Guidance Systems

A number of attempts have been made to automate the guidance of agricultural vehicles with the goals of reducing operator labor and improving overall machine performance. In a comprehensive review of automatic guidance in agriculture, Jahns (1976), organized previous research by the method used to produce guidance signals.

This review is restricted to systems that had some relevance to the guidance system under study. The following types of guidance were discussed:

1. Indirect vehicle guidance by a master vehicle.
2. Vehicle guidance by a directrix generated during a previous operation.
3. Vehicle guidance by a mechanical device installed in the field.
4. Vehicle guidance by a leader cable using alternating current.

Master/Slave Vehicle Guidance

The capacity associated with an operator controlled machine can be increased by coupling one or more driverless vehicles to one controlled by a single human operator. Field operations could be performed with several smaller and lighter tractor-implement combinations, or two or more different field operations could be performed in succession. The method of coupling the vehicles varies. A mechanical linkage coupling that was marketed in England has not recieved wide acceptance possibly due to problems with turning at headlands (Jahns, 1983). Non-contact linkages offer some advantages and a system using an energized coil on the leading vehicle was cited by Jahns (1983) and reported by Nielsen, et al. (1977). The slave tractor had, in these cases, three sensing coils to detect the location and relative position of the signal coil on the master vehicle. The amplitude of the voltages induced in the coils was used to locate the master vehicle and provide a signal for steering control.

Guidance by Directrix from a Previous Pass

This method of guidance relies upon some distinct feature of the crop or soil that is left by the implement from a previous pass. Examples include a plow furrow (Kirk, et al., 1976), or the cut edge of a crop being harvested. The sensing machanism follows this feature in a similar way that a blind man would follow a curb.

Optical ranging was used in a system described by Ambler, et al., (1980) to follow a plow furrow and to orient the tractor during headland turns. The furrow-following sensor consisted of a band of light projected across the furrow just in front of the tractor axle, and two sets of photo receptors. Light reflected from the bottom of the furrow was collected on one bank of receptors while light reflected from the unplowed surface was collected by a second set of receptors. A deviation from the desired path caused more of the reflected light to be collected by one set of receptors at the expense of the other set. After amplification, the detector outputs were compared. The difference between the two voltages gave a linear approximation of the error which was subsequently used to initiate a correction. Field tests of this portion of the system found it to be capable of following a furrow with an accuracy comparable to that of a human operator. The tractor's movements near a headland were controlled by a microprocessor-based controller which recieved input data from an opto-electronic range and bearing meter, and from a heading indicator. The range meter worked in conjunction with a set of reflecting posts that were placed along the field boundary. When this range and bearing system indicated to the microprocessor that a reverse turn was required, the microprocessor initiated a pair of arcs of predetermined duration that constituted a 180 degree turn. The tractor then used the optical ranging system to estimate its position and decide if any correction was required before returning guidance control to the furrow following device. After any required correction was made, the optical furrow follower would again be allowed to guide the tractor back across the field.

The feasibility of the concept was demonstrated, but the headland turning system was not found to be reliable enough to allow unattended operation, nor were modifications of the existing system found likely to significantly improve it.

A system developed by Parish and Goering (1970) used microswitches and pressure plates to locate the edge of a plot of standing alfalfa. The positive signal from the sensor switches was used to alter the speed of the left and right side drive motors on a hydrostatically powered windrower. A circuit using time delay relays was implemented to cause a turn of a given duration and to reverse the machine at the field ends. Tests of the system found it to be capable of following the edge of a crop with root-mean-square errors, from the ideal path, of 23 to 30 centimeters. That level of accuracy was comparable to an average human operator. This type of guidance tends to be specific to a single crop operation and implement, as for example, windrowing alfalfa. Other limitations include the necessity for several failsafe devices to stop the machine in the event of a malfunction. Use of this method of guidance has not been widely accepted in the industry, possibly because of farmer scepticism of the system's reliability when operating without human supervision.

The guidance of sugar beet harvester shares is a variation on this type of control system (Marchant and Chitty, 1966). In this case, the directrix employed was the crowns of the sugar beets in the target row. A set of spring loaded feelers was mechanically connected to a pair of sensitive hydraulic valves which, in turn, controlled the position of the harvester relative to the towing tractor and ground. A deviation from the row caused the crowns of the beets to displace the feelers, and this in turn, caused a corrective adjustment to be made via the control valves. Should the harvester lose the row completely, the tractor operator then acted as the fail-safe and made a course correction. This system is rather simple and rugged. It represents one form of automatic guidance system that is now considered "state of the art" and has been adopted commercially. Such a system is specific to, and dependant upon, the large, stable, directrix provided by mature sugar beet crowns (Hesse, 1974).

A driverless combine harvester was developed by Iseki Agricultural Machinery Mfg. Co. Ltd. in 1974 (Kanetoh, 1978). This machine was capable of driverless operation in a rectangular field once the headlands had been harvested with the aid of a human operator. Guidance along the edge of the unharvested crop was achieved with mechanical sensors. The mechanical sensors were connected to electrical transducers and activated hydraulic steering controls through a logic circuit. Ninety degree turns were accomplished through a preprogrammed set of maneuvers after which, the machine proceeded forward. If no crop was encountered after a fixed time interval the harvester was shut down. No indication of the path accuracy, or the level of commercial success of the machine was given.

A system for maintaining combine harvesters and forage harvesters accurately on corn rows has been offered commercially as a purchase option by Claas (Busse, et al., 1977). In this system the sensor used was a formed steel rod extending from the gathering points of the harvester. Deflection of these rods by the corn stalks was transformed to an electrical signal by magnetically influenced resistors located at the base of the rods and sensitive to their position. A similar magnetically influenced resistor was used to detect the position of the steering wheels on the harvester. An electronic control unit combined the signals of the crop row sensors and steering wheel positions to determine the necessary directional corrections and provided the control signal to a solenoid block valve to effect those corrections. Deviations from the row center were reported to be no more than $\pm$ 5 centimeters at speeds up to 9.7 km. This system did not eliminate the need for an operator, as he was still required to provide control during headland turns and act as the ultimate failsafe device. The system did, however, relieve the fatigue and monotony associated with continuous harvest and allowed the operator to monitor other machine functions while the harvester followed the crop row (Busse, et al., 1977). The estimated cost of retrofitting such a system to a combine harvester was $1500. No indication was given as to the degree of farmer acceptance of the optional system.

A system for performing other field operations on rows of growing plants, using the plant stems as the directrix, was investigated by Hesse (1974). In this case, capacitive sensors were employed to locate the plants in the row. The impulses given by the sensors as they contacted plants in the row were used to control a monostable electronic flip-flop. After amplification, the pulse signal from the flip-flop was used to activate an electrohydraulic on-off valve. This valve controlled the position of the implement relative to the row through a hydraulic cylinder. The impulse given by the sensor was of short duration, as opposed to a continuous error signal. A predetermined time interval, delta t, was chosen and the hydraulic valve was opened for that fixed interval of time so that the magnitude of the correction associated with each impulse was a constant. This system was found to be technically capable of steering cultivator machines automatically.

A limitation common to all systems utilizing a directrix made during a previous pass, or operation, is the potential variability of the directrix. The collapse of a furrow wall or a section of missing plants in a target row can cause the guidance mechanism to lose its place. Few, if any, systems are presently capable of re-establishing a directrix once it has been damaged, or temporarily lost. (Jahns 1983).

Guidance by a Directrix Installed in the Field

Systems of this type usually have significant installation costs, but have the advantage of being useful for many years. Jahns (1983) described a system proposed, but never built, by Reece (1968) for automatic guidance using concrete rails on the sides of fields of a uniform size and shape. In this system, the implement was to be drawn over the field in a predetermined pattern by an electric winch and cables. The winches were, in turn, to be mounted on a track on the concrete rails and controlled by a central computer. The track and cable were to be equipped with distance markers to facilitate the exact positioning of the implement carrier.

A system for towing implements across an area with cables was tested by LePori, et al. (1986). The goals of this system were to reduce soil compaction caused by wheel traffic and to increase the tractive efficiency of the field operation. An implement, in this case a tandem disc, was towed under a mobile truss by a diesel power unit with steel cables. An implement carrier was employed as a means of hitching implements and reversing them at the end of the pass across the field. The system has the potential for automatic operation as the power unit is a part of the mobile truss, and would not necessarily require an operator. Problems encountered with the system centered on the inability of the mechanism to maintain a straight line path across the field because of side forces from the soil on the implement. The cable towing system was unable to provide sufficient corrective side forces to counteract the implement forces. Errors from the desired path as large as one implement width were reported near the center of the test plot. A system to keep implements on the desired path would be required (Le Pori, et al., 1986).

Alcock (1982) described an automatic position control system for an electrically powered tool-frame. That system utilized a center pivot irrigation gantry as a revolving directrix in the field. A simple system of limiting switches caused the tool-frame to follow a reference point on the irrigation gantry. As the gantry revolved, the reference point was to be incremented out generating a spiral path. In subsequent work, Alcock and Jahns (1983) evaluated such a system's performance characteristics with a computer program. The principal disadvantage of the system noted was the low work rate achieved at the inner portion of the spiral where the tangential speed of the directrix and tractor were low.

The most thoroughly researched guidance mechanism appears to be the leader cable system that is installed in the field. Many systems have been proposed, and some developed, that make use of the magnetic field generated by an alternating current in a buried conductor. A set of three mutually perpendicular coils is typically used as a transducer to sense the magnetic field. The amplitudes of the induced voltages in the three coils can be used to calculate the relative position of the transducer to the leader cable, and the signal can be amplified to provide a signal for steering control (Lawson, et al., 1963, Rushing and Allen, 1969). This arrangement has been used to guide ships through shipping corridors (Gilles, et al., 1980), and agricultural vehicles in vineyards and orchards (Fabian and Jahns, 1976), as cited by Jahns,(1983).

A system was tested by Telle and Perdok (1979), that utilized two coils to sense the location of the conducting leader cable. The coils were separated on the front of the tractor with their axes parallel. The induced voltages in these two coils were compared and the difference used as an error signal. With the coils located an equal distance on each side of the leader cable, voltages of equal magnitude would be generated in each, giving the zero error condition. As with a three coil system, the vehicle could be made to operate to the side of the cable by displacing both coils by some lateral distance relative to the axis of the vehicle. In the systems evaluated by Telle and Perdok (1979), headland turns were made by a human operator while the automatic system was used only for straight line guidance.

A leader cable system evaluated by Young, et al.(1981) was found to be able to maintain a path, corresponding to a straight line buried conductor, of $\pm$ 4.1 centimeters at speeds of up to 13 km/hr. When the buried conductor cable was placed in a sinusoidal pattern, the maximum lateral path error of the tractor, at a speed of 11 km/hr, was 18.5 centimeters. That system utilized a microprocessor to evaluate the signals recieved from the sensing antennas according to an algorithm developed for the specific application. The algorithm determined the appropriate magnitude of turn correction for the error signal, and initiated the correction.

A major drawback to the adoption of leader cable systems for general agricultural use has been the need to implant cables at a spacing equal to the minimum width of the implement to be used, and the associated costs. The development of sensing systems capable of accurately guiding a vehicle parallel to widely spaced leader cables would eliminate much of this cost. Work has been done in this area by Brook (1968), and Gilles, et al. (1980). Jahns (1983) reported that a system developed at the Bundesforschungsanstalt fur Landwirtschaft in West Germany, had been shown to be capable of repeating a predetermined path with an accuracy of 2.5 cm using leader cables spaced up to 50 m.

Bibliographies relevant to automatic guidance in agriculture were compiled by Jahns (1976) and again by Jahns (1983).

Use of Electricity to Power Field Machines

In an examination of thermodynamic considerations for the use of electric power in agriculture, Fluck and Baird (1984) compared the overall energy efficiencies of electric power and natural gas for irrigation pumping. For that analysis, the boundaries of the comparative systems were from the point of fuel combustion to the shaft input at the pump. In the case of electrical power, the combustion was assumed to take place at a power plant, in a coal fired boiler, while for the natural gas case the combustion took place in an internal combustion engine in the field. Estimates of the energy content of the fuels used included the energy required to recover, refine, and transport the fuel to the combustion site. Fluck and Baird found the overall efficiencies to be identical for electricity and natural gas at 23%. They further stated that since the percentage of fuel energy converted to mechanical energy was essentially the same for these two power sources, it was then appropriate that the choice of energy source alternatives be made on the basis of factors such as operating costs, reliability and maintenance, and fuel availability. Overall efficiency for diesel fuel, as a power source, was estimated to be slightly lower than natural gas, because of the larger quantity of energy required to produce and refine diesel fuel (Fluck and Baird, 1984).

In a report on the feasibility of the use of electric tractors in U.S. agriculture (Anon., 1980), it was estimated that the efficiency of battery powered tractors, from the charging outlet to the drive motor shaft of the hypothetical tractors, was 33%. The estimate was based on the component efficiencies of the battery charger, battery discharge efficiency, motor efficiency, and controller efficiency. Assuming an electric power plant efficiency of 25.7%, including line losses, (Fluck and Baird, 1984), a battery powered tractor would have an overall efficiency of 8.5% from the fuel combustion to the tractor transmission. That is, it would convert 8.5% of the energy available in the fuel to usable shaft energy. However, efficiency losses of about 63% were attributed to the battery charging, discharging, and to the controllers (Anon., 1980). A tractor directly utilizing alternating current would eliminate the need for batteries, a charger, and controller. Elimination of these components would allow an overall efficiency closer to that found by Fluck and Baird for irrigation pumping.

Jordan (1983) illustrated the advantages that electric motors have over internal combustion engines in maintaining a reasonably high efficiency under varying loads. Internal combustion engines are able to maintain their maximum efficiency only over a narrow range of power output levels (Buck and Hughes, 1981).

Elamin (1981) compared a battery powered electric tractor to a gasoline powered garden tractor in a variety of field operations. Energy consumed in these operations was measured on a per hectare basis. A comparison was made between the energy that flowed through a watt-hour meter to charge the battery powered tractor, and the energy consumed by the petroleum powered tractor when the heat content of the fuel used was converted to watt-hours. Elamin noted that the electric tractor energy values were not reflective of the fuel energy consumed at the power plant in producing the electricity. He suggested multiplying the energy figures by a factor of 3 to represent a power plant efficiency of 33%. The comparison made, however, did not include this factor and indicated a substantial efficiency advantage for the electric vehicle. Elamin further suggested that petroleum powered garden tractors are not very efficient and that the energy consumption values found for the petroleum powered garden tractor were not representative of farm sized tractors.

A comparison was made by Vik (1985) between a 50 kW battery powered electric tractor and a diesel tractor of comparable horsepower. The heat content of the diesel fuel burned was compared to the electrical energy leaving the batteries of the electric tractor, as measured by a DC watt-hour meter. Both tractors were required to perform the same chore routine during the tests. The measured values of electrical energy were divided by a factor of .70 to account for inefficiencies in the charging and discharging of the batteries. This on-farm comparison of energy use indicated an energy savings of 57% to 76% for the electric tractor. Since this was a comparison of the energy as used on the farm, no consideration was given to the efficiency of the process used to generate the electrical energy used by the electric tractor.

Soil Tool Draft Models

McKyes (1985) reviewed the development of models for predicting soil cutting forces based on a development of the universal earthmoving equation proposed by Reece (1965). Reece proposed the following equation to determine the force necessary to cut soil with a tool.

$$P = (\gamma g d^2 N_G + c d N_c + q d N_q) w$$

Where: P = total tool force; γ = total soil density; g = acceleration due to gravity; d = tool working depth below the soil surface; c = soil cohesion strength; q = surcharge pressure acting vertically on the soil surface; w = tool width; and N_G, N_c and N_q are factors dependent upon the soil friction properties and on the tool geometry and tool to soil strength properties.

A method developed by Hettiaratchi (1969), (Hettiaratchi and Reece, 1974), and cited by McKyes (1985) to solve problems involving soil-tool force problems began with the basic equation given above. A term to include soil to tool adhesion (N_a) was added. Values of N_G, N_c, N_q, and N_a for the extreme cases of very rough and very smooth blades were calculated analytically for a variety of soil friction angles and tool rake angles. Methods were also given for interpolating N values for the more realistic cases where the tool blade is neither perfectly rough nor perfectly smooth. The treatment of the problem was two dimensional; ie. forces involved in moving soil to the side of the cutting tool were ignored.

A different approach to the prediction of tillage tool cutting forces was taken by Luth and Wismer (1971). They utilized similitude methods to reduce the number of independant variables and were thus able to study tool geometries and soil properties without a complete analytical solution. After appropriate variables were combined to form dimensionless terms, experimental data was used to determine coefficients and exponents for the terms. This method has been shown to give good results in certain specific sand and specific clay soils (Luth and Wismer, 1971, and Wismer and Luth, 1972).

In a three dimensional analysis of soil failure around a soil tool, Godwin (1974) observed that the shape of the soil failure crescents on the soil surface and to the sides of the blade were elliptical, but nearly circular. Assuming this circular shape to be reasonably accurate, the volume of soil affected by the soil tool to the side of the tool, could be estimated. The soil immediately ahead of the tool was analyzed two dimensionally as described earlier, (Hettiaratchi and Reece, 1974). The forces required to fail a side crescent were estimated by determining the incremental forces on a small soil segment and then integrating that force over the volume bounded by the circular arc that described a side failure crescent. Using these forces, Godwin and Spoor (1977) developed an equation for draft similar to that of Hettiaratchi and Reece.

$$H = (\gamma g d^2 N_G + c d N_c + q d N_q)(w+s)\sin(\alpha + \delta) + c_a d w \cot(\alpha)$$

where $s = r(1 - (d\cot\alpha/r)^2)^{1/2}$

In this expression, **s**, represents the ultimate width of each side crescent of disturbed soil and r is the forward distance of soil failure. The calculation of **s** however, is dependant upon tool aspect ratios (depth/width) and soil strength. Tests were performed with narrow flat tools in sandy soils (Payne and Tanner, 1956), (Godwin, 1974), that allow estimates of **s**, and then draft, to be made using the Godwin and Spoor model. These calculations of draft would, however, be limited to the soil types tested by Payne and Tanner, and Godwin.

McKyes and Ali (1977) attempted to develop a method of predicting soil-tool cutting forces that was independant of the need for many experimentally determined soil inputs. They began with a two-dimensional wedge shaped zone of soil, forward of the cutting tool, that was assumed to fail in a straight line from the tool cutting tip to the soil surface, at an angle beta, to the soil surface. In addition, a circular segment of soil to each side of the blade and forward failure zone, was assumed to fail (Figure 1).

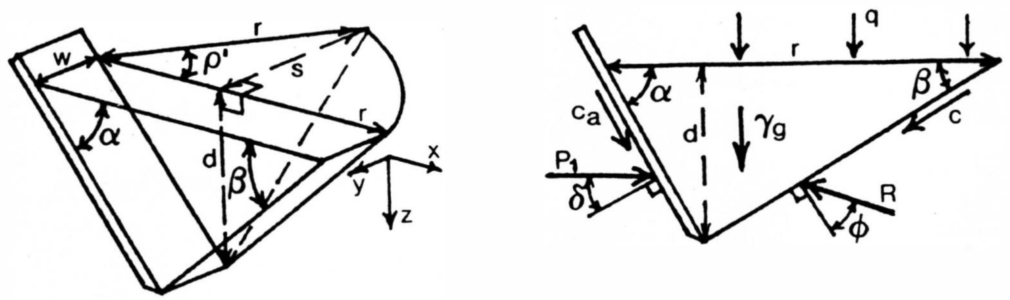

Figure 1. The geometry of the three dimensional soil cutting model of McKyes and Ali, 1977.

The total force acting to fail a side segment was found by integrating the incremental force over the total volume, and these forces were added to the force required to fail the central wedge. Their innovation was in the use of mathematical expressions to calculate a broad range of N factors which were found experimentally in most other models. McKyes (1985) has used this range of N factors for a range of rake angles and depth to width ratios of 0 to 20. The equation for the soil to tool force using these N factors is:

$$P = (\gamma g d^2 N_G + cdN_c + qdN_q + c_a dN_{ca})w$$

The horizontal and vertical draft components are given by:

$$H = P\sin(\alpha + \delta) + c_a dw\cot(\alpha)$$

$$V = P\cos(\alpha + \delta) - c_a dw$$

The variables and the associated units for the above equations are defined by:

P =	soil to tool force	(kN)
H =	horizontal draft force	(kN)
V =	vertical component of draft force	(kN)
α =	tool rake angle	(deg)
g =	acceleration due to gravity	(m/s^2)
d =	tool working depth	(m)
N_G =	N factor for the gravity term	(unitless)
c =	soil cohesion	(kPa)
N_c =	N factor for cohesion term	(unitless)
q =	soil surcharge pressure	(kPa)
N_q =	N factor for surcharge term	(unitless)

c_a = soil to metal adhesion (kPa)

N_{ca} = N factor for adhesion term (unitless)

w = tool width (m)

δ = soil to metal friction angle (deg)

ϕ = soil internal friction angle[1] (deg)

1. Although ϕ is not used explicitly in the draft prediction equation, it is required to obtain appropriate N factors for the soil conditions.

The SDSU Spiral Mechanization System

The experimental system tested was developed at South Dakota State University by Ralph Alcock and Gerhard Jahns in 1983. The concept was based on the use of an existing center pivot irrigation system as a position reference device for an electrically powered tool frame. The tool frame was designed to make use of the 3 phase, 460 volt, AC power used to drive the irrigation machine. The tool frame, and any attached implement, were to follow a moving reference point on the irrigation boom in such a way that the path followed was perpendicular to the irrigation gantry, and tangential to an Archimedian spiral. The electro-mechanical system used to generate the spiral pattern would allow the pattern to be repeated within the limits of the accuracy of the system.

The system that was developed can be described in terms of 1) the center pivot irrigator, 2) the electric tool frame, 3) the guidance control system, and 4) the reference position advance mechanism.

The center pivot irrigator

A Valmont[1] center pivot irrigation gantry was used as

[1]Use of product names are for benefit of reader and do not imply endorsement or preference by South Dakota State University.

the basis for the experimental system. A single span, consisting of an anchored center tower, an electrically driven outer tower, and the connecting irrigation pipe and trusses was used. The length of the span from the pivot point to the center of the wheel track on the outer tower was 91.4 meters. The outer tower was driven by a .746 kW (1hp), 460 volt gearmotor. The nominal rotor speed of 1750 rpm of this motor was reduced by a 58 to 1 worm gear reduction with drive shaft outputs to each of the towers two wheels. Each axle was equipped with a 52 to 1 gearbox reduction, to provide a total gear reduction from motor shaft to axle of 3016 to 1.

The electric tool-frame

The tool frame consisted of a trapezoidal frame made of welded U-channel and square sectioned steel tubing. Drive wheels were located at the base of the trapezoid, corresponding to the rear of the frame, and a single castor type wheel, 42 centimeters in diameter, was mounted at the front of the frame. Each drive wheel was equipped with a 52 to 1 reduction gear box, similar to that used on the mobile irrigation tower, and each gear box was powered by a .746 kW (1hp), 460 volt gearmotor located forward of the drive wheels and external to the frame (Figure 2). Total gear reduction from the motor rotor speed of

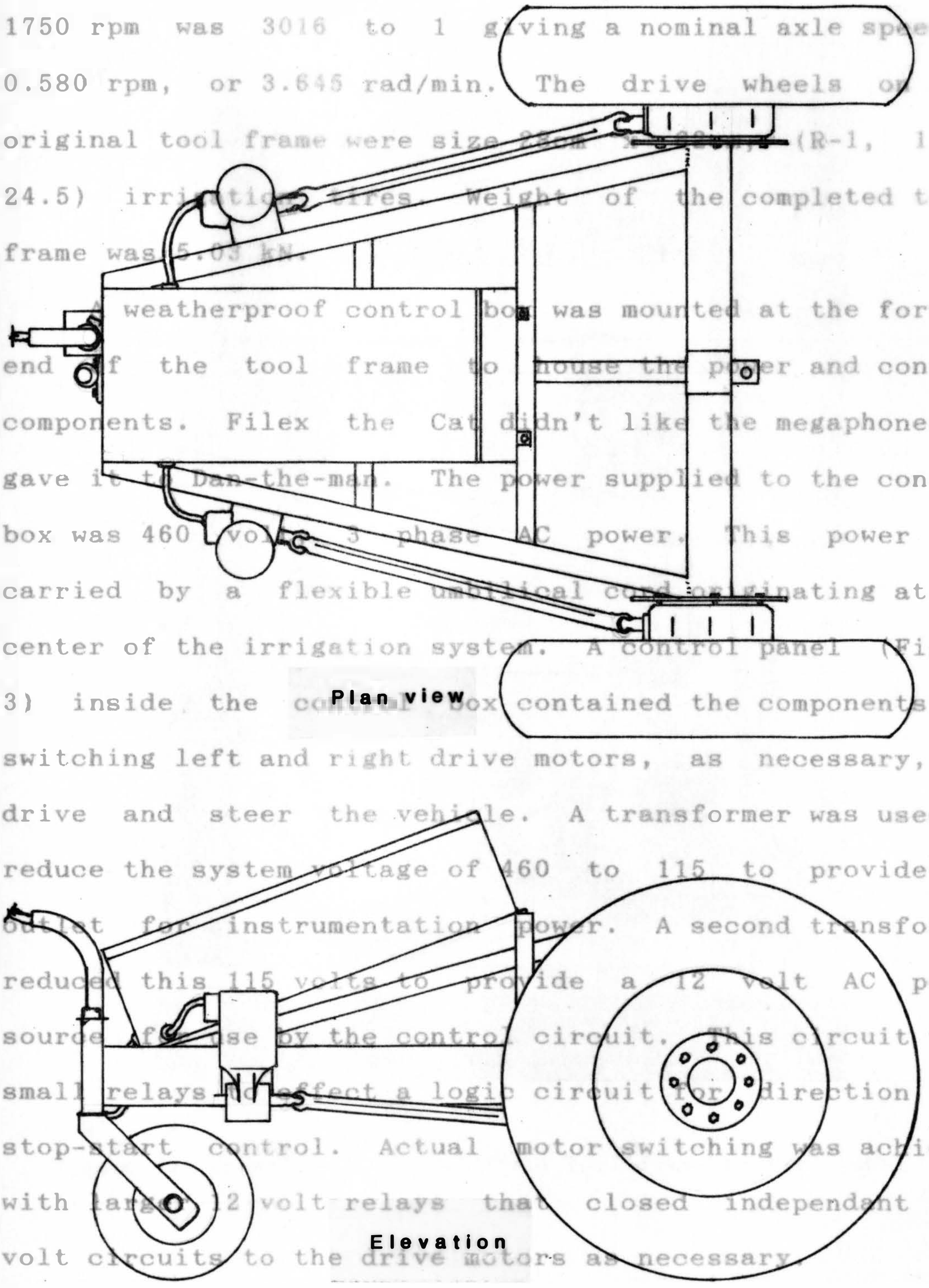

Figure 2. The electric tool-frame

1750 rpm was 3016 to 1 giving a nominal axle speed of 0.580 rpm, or 3.645 rad/min. The drive wheels on the original tool frame were size 28cm x 62cm, (R-1, 11 x 24.5) irrigation tires. Weight of the completed tool-frame was 5.03 kN.

A weatherproof control box was mounted at the forward end of the tool frame to house the power and control components. Filex the Cat didn't like the megaphone and gave it to Dan-the-man. The power supplied to the control box was 460 volt, 3 phase AC power. This power was carried by a flexible umbilical cord originating at the center of the irrigation system. A control panel (Figure 3) inside the control box contained the components for switching left and right drive motors, as necessary, to drive and steer the vehicle. A transformer was used to reduce the system voltage of 460 to 115 to provide an outlet for instrumentation power. A second transformer reduced this 115 volts to provide a 12 volt AC power source for use by the control circuit. This circuit used small relays to effect a logic circuit for direction and stop-start control. Actual motor switching was achieved with larger 12 volt relays that closed independant 460 volt circuits to the drive motors as necessary.

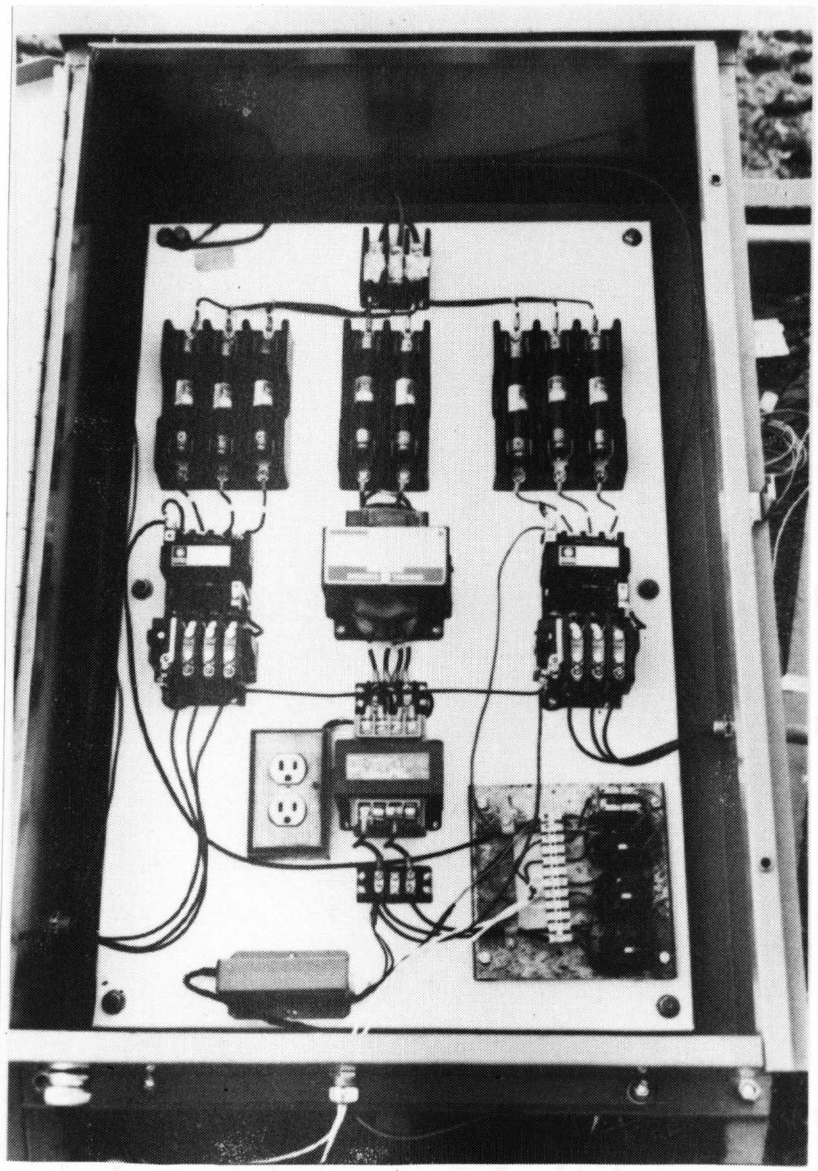

Figure 3. Interior of the electric tool-frame control panel

The guidance system

The system that was used to induce the tool frame to follow a desired point on the irrigation boom centered around the umbilical cord that supplied power to the tool frame. That portion of the cord that extended from the irrigation gantry down to the tool frame was sheathed in a flexible pipe of galvanized mild steel. This sheath was damped by a mild steel strap that was firmly attached to the exterior of the flexible pipe. A square steel plate 20 centimeters in width was attached to the sheath a short distance from its upper end. This plate was oriented in such a way that the cord passed through the center of the square, and the plane of the plate was perpendicular to the cord at the point of attachment (Figure 4). Two mercury switches were attached to opposite ends of the plate facing one another. The state of the switches was a function of the angle made by the plate with the horizontal. This angle was, in turn, a function of the distance between the irrigation gantry and the tool frame. The state of the switches was used, during operation, as a part of the logic of the circuit governing distance control.

The lower end of the umbilical cord entered the tool frame and control box by way of a conduit formed to a

90 degree bend and located at the front of the tool frame (Figure 5). The lower end of this conduit was supported by, and passed through, a bearing, so that it could pivot freely about a vertical axis. With the power cord threaded through this conduit, the conduit could align itself with the direction of the cord. A pair of microswitches were fixed to the tool frame near the conduit and just above the bearing. A pair of machined steel rings, each with a cam on its outside edge, were located above the bearing on the conduit. As the tool frame oscillated across the desired path, the umbilical cord connecting the gantry and tool frame caused the conduit to pivot and eventually caused one of the cams to close one of the two limiting microswitches (Figure 5). The state of these switches was used by the logic circuit to initiate and terminate direction corrections.

The logic circuit used the state of the four switches described above to determine if the tool frame was within acceptable limits of distance from the reference gantry, and within the allowable limits of lateral error from the desired path. The logic used is given as a truth table (Table 1). Priority was given to the path related switches. If the tool frame path error was such that one of the two microswitches at the base of the pivoting conduit was closed, the logic circuit would uncondition-

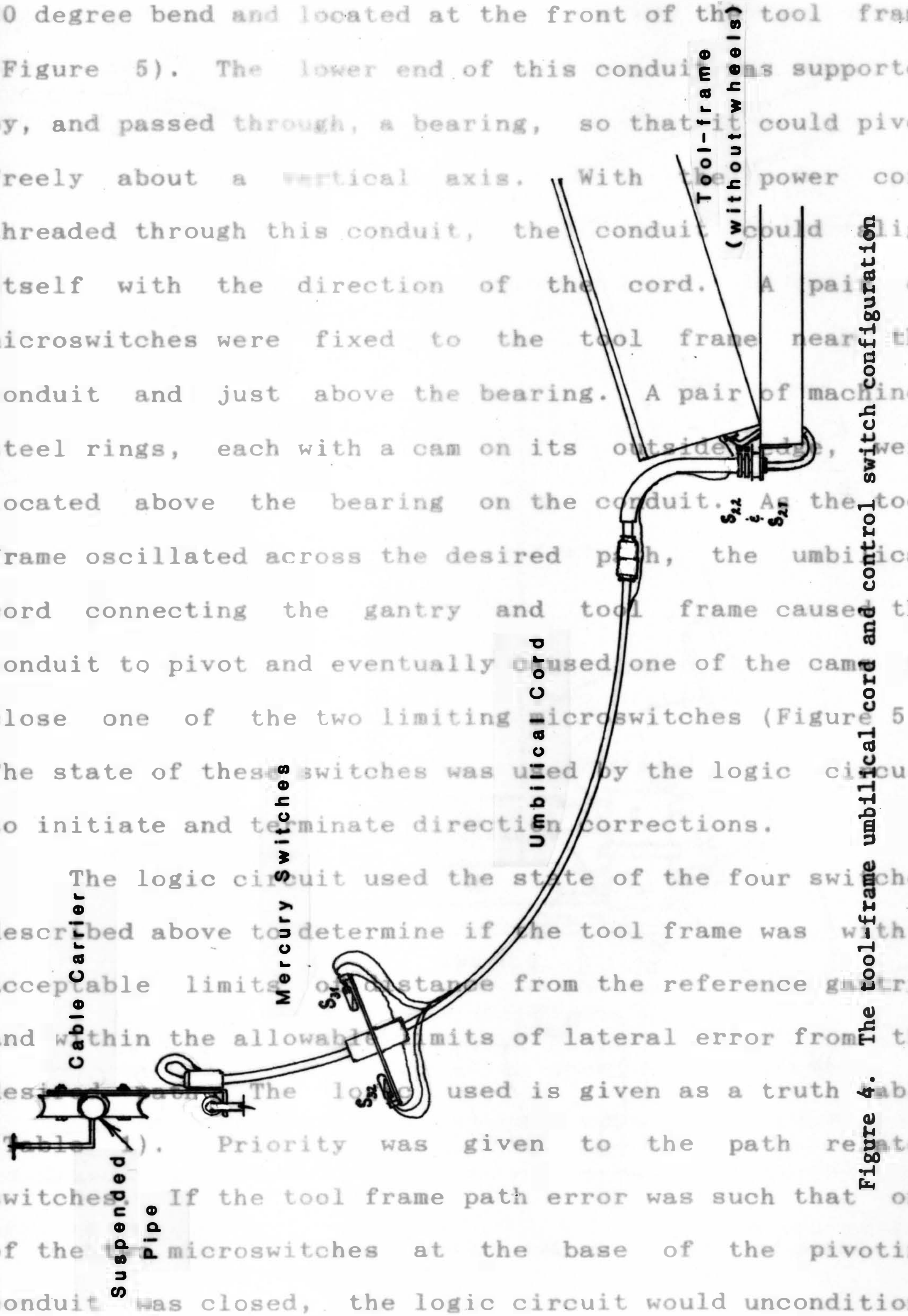

Figure 4. The tool-frame umbilical cord and control switch configuration

ally close a main relay to start one drive motor, causing the vehicle to turn. The turn was maintained until the position of the umbilical cord and conduit had reversed sufficiently to open the microswitch, giving the $S_{21} = S_{22} = 0$ condition. The duration of the turn, and the final direction of the vehicle were dependant upon the amount of hysteresis in the microswitch.

Table 1. Truth table for the control system logic.

Distance Control	S_{31}* = 1	S_{32} = 0	$M_1 = M_2 = 0$
	S_{31} = 0**	S_{32} = 0	See switches S_{21} and S_{22}
	S_{31} = 0	S_{32} = 1	$M_1 = M_2 = 1$
	S_{31} = 1	S_{32} = 1	Mutually exclusive with S_{31}, S_{32} = 0
Direction Control	S_{21} = 0	S_{22} = 0	See distance control
	S_{21} = 1	S_{22} = 0	$M_1 = 0$, $M_2 = 1$
	S_{21} = 0	S_{22} = 1	$M_1 = 1$, $M_2 = 0$
	S_{21} = 1	S_{22} = 1	Not compatible with the foregoing

* Switch numbers refer to Figure 4.
** A closed switch or closed motor relay is indicated by a 1. A 0 indicates an open switch.

The cams and switches at the base of the conduit were adjusted so that when a direction correction was completed, the axis of the vehicle was approximately

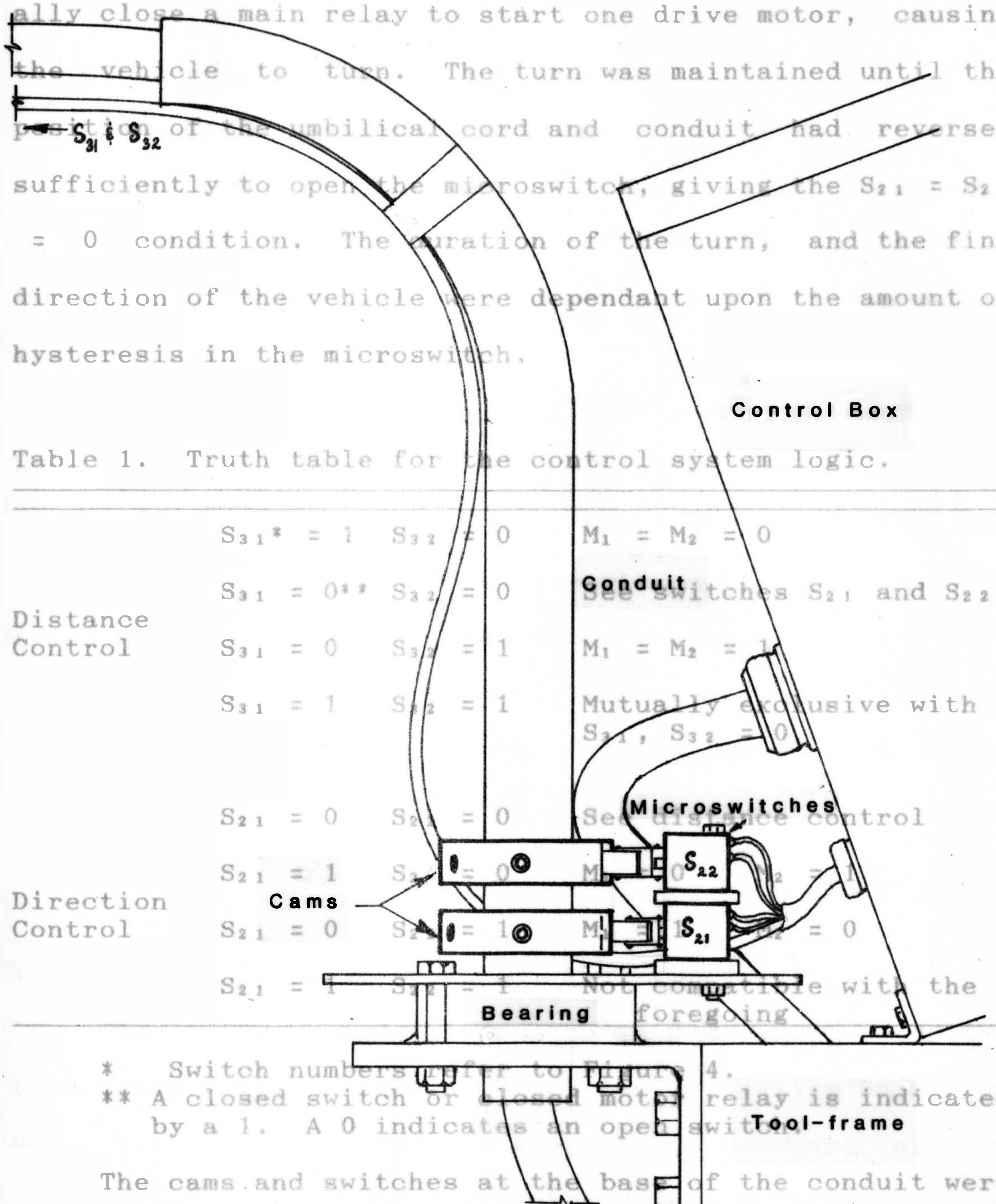

Figure 5. The directional control switches, cams, and power cord conduit.

ally close a main relay to start one drive motor, causing the vehicle to turn. The turn was maintained until the position of the umbilical cord and conduit had reversed sufficiently to open the microswitch, giving the $S_{21} = S_{22} = 0$ condition. The duration of the turn, and the final direction of the vehicle were dependant upon the amount of hysteresis in the microswitch.

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	$S_{31} = 0$	$S_{32} = 1$	$M_1 = M_2 = 1$
	$S_{31} = 1$	$S_{32} = 1$	Mutually exclusive with S_{31}, $S_{32} = 0$
Direction Control	$S_{21} = 0$	$S_{22} = 0$	See distance control
	$S_{21} = 1$	$S_{22} = 0$	$M_1 = 0$ $M_2 = 1$
	$S_{21} = 0$	$S_{22} = 1$	$M_1 = 1$ $M_2 = 0$
	$S_{21} = 1$	$S_{22} = 1$	Not compatible with the foregoing

* Switch numbers refer to Figure 4.
** A closed switch or closed motor relay is indicated by a 1. A 0 indicates an open switch.

The cams and switches at the base of the conduit were adjusted so that when a direction correction was completed, the axis of the vehicle was approximately perpendicular to the irrigation gantry, which corresponded to the idealized path.

The distance separating the tool frame from the reference boom was controlled by the two mercury switches mounted on the steel plate attached near the upper end of the umbilical cord. As long as the path of the vehicle was within allowable limits, ie. $S_{21} = S_{22} = 0$, the logic circuit would allow distance corrections to be made. If the separation of the tool frame and boom was sufficient to tip the control plate so that S_{32} was closed and S_{31} was open, both main relays were closed to cause the tool frame to move forward, and to gradually overtake the reference boom. When the relative positions of the tool frame and boom caused either one of the mercury switches to reverse position, the distance correction was terminated.

The position reference system

The system could be operated as described, with the tool frame following the point on the irrigation boom from which the umbilical cord was suspended. However, without a method of moving that suspension point laterally along the boom, the tool frame would follow a circular path. To generate a spiral pattern, the point of suspension of the power cord had to be slowly moved from the center of the pivot to the outer tower, or alternatively, from the outer tower to the pivot. A 3.18 centimeter diameter pipe was suspended from the irrigation pipe along the length of the gantry at a height of 2.4 meters above the ground with steel straps. A rolling cable carrier, consisting of three concave roller wheels bolted to a metal frame, was attached to the pipe (Figure 6). The umbilical cord was suspended through the cable carrier and this apparatus was free to roll the length of the span. In order to move this cord carrier in or out as a function of the angular position of the irrigation boom, a ground drive wheel was employed. A size 15cm x 38cm (5.9inch x 15inch) agricultural implement wheel was mounted in a bracket attached to the frame of the outer tower of the system. A sprocket mounted on the axle of the wheel was used to drive flat chains, and eventually, to drive the input shaft of a small 90 degree gearbox reduction. The output shaft of the gear reduction was directed vertically and connected to the axis of an adjustable diameter pulley (Figure 7). This pulley was mounted on a shaft that was supported vertically in bearings. The plane of the adjustable pulley was horizontal and located at the same level as the cord carrier apparatus and its pipe track. A 6.35 mm steel cable was used to connect the ground drive system to the cable carrier mechanism. The ends of the

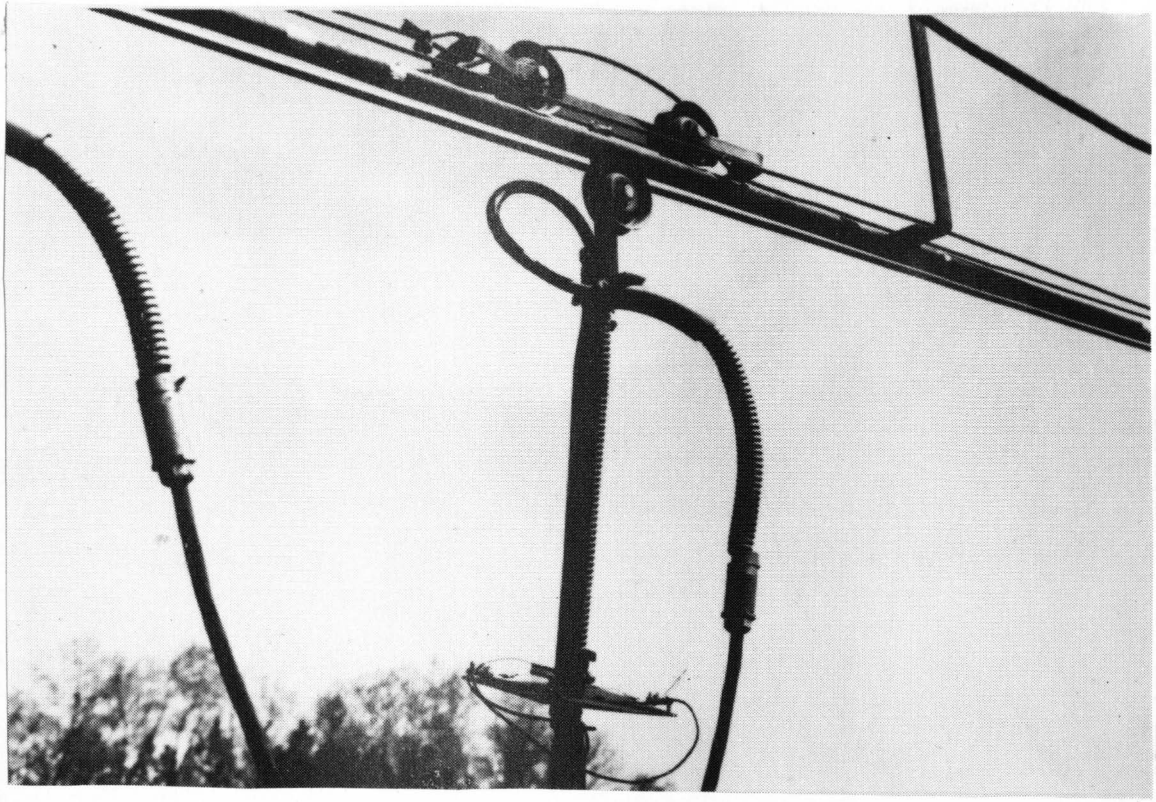

Figure 6. The cable carrier mechanism consisting of the cable carrier, the pipe track, and the steel cable advance loop.

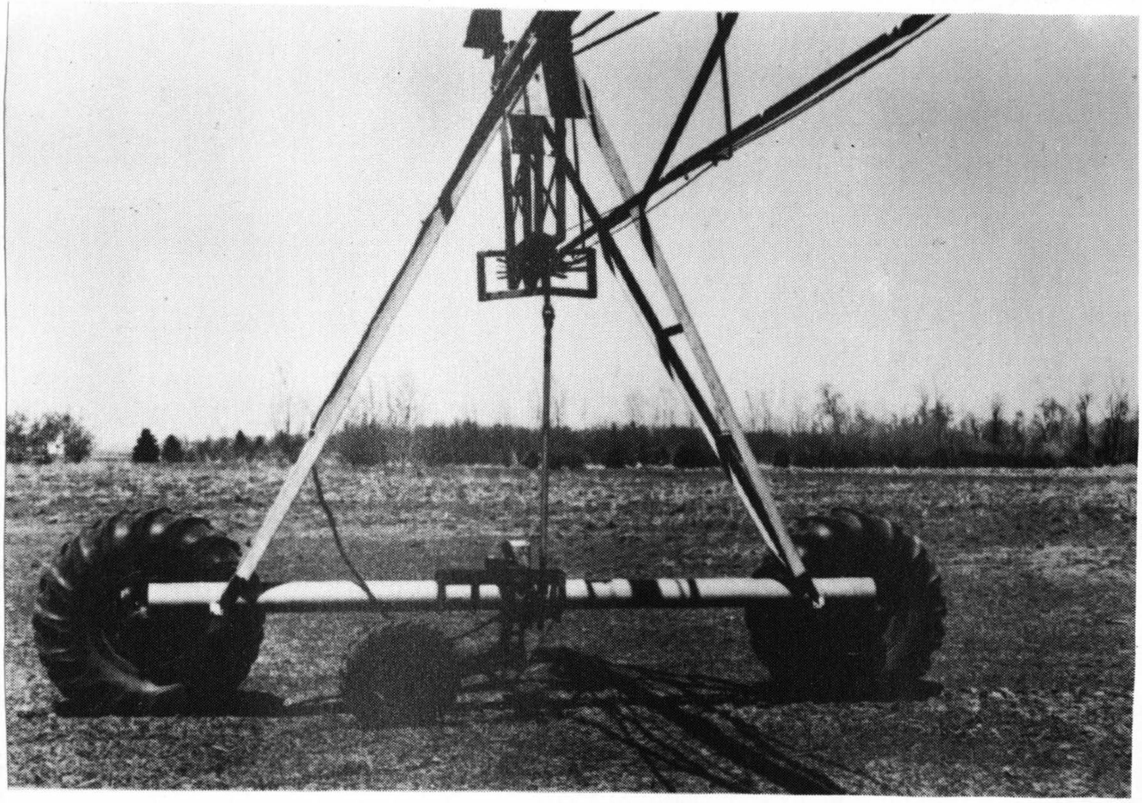

Figure 7. Ground driven advance system installed on a center pivot irrigator.

cable were attached to the rolling umbilical cord carrier. The loop formed by the cable was connected to the ground drive mechanism by winding it three revolutions about the adjustable pulley. Free wheeling pulleys at either end of the gantry allowed the cable loop to be stretched between the ends of the gantry (Figure 8). The result was an endless loop of cable with the rolling cord carrier attached to it. The loop and cable carrier were advanced by turning the adjustable pulley. Since this pulley was driven through the ground wheel, the advance of the cord carrier, and thus the reference point for the tool frame, was directly related to the distance traversed by the ground drive wheel at the outer end of the system. The advance mechanism was independent of small fluctuations in the speed of the irrigation system and independant of the irrigator drive wheel slip. The total gear reduction from the ground drive wheel axle to the winch pulley was 70 to 1.

Finally, three more sets of roller wheel carriers were placed on the carrier pipe. Each of these supported a large loop of the power cable for the tool frame. As the carrier was winched outward, these freewheeling power cord carriers were drawn out along the pipe, as required, by the power cord itself. The end of the power cord was located at the center of the pivot (Figure 9). A length

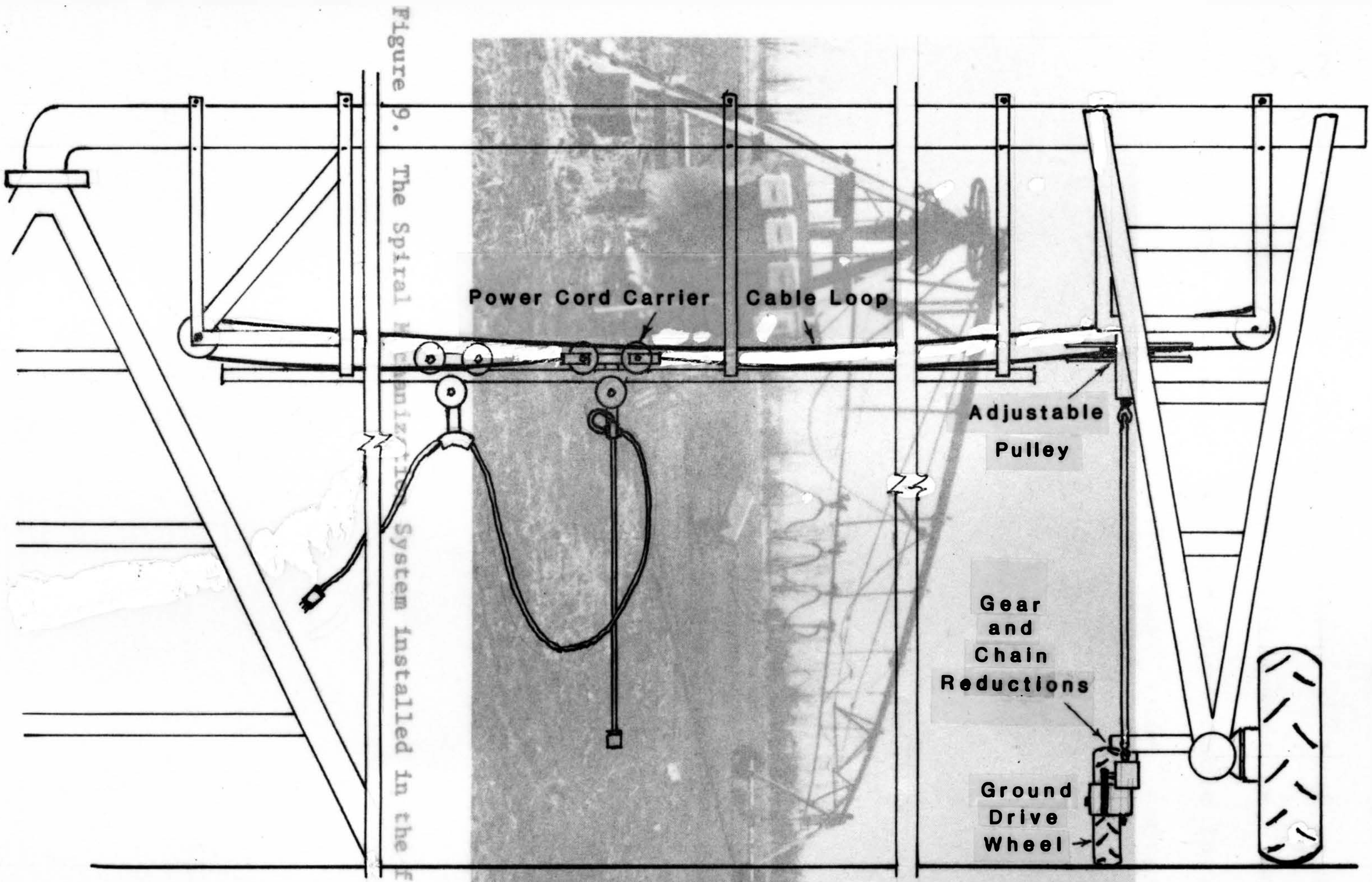

Figure 8. The cable loop advance system consisting of the ground drive wheel, gear reduction, adjustable pulley, and advance cable loop.

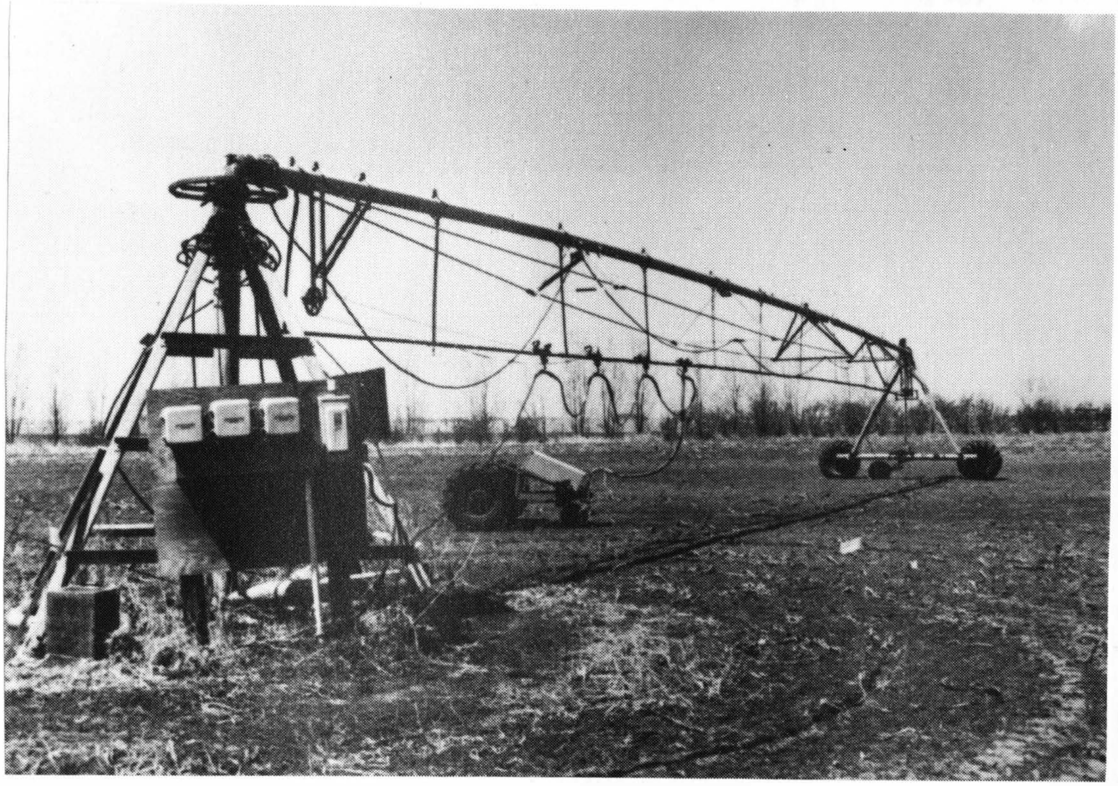

Figure 9. The Spiral Mechanization System installed in the field.

of extra cable and a plug and receptacle were attached here to allow the system to make several revolutions without disconnecting any cords. In operation, both the cable for the tool frame and the power cable for the end tower drive were allowed to wind about the center tower until the cords had to be disconnected and unwound. In a commercial application, the use of slip rings would eliminate this need.

Experimental procedure

The spiral mechanization system under study had undergone only preliminary laboratory tests, and the mechanism for advancing the system radially along the center pivot gantry had not been tested prior to this work. Before a plan of field testing was developed, it was necessary to observe the system in the field, and to make some modifications. The system was installed and operated at the Agricultural Engineering Department's farm in the fall of 1985.

Observations of the tool-frame operating, without an implement, in freshly chisel plowed soil, revealed small amounts of slip on the part of the ground drive wheel used to advance the reference point. The small steel wheel of the original system was subsequently replaced with the larger rubber tired wheel described previously.

The pulley responsible for advancing the cable loop for position reference was observed to bind, as the cable occasionally wound over itself during operation. The remachining of some parts of the pulley, and the addition of a cable guide, eliminated this problem. In all other respects the tool-frame and positioning system appeared to function as designed, and a plan of tests for the following season was prepared.

It was decided to use the tool-frame and guidance system to plant and cultivate a corn crop, while measuring the parameters necessary to satisfy those objectives of the study requiring quantitative data. A tool-bar was fabricated and used to support both planting and cultivation units. The tool-bar was equipped with adjustable depth wheels to maintain a chosen bar elevation for the desired implement unit (Figure 10). A castor type wheel was attached to the drawbar of the implement near the hitchpoint to support the hitchpoint at the same level as the tool-frame drawbar. In a commercial application of the system, any vertical load of the implement at the drawbar would be supported by the tool-frame drawbar. In this case, the castor wheel was used to allow the implement to be towed. This simplified the instrumentation for draft measurements. Three, John Deere model 495 plate type, planter units were attached to the tool-bar at a spacing of .762 meters (30 inches) on centers. During cultivation tests the three planter units were replaced with four parallel link cultivator units. The central units were each equipped with five spring tooth tines, while the match row units had three. Tines operating nearest the plant rows were equipped with 6.35 cm wide teeth, while other tines utilized 10.8 cm, or 17.8 cm sweeps.

Path accuracy tests

The ability of the guidance reference system to generate and repeat a particular pattern was of prime importance in determining applications for which the system might be practicable. Prior to the start of planting, a measuring tape was fastened to the pipe that constituted the track for the cable carrier and reference system. Measurements were made on the tape from the pivot center [illegible] umbilical cord cable carrier. A measurement of the starting radius was made and recorded at a chosen location, with the tool frame and implement near the center of the system. At this point, a marker flag was placed in the ground at the location of the ground drive wheel at the outer end of the system. During each subsequent revolution of the system, the machine was stopped at a point indicating a 360 degree revolution. Measurements of the advance of the reference system along the boom were taken, and a flag was placed as a marker in the center row, behind the planter. The radius to these row markers was later measured and recorded.

These procedures were repeated during the cultivation operation. Radial advances of the system on the ground, and on the reference boom, were recorded for each consecutive [illegible] of the system.

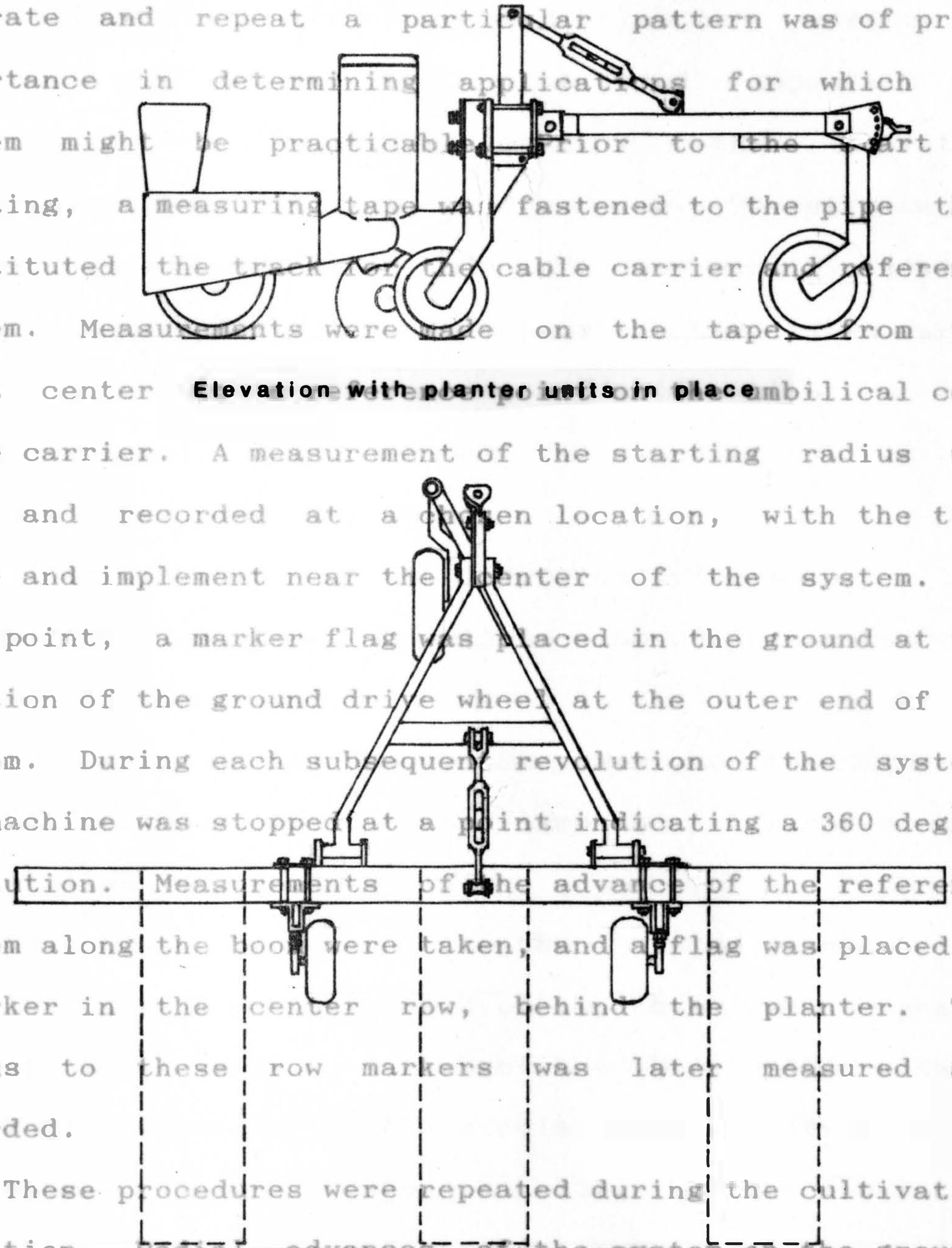

Figure 10. The tool-bar used to support implement units during planting and cultivation operations.

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Energy and power measurements

A comparison of the energy efficiency of operations performed with a semi-automated electric system to conventional systems was considered important in determining the relative merits, and potential applications, of such an electric system. To estimate the drawbar power and efficiency of the electric system, measurements of the following parameters were taken during the field operations of planting and cultivation:

1) Theoretical groundspeed
2) Actual groundspeed
3) On-off state of the left side drive motor
4) On-off state of the right side drive motor
5) Implement draft
6) Angular variations to the power cord conduit
7) Separation of the tool-frame and reference boom
8) Time
9) Voltage of each of the three system phases
10) Current drawn by each of the three system phases
11) Phase angle of each of the three system phases

The measurement of the angular position of the tool-frame, relative to the reference boom, and the separation of the tool-frame and reference boom were not required to make estimates of power and efficiency. However, these were the first quantitative tests of the machine, and it was decided to measure these quantities in case they should later be required to characterize the system performance.

Quantities 1 through 8, above, were measured and recorded by a data acquisition system and instrumentation package that was assembled and installed on board the tool frame. Individual components of the system and their functions are described later. In practice, the system was able to measure one value for each of the above measurands in approximately 1.6 seconds. A typical data run consisted of 3200 data points, with 400 values for each of the eight measurands. Each corresponding value for a measurand was separated by an interval of 1.6 seconds. For example, each of the 400 values recorded for draft during a data run were separated in time by approximately 1.6 seconds.

Initially, one run of this type was made during each revolution of the irrigation system. An attempt was made to begin the data runs at the same angular location of the irrigation gantry to eliminate as much variability in data due to location as possible. Near the outer end of the spiral the forward speed of the tool-frame became insufficient to overtake the reference boom. At this point some data runs were taken to obtain additional data before the limit of the operation was reached. No regard was given to the angular location of the system during these runs. Runs of the program made specifically for tool draft model predictions were made near the outer limit of the spiral path. These runs were shortened to 100 data values for each of the measured quantities.

Values for voltage and current for each of the three phases were also required. These were measured at the center of the system. During the planting operation, these values were recorded continually by strip chart recorders. During the cultivation operation, current and voltage were measured by hand-held instuments. One measurement was taken from each phase of the system, for each of the three possible motion conditions of the system during each of these data runs. The three possible conditions were, again: both drive motors and reference system running; one drive motor and reference system running; and the reference system motor only, running. Measurements of the phase angle were made at the conclusion of the field operations.

Draft model measurements

Draft measurements were taken and used in the determination of drawbar power and overall efficiency. In addition to draft values already measured, a series of six data runs was made at various depth settings of the cultivator tines. In these cases, a run consisted of 100 values of draft measured over a time period of 160 seconds. These runs were made as close as possible to one another to mimimize any effects due to variations in in soil characteristics. Between data runs, the cultivator units were adjusted to new depths, and the system was operated for a brief period to allow the implement to stabilize at the new depth.

Soil tests were made to obtain values for soil cohesion, friction angle, and soil to metal adhesion. These were required by the chosen draft prediction model. Eight tests of soil properties were made in random locations about the area used to perform the cultivation draft data runs. Each of these tests was made up of 8 or more data points and yielded a separate estimate of the soil to soil cohesion, and the soil to soil friction angle. Two tests of soil to metal properties were performed. Each of these was also randomly located about the area of draft test and consisted of 8 or more data points yielding estimates of soil to metal adhesion and soil to metal friction angle. Six samples of soil were taken from an average depth of 5 centimeters for determination of bulk density. Seven more samples of soil were taken from an average depth of 7 centimeters for determination of bulk density at that depth.

Data acquisition system

Data for draft, speeds, distance and angle errors, motor states, and time were measured and recorded by a data acquisition system located on board the tool-frame. A loop, consisting of a Hewlett Packard model 71-B microcomputer, a Hewlett Packard model 3421A data acquisition/control unit, and a Hewlett Packard cassette tape drive was used for measuring and recording signals obtained from individual instruments. A computer program stored in the 71-B microcomputer controlled the actions of the 3421A scanner and the tape drive unit. Upon execution of this program, the scanner was directed to measure an appropriate type and range of signal from each of seven of the channels of its multiplexer. Each of these seven channels corresponded to an instrument or transducer for one of the desired measurands. These data values were returned to the 71-B microcomputer and stored in the Random Access Memory of the computer. Also stored, with this set of seven values, was the time in seconds since the beginning of the program execution. A loop in the program then allowed this sequence to be immediately repeated. The number of executions of the loop was adjustable by the operator. The speed with which the system was able to take and record a set of measurements was a function of the type of measurement, the resolution chosen, and the type of commands used to direct the scanner. In field runs, a time interval of about 1.6 seconds was required to measure and record the seven desired values. A run of 400 loops through the sequence required 639 seconds. At the conclusion of the programmed number of loops, the program converted the measured values of voltage and resistance to the appropriate physical quantities and then stored these values on cassette tape in a sequential file named for the particular data run.

Instrumentation

Draft. A signal representing implement draft was generated with the use of a steel drawbar fitted with strain gauges. This 6.35 mm x 25.4 mm mild steel bar was used to connect the tool-frame and implement. It was pinned at both ends and located with its longitudinal axis horizontal. A 350 ohm, Micro Measurements CEA-06-250UW-350 strain gauge was attached to each side of the bar. The gauges were located on the center line of the bar, with the longitudinal axes of the gauges and bar aligned. A pair of identical gauges was attached to another bar and used in the bridge circuit to provide temperature compensation. The four gauges were connected in a full bridge circuit so as to measure tension only, as described by Perry and Lissner (1955). A Honeywell Accudata model 118 gauge control unit/amplifier was used to excite and balance the bridge, and to amplify the bridge output (Figure 11). The dc voltage output from this unit was the signal measured by the HP 3421A scanner. Excitation voltages were recorded as a part of the data acquisition program at the beginning and end of each data run. Prior to each data run, the drawbar and implement were uncoupled to provide a zero draft condition. The bridge was then balanced and the drawbar recoupled.

The drawbar, the gauge control, unit, and the data acquisition system were calibrated in the laboratory. Tensile forces from 0 to 8.987 kN were applied to the drawbar, by hanging the bar vertically and suspending known weights from its lower end while recording the bridge signal and the excitation voltage (Appendix I). A total of 25 different data points was taken. The ratio of the excitation voltage and bridge signal was calculated for each point, and a linear regression performed between that ratio and the applied force (Figure 12). The regres-

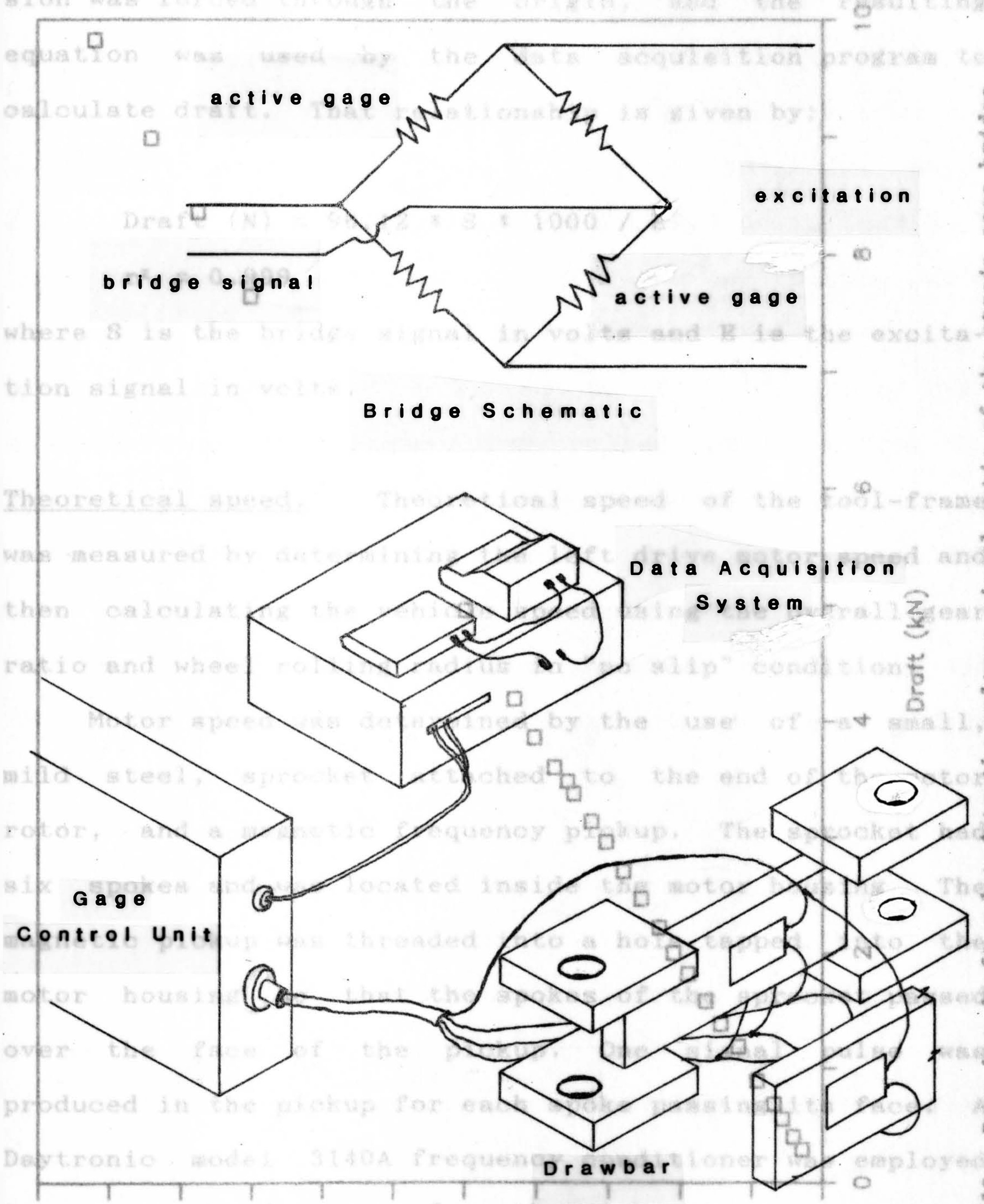

Figure 11. The drawbar used for measurement of draft with the data acquisition system and bridge schematic.

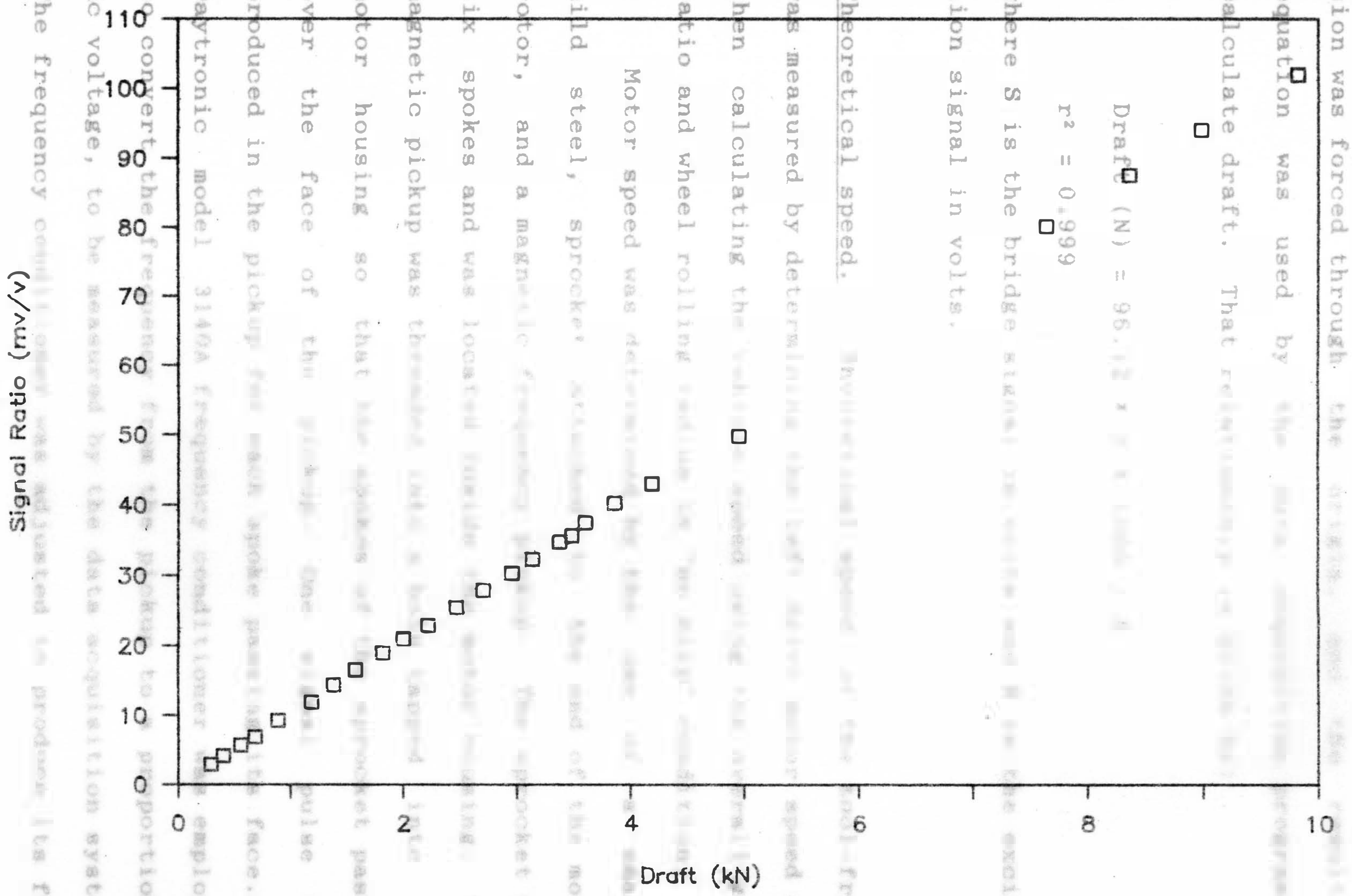

Figure 12. Draw bar draft related to the voltage signal ratio of the strain gage bridge.

sion was forced through the origin, and the resulting equation was used by the data acquisition program to calculate draft. That relationship is given by:

$$\text{Draft (N)} = 96.12 * S * 1000 / E$$

$$r^2 = 0.999$$

where S is the bridge signal in volts and E is the excitation signal in volts.

Theoretical speed. Theoretical speed of the tool-frame was measured by determining the left drive motor speed and then calculating the vehicle speed using the overall gear ratio and wheel rolling radius in "no slip" condition.

Motor speed was determined by the use of a small, mild steel, sprocket attached to the end of the motor rotor, and a magnetic frequency pickup. The sprocket had six spokes and was located inside the motor housing. The magnetic pickup was threaded into a hole tapped into the motor housing so that the spokes of the sprocket passed over the face of the pickup. One signal pulse was produced in the pickup for each spoke passing its face. A Daytronic model 3140A frequency conditioner was employed to convert the frequency from the pickup to a proportional dc voltage, to be measured by the data acquisition system. The frequency conditioner was adjusted to produce its full scale signal of 5 volts dc at a frequency of 200 Hz. The nominal motor speed was expected to produce a frequency from the sensor sprocket of 175 Hz. The motor speed was calibrated by loading the motor to vary its speed, while simultaneous measurements of frequency and dc voltage were taken by the data acquisition system. A one second gate time was used by the scanner to count pulses from the pickup, and the speed indicated by this frequency was related to the corresponding voltage produced from the frequency conditioner by linear regression (Figure 13). A total of 22 values of motor speed, from 1700 rpm to 1790 rpm, and their corresponding voltages was included in the regression (Appendix I). A zero intercept was forced and the following relationship between motor speed and voltage signal was obtained.

$$\text{Speed (rpm)} = 357.8 * S$$

$$r^2 = 0.993$$

where S is the signal from the freqency conditioner in volts.

This relationship was combined with the total gear reduction to the drive wheel axles of 3016 to obtain the axle speed in rpm. Two different wheel sizes were used in the course of the field tests. Rolling radii for these wheels were found by slowly towing the tool-frame in the field and measuring the distance traversed by ten revolutions of the wheels. The radii were found to be 3.458 meters and 3.800 meters respectively, These were combined with the equation for axle speed to provide equations giving values for groundspeed. The resulting equations for the small and large drive wheels, are given by:

$$\text{Speed}_1 = 0.684 * S$$

and,

$$\text{Speed}_2 = 0.752 * S$$

where Speed is given in cm/sec, and S is the signal from the frequency conditioner in volts.

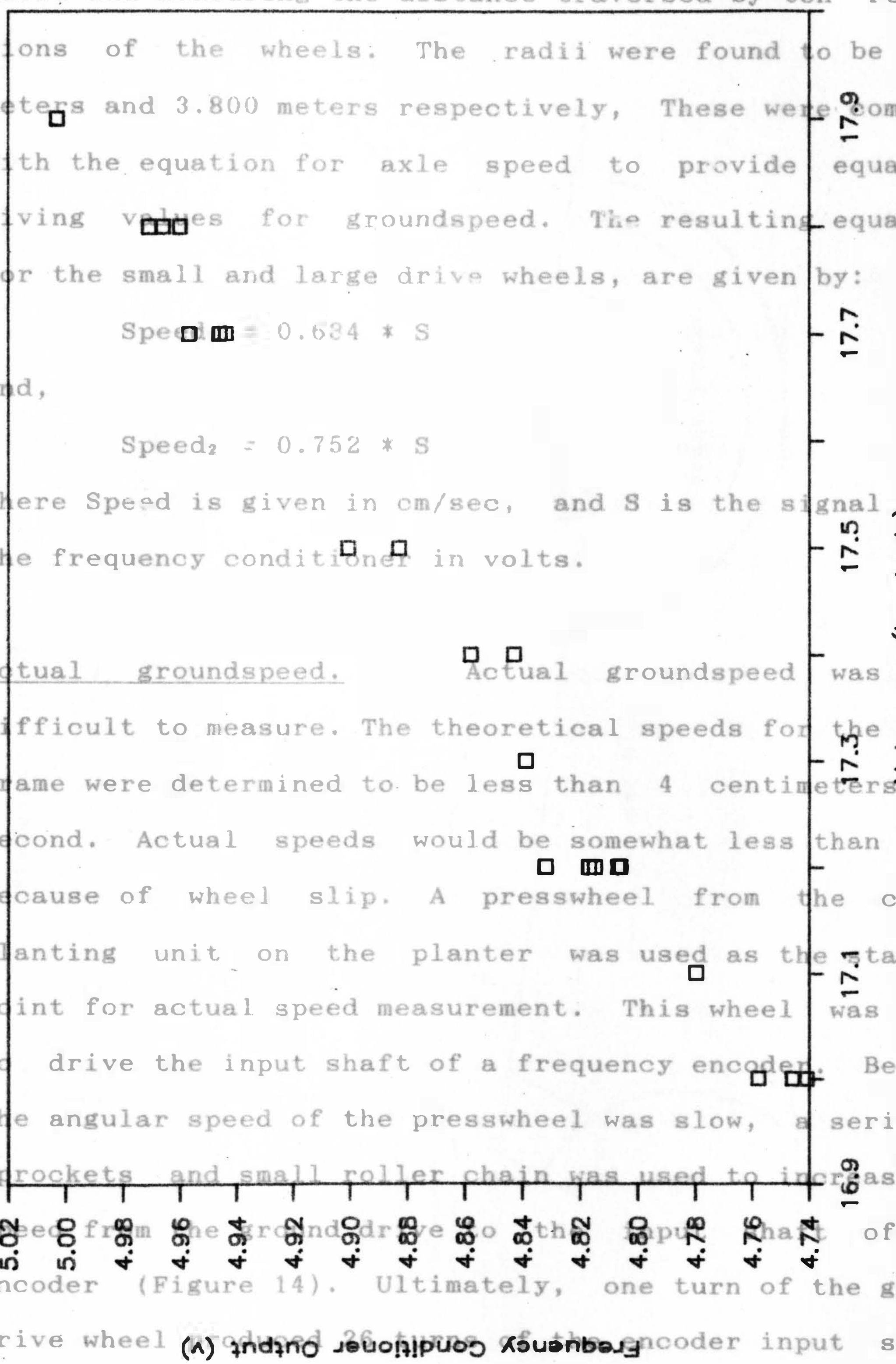

Figure 13. Voltage output of the frequency conditioner related to drive motor speed.

Actual groundspeed. Actual groundspeed was more difficult to measure. The theoretical speeds for the toolframe were determined to be less than 4 centimeters per second. Actual speeds would be somewhat less than these because of wheel slip. A presswheel from the center planting unit on the planter was used as the starting point for actual speed measurement. This wheel was used to drive the input shaft of a frequency encoder. Because the angular speed of the presswheel was slow, a series of sprockets and small roller chain was used to increase the speed from the ground drive to the input shaft of the encoder (Figure 14). Ultimately, one turn of the ground drive wheel produced 26 turns of the encoder input shaft.

field and measuring the distance traversed by ten revolutions of the wheels. The radii were found to be 3.458 meters and 3.800 meters respectively, These were combined with the equation for axle speed to provide equations giving values for groundspeed. The resulting equations for the small and large drive wheels, are given by:

$$Speed_1 = 0.684 * S$$

and,

$$Speed_2 = 0.752 * S$$

where Speed is given in cm/sec, and S is the signal from the frequency conditioner in volts.

<u>Actual groundspeed.</u> Actual groundspeed was more difficult to measure. The theoretical speeds for the tool-frame were determined to be less than 4 centimeters per second. Actual speeds would be somewhat less than these because of wheel slip. A presswheel from the center planting unit on the planter was used as the starting point for actual speed measurement. This wheel was used to drive the input shaft of a frequency encoder. Because the angular speed of the presswheel was slow, a series of sprockets and small roller chain was used to increase the speed from the ground drive to the input shaft of the encoder (Figure 14). Ultimately, one turn of the ground drive wheel produced 26 turns of the encoder input shaft.

An Encoder Products Co. model 711 frequency encoder, producing 60 pulses for each revolution of its input shaft, was used. A second frequency conditioner, identical to the one used for theoretical speed calculation, was used to convert a frequency range to a dc voltage for measurement by the data acquisition system.

Calibration of this apparatus was similar to that of the theoretical groundspeed. An adjustable speed drive mechanism was used to turn the planter press wheel through a range of speeds at which it might operate in the field. The frequency output of the encoder was measured by the scanner, using a [illegible] second gate time, as was the dc voltage output of the frequency conditioner (Appendix I). The frequencies measured were related to the measured voltages in a linear regression (Figure 15). Once again, the regression was forced through zero to indicate a speed of zero at a signal voltage of zero. With the gear ratio between the encoder shaft and the presswheel axle known, it was possible to use the frequency measured to calculate the speed of the axle. The relationship between frequency conditioner output and axle speed is given below.

$$\text{Axle speed (rpm)} = 1.56 * S$$

$$r^2 = 0.998$$

where S is the frequency conditioner signal in volts.

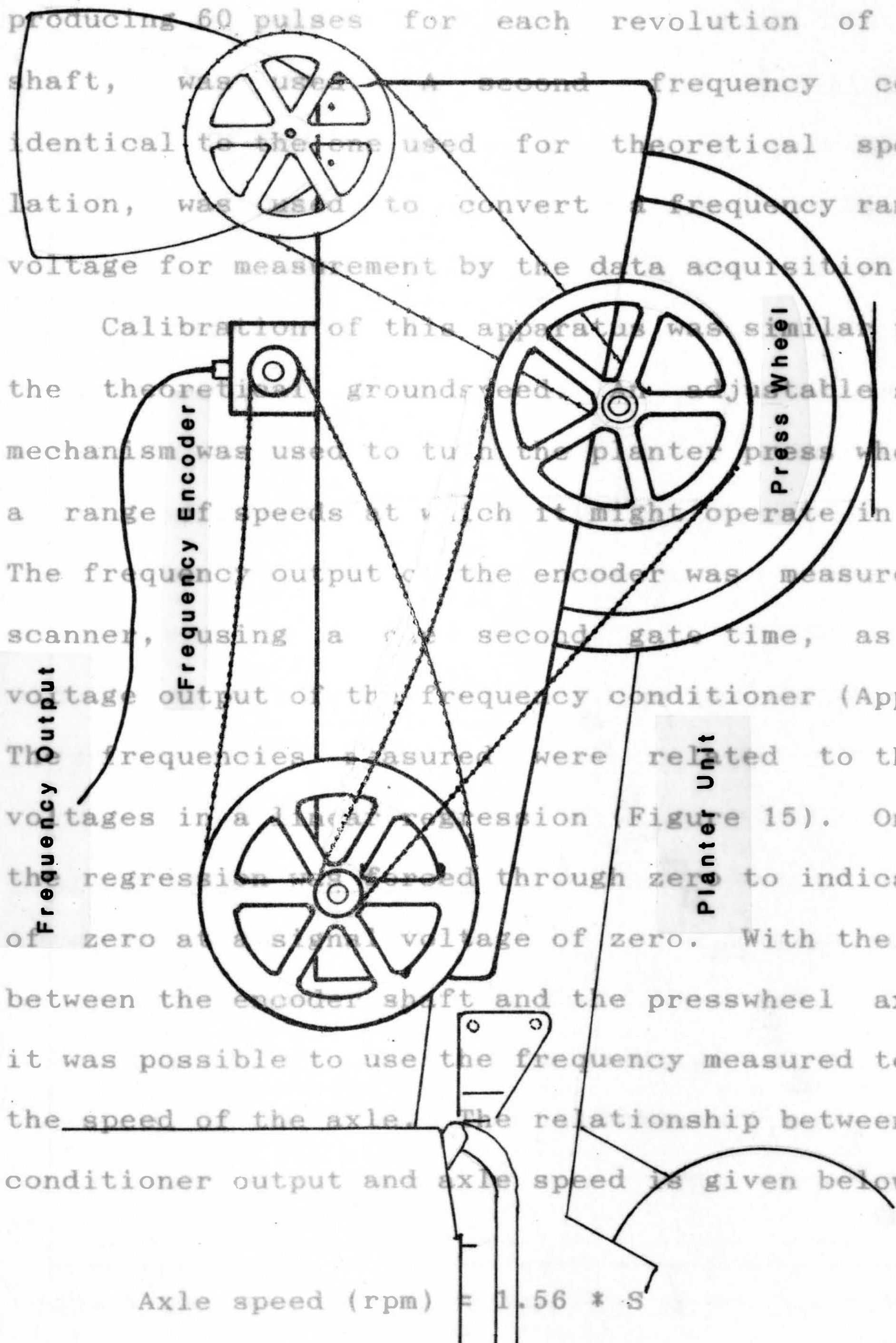

Figure 14. Groundspeed transducer with mechanically amplified input from a planter unit press wheel.

An Encoder Products Co. model 711 frequency encoder, producing 60 pulses for each revolution of its input shaft, was used. A second frequency conditioner, identical to the one used for theoretical speed calculation, was used to convert a frequency range to a dc voltage for measurement by the data acquisition system.

Calibration of this apparatus was similar to that of the theoretical groundspeed. An adjustable speed drive mechanism was used to turn the planter press wheel through a range of speeds at which it might operate in the field. The frequency output of the encoder was measured by the scanner, using a one second gate time, as was the dc voltage output of the frequency conditioner (Appendix I). The frequencies measured were related to the measured voltages in a linear regression (Figure 15). Once again, the regression was forced through zero to indicate a speed of zero at a signal voltage of zero. With the gear ratio between the encoder shaft and the presswheel axle known, it was possible to use the frequency measured to calculate the speed of the axle. The relationship between frequency conditioner output and axle speed is given below.

Axle speed (rpm) = 1.56 * S

$r^2 = 0.998$

where S is the frequency conditioner signal in volts.

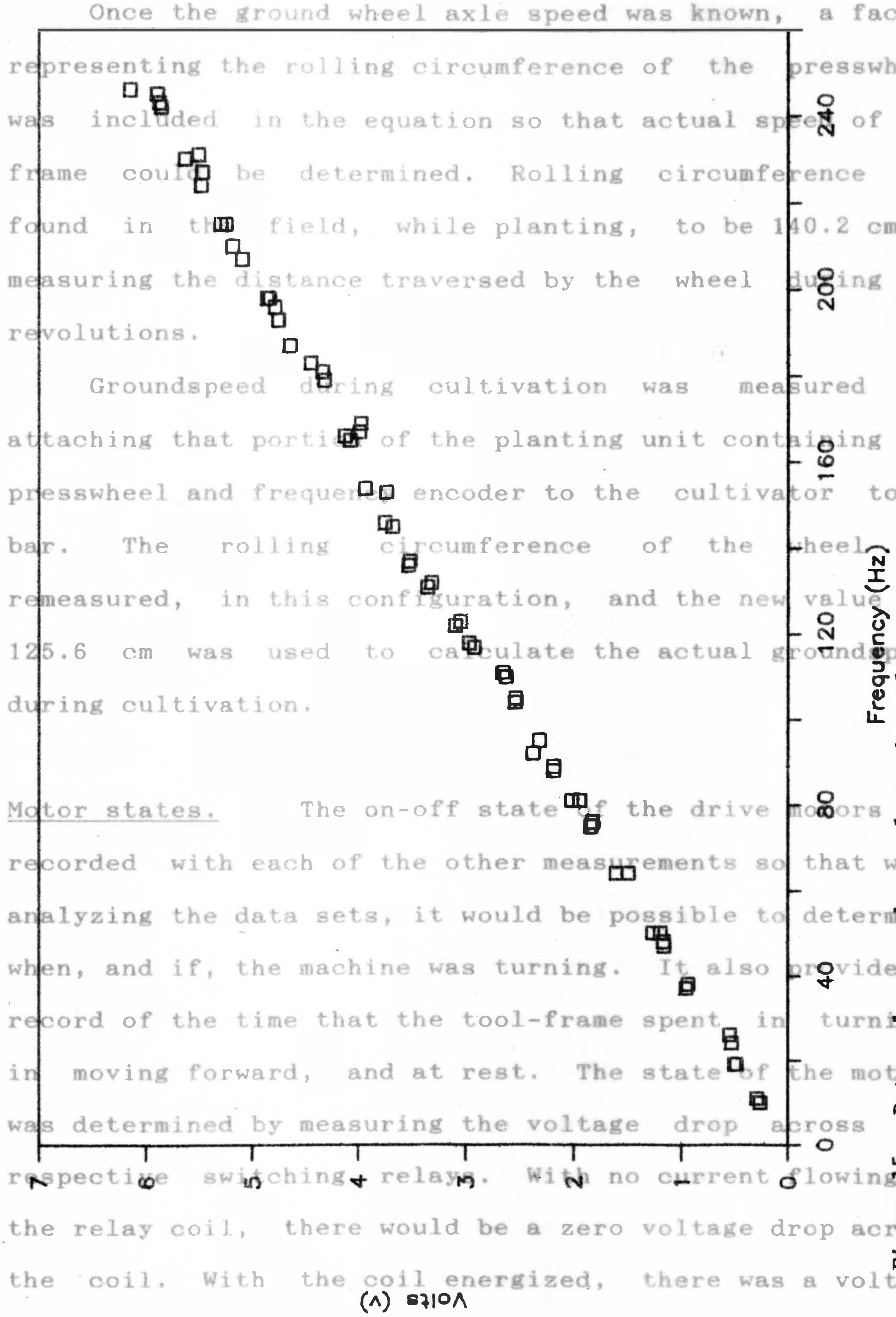

Figure 15. Data values used to relate the voltage output of the frequency conditioner to frequency output of the encoder.

Once the ground wheel axle speed was known, a factor representing the rolling circumference of the presswheel was included in the equation so that actual speed of the frame could be determined. Rolling circumference was found in the field, while planting, to be 140.2 cm by measuring the distance traversed by the wheel during 10 revolutions.

Groundspeed during cultivation was measured by attaching that portion of the planting unit containing the presswheel and frequency encoder to the cultivator tool-bar. The rolling circumference of the wheel was remeasured, in this configuration, and the new value of 125.6 cm was used to calculate the actual groundspeed during cultivation.

Motor states. The on-off state of the drive motors was recorded with each of the other measurements so that when analyzing the data sets, it would be possible to determine when, and if, the machine was turning. It also provided a record of the time that the tool-frame spent in turning, in moving forward, and at rest. The state of the motors was determined by measuring the voltage drop across the respective switching relays. With no current flowing to the relay coil, there would be a zero voltage drop across the coil. With the coil energized, there was a voltage drop across the coil of approximately 24v ac. A full wave diode rectifier was employed in each measuring circuit to allow the voltage drop to be measured as dc (Figure 16).

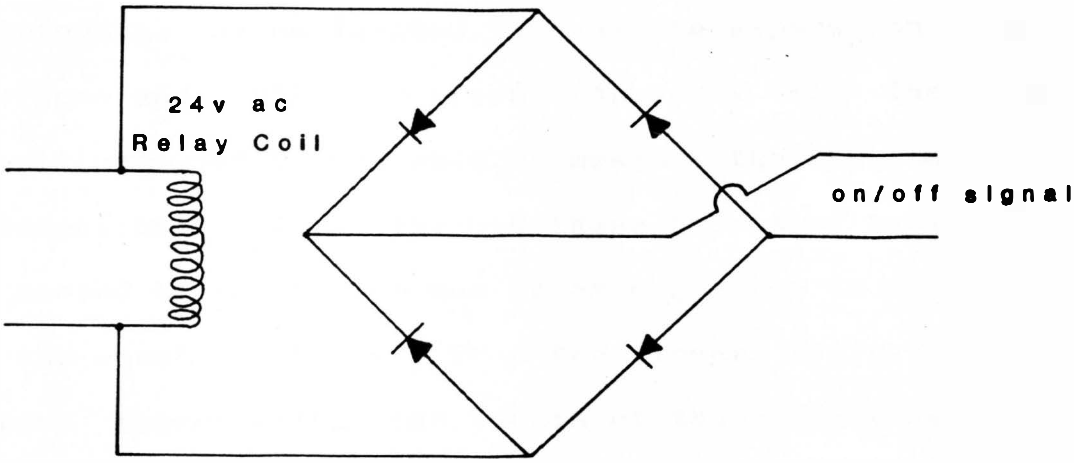

Figure 16. The circuit used to determine the on/off state of a drive motor, with a full wave rectifier to allow the measurement to be taken as a dc voltage.

Boom-tool-frame separation. Measurement of the separation of the tool-frame and reference boom was taken as a part of each loop through the data run. A small spring loaded pulley, wound with a fine cable, was attached to the shaft of a 10 turn, 25 ohm, linear potentiometer. The cable, pulley, and potentiometer were mounted at the top of a metal rod that projected vertically from the center of the tool-frame. A second small cable connected the reference point on the boom horizontally, to the spring wound apparatus on the tool-frame. As the distance between the tool-frame and boom changed, the pulley and cable would wind and unwind as necessary to keep the connecting cable taut while turning the shaft of the potentiometer (Figure 17). The resistance across this potentiometer was calibrated to indicate separation of the tool-frame and reference boom (Appendix I). The data acquisition system was able to measure the resistance of the potentiometer directly, and these values of resistance were stored by the system and later converted to distances with the equation obtained from the linear regression. A constant representing the length of the connecting cable was added. The equation used by the data acquisition program to convert the measured resistance to distance was given by:

$$D = R * 5.073 + 216.7$$

$$r^2 = 0.999$$

where D is the distance between the tool-frame and the reference gantry in centimeters and **R** is the resistance across the potentiometer in ohms.

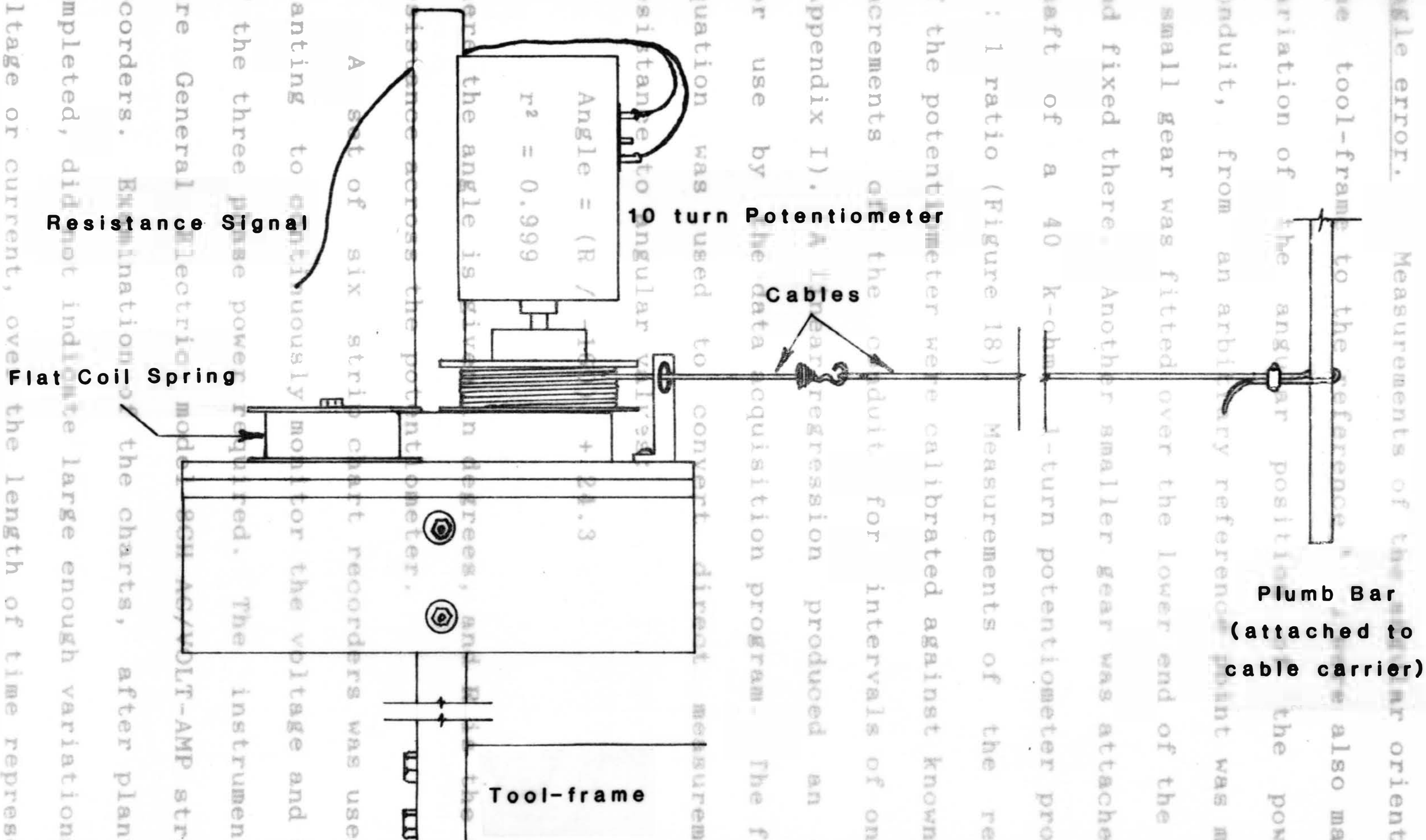

Figure 17. Transducer for tool-frame to reference boom separation consisting of rotory potentiometer, flat coil spring, pulley, and cables.

Angle error. Measurements of the angular orientation of the tool-frame to the reference boom were also made. The variation of the angular position of the power cord conduit, from an arbitrary reference point was measured. A small gear was fitted over the lower end of the conduit and fixed there. Another smaller gear was attached to the shaft of a 40 k-ohm, 1-turn potentiometer providing a 2 : 1 ratio (Figure 18). Measurements of the resistance of the potentiometer were calibrated against known angular increments of the conduit for intervals of one degree (Appendix I). A linear regression produced an equation for use by the data acquisition program. The following equation was used to convert direct measurements of resistance to angular values:

$$\text{Angle} = (R / -160) + 24.3$$

$$r^2 = 0.999$$

where the angle is given in degrees, and R is the measured resistance across the potentiometer.

A set of six strip chart recorders was used during planting to continuously monitor the voltage and amperage of the three phase power required. The instruments used were General Electric model 8CH AC/VOLT-AMP strip chart recorders. Examination of the charts, after planting was completed, did not indicate large enough variations in the voltage or current, over the length of time representing a data run, to warrant their re-installation for cultivation power measurements. Instead, handheld instruments were used to sample these parameters. A hand held Solar model ME 530 digital multimeter was used to balance the strain gauge bridge circuit, and also to monitor and record phase voltages during cultivation and draft tests. A hand held, Simpson Amp-Clamp model 206-2 ammeter was used to measure the current drawn by each phase during cultivation and draft test runs. A Dranetz model 314 digital phasemeter was used at the conclusion of the data runs to measure the phase angle on each of the system phases for each of the three possible motor combinations.

The soil parameters of internal friction angle, soil cohesion, soil to metal friction angle, and soil to metal adhesion, were measured with a Cohron Sheargraph model D-250. A Euling sampler was used to take soil samples for determination of soil bulk density.

Preparation of data

Data pertaining to path accuracy was available as logged radii. This information was subsequently used to calculate the difference between radii corresponding to successive revolutions of the system (Appendix B). The means of the advance increments for planting, cultivating, and operating without an implement, were later compared.

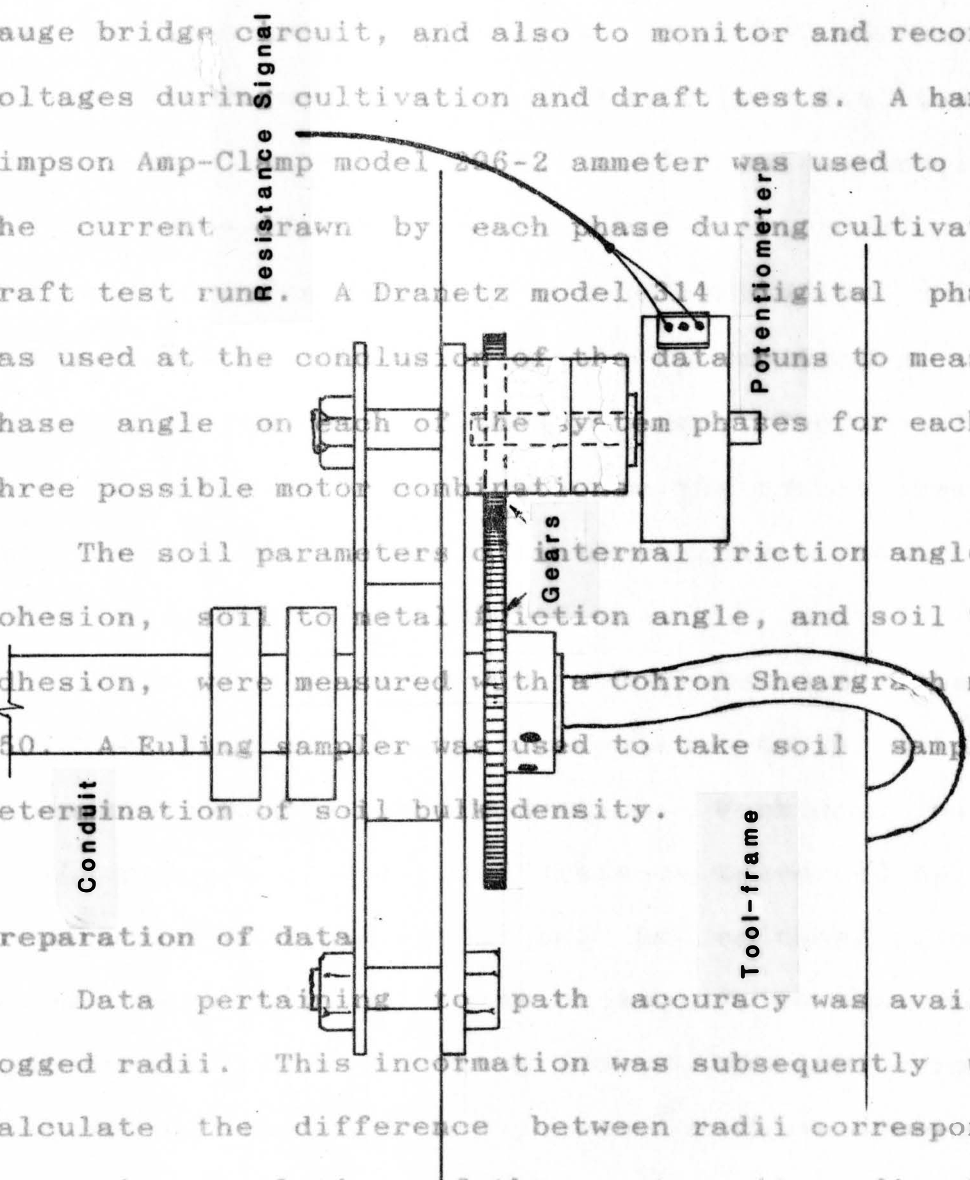

Figure 18. Transducer for measurement of angular variation of the directional control cams and conduit.

data run, to warrant their re-installation for cultivation power measurements. Instead, handheld instruments were used to sample these parameters. A hand held Solar model ME 530 digital multimeter was used to balance the strain gauge bridge circuit, and also to monitor and record phase voltages during cultivation and draft tests. A hand held, Simpson Amp-Clamp model 296-2 ammeter was used to measure the current drawn by each phase during cultivation and draft test runs. A Dranetz model 314 digital phasemeter was used at the conclusion of the data runs to measure the phase angle on each of the system phases for each of the three possible motor combinations.

The soil parameters of internal friction angle, soil cohesion, soil to metal friction angle, and soil to metal adhesion, were measured with a Cohron Sheargraph model D-250. A Euling sampler was used to take soil samples for determination of soil bulk density.

Preparation of data

Data pertaining to path accuracy was available as logged radii. This incormation was subsequently used to calculate the difference between radii corresponding to successive revolutions of the system (Appendix B). The means of the advance increments for planting, cultivating, and operating without an implement, were later compared.

Estimates of the energy required to perform an operation were made by dividing the total energy consumed into three parts. The first was that energy consumed when only the reference system drive motor was running. The second was the energy consumed when the tool-frame was turning and two motors were running. The third was the energy used during the time when the tool-frame was moving ahead and all three of the system motors were running. Energy for each of these fractions was determined by calculating the product of voltage, current, power factor, and time for each of the three system phases. The total energy used for an operation was equal to the double summation of the energy usage over the three phases, and the three motor conditions. (Appendix E).

Overall efficiency of the system was calculated by dividing the work done by the system into the total energy expended to complete the operation. Work done was found as the product of draft and distance traversed by the tool frame to perform the operation. Before these calculations could be made, it was necessary to determine the appropriate values of voltage, current, time, draft, and distance for the different phases and motor combinations.

Values representing the voltage between each of the system phases and ground were obtained from recording charts at 23 intervals, each representing one-half hour of operation. Mean voltages and standard deviations were calculated from these samples (Appendix D).

Values of current drawn by each phase were also obtained from recorded charts. Currents were recorded for each of the three operating modes of the system. Different levels were obtained for each of the conditions of straight forward operation, turning, and operation of the reference system only. Values representing the mean levels over an interval of 30 to 60 minutes were recorded for each of the three possible conditions, wherever they were clearly visible on the charts. These values were recorded for each phase of the system and means calculated for each sample. Sample sizes varied from 8 to 11 points per phase/motor combination because currents were not always discernable on the chart recordings for all three levels (Appendix D).

Other information pertinent to the energy calculations, as well as draft measurements, was contained in the files created by the data acquisition program on magnetic tape. Data from these tapes were removed with the aid of an HP 82169A interface to text files located on floppy discs. A typical file contained the seven measurands and the times corresponding to the measurement loops in separate columns (Table 2). These files were subsequently loaded into a spreadsheet for manipulation

Table 2. Sample portion of a data acquisition file.

gantry-tool frame separation. (cm)	angle error (degrees)	left motor on/off	right motor on/off	theo. speed (cm/sec)	actual speed (cm/sec)	draft (N)	time (seconds)
332.69	7.37	0.	0.	.00	.00	559.20	1.63
335.37	7.43	0.	0.	.00	.00	554.78	3.22
335.37	7.42	0.	0.	.00	.00	548.15	4.82
337.71	7.43	0.	0.	.00	.00	561.41	6.41
340.60	7.43	0.	0.	.00	.00	546.81	8.01
341.51	7.18	1.	1.	3.45	2.36	851.05	9.61
341.66	6.45	1.	1.	3.38	2.96	900.43	11.21
341.11	6.58	1.	1.	3.39	2.68	843.46	12.80
340.65	6.62	1.	1.	3.39	2.90	810.32	14.40
339.79	6.59	1.	1.	3.39	2.48	736.06	16.00
339.64	6.67	1.	1.	3.39	2.84	670.16	17.59
338.37	6.72	1.	1.	3.39	3.16	660.65	19.19
336.34	7.17	1.	1.	3.39	2.92	736.35	20.79
334.51	7.18	1.	1.	3.39	3.06	686.78	22.39
334.31	7.43	1.	1.	3.39	2.92	658.05	23.99
330.86	8.03	1.	1.	3.39	2.66	803.21	25.59
327.41	8.51	1.	1.	3.39	1.92	687.55	27.18
326.70	7.40	1.	0.	3.39	1.42	797.06	28.78
326.75	6.45	1.	1.	3.39	2.52	908.21	30.38
323.20	7.42	1.	1.	3.39	2.55	861.14	35.17
321.68	7.78	1.	1.	3.45	3.04	746.05	36.76
321.37	8.15	1.	1.	3.39	2.96	732.12	38.36
319.95	8.14	1.	0.	3.39	1.53	589.94	39.96
319.39	6.41	1.	1.	3.39	3.16	549.21	41.56
318.13	6.57	1.	1.	3.39	2.73	607.14	43.15
315.18	7.02	1.	1.	3.39	2.75	742.88	44.75
311.48	7.42	1.	1.	3.39	3.13	783.90	46.35
310.11	7.78	0.	0.	.01	.00	626.35	47.95
309.86	7.77	0.	0.	.00	.00	590.33	49.55
312.39	7.78	0.	0.	.00	.00	593.11	51.15
313.97	7.78	0.	0.	.00	.00	574.67	52.74
315.84	7.78	0.	0.	.00	.00	581.68	54.34
319.70	7.42	0.	0.	.00	.00	586.87	55.93
321.37	7.42	0.	0.	.00	.00	584.95	57.53
325.13	7.43	0.	0.	.00	.00	569.10	59.13
326.40	7.43	0.	0.	.00	.00	573.23	60.73

and analysis. Each of the dynamic measurands, as recorded by the data acquisition program, was plotted against time. These plots were reviewed for anomalies or inconsistencies before any analysis was done. Typical plots for each of the measurands from a typical data run are given in Figures 19 through 22. A summary of these data files is contained in Appendix F.

Examination of the plots representing actual groundspeed indicated a number of points at which the actual groundspeed momentarily exceeded the theoretical speed. Since this is an unlikely, if not impossible, situation, the system for measuring actual groundspeed was reviewed, and some limitations observed. The chain and sprocket system that was used to increase the speed of the input shaft of the frequency encoder from the ground drive wheel contained a small amount of slack. In a steady state test, as in the laboratory calibration of the system, this slack in the chains did not affect the system noticeably. In the field, however, any freedom of motion in that mechanism tended to produce an intermittent surge and lag in the speed of the input shaft to the frequency encoder. The result was an occasional indication of an actual groundspeed in excess of, or far below, the theoretical groundspeed. The problem was accentuated during the cultivation operation, when the ground drive wheel was

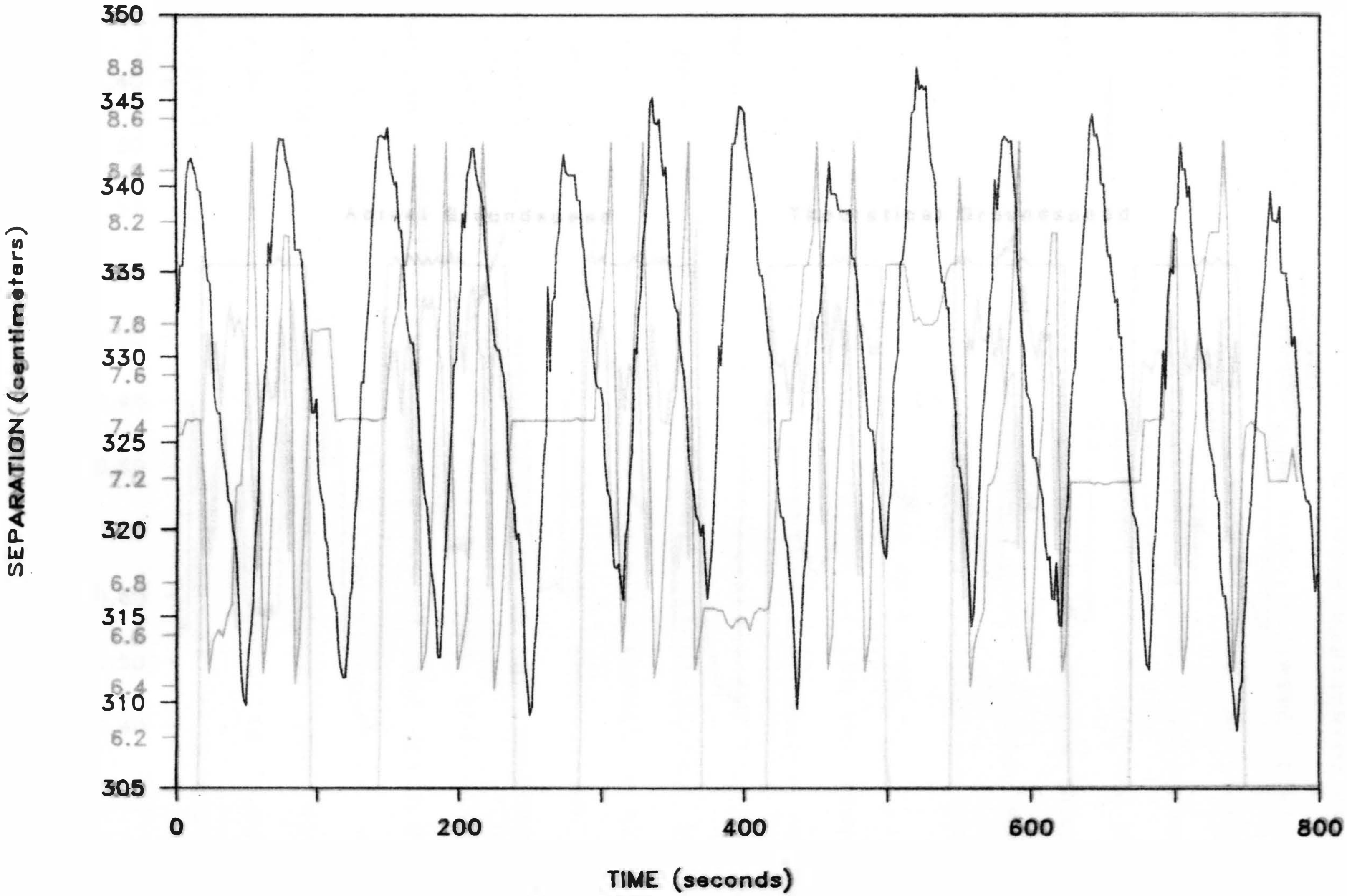

Figure 19. Tool-frame-reference boom separation measured during a planting operation.

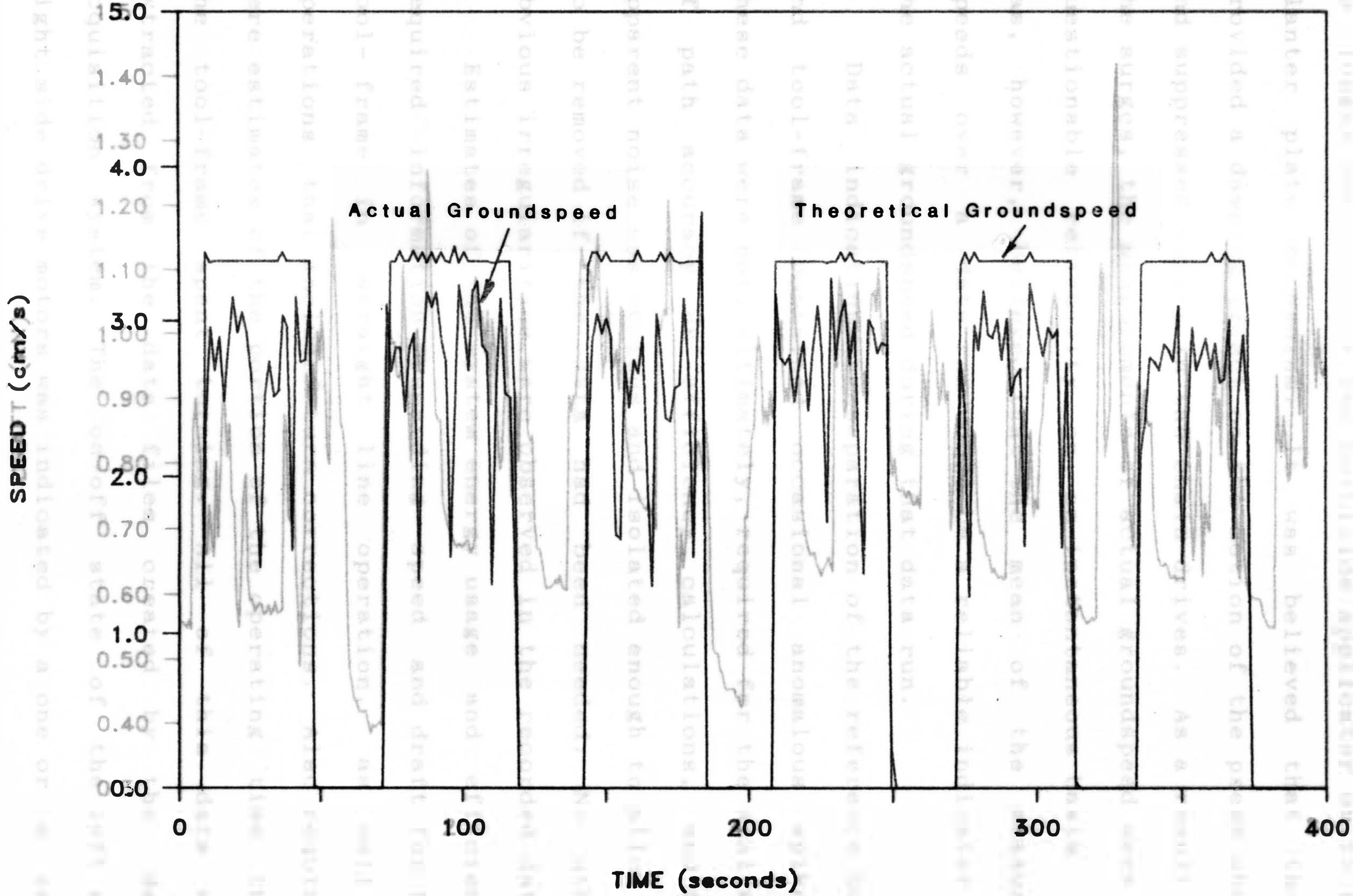

Figure 21. Theoretical and actual groundspeeds measured during a planting operation.

no longer connected to the herbicide applicator or to the planter plate mechanism. It was believed that these provided a damping effect on the motion of the press wheel and suppressed surges in the chain drives. As a result of the surges, the measurements of actual groundspeed were of questionable reliability on an instantaneous basis. It was, however, believed that the mean of the measured speeds over a data run would be a reliable indicator of the actual groundspeed during that data run.

Data indicating the separation of the reference boom and tool-frame contained occasional anomalous spikes. These data were not, ultimately, required for the analysis of path accuracy or efficiency calculations, and the apparent noise was obvious and isolated enough to allow it to be removed if the data had been needed. No other obvious irregularities were observed in the recorded data.

Estimates of the system energy usage and efficiency required information regarding speed and draft for the tool- frame in straight line operation, as well as operations that included turn corrections. Also required were estimates of the portion of the operating time that the tool-frame spent turning. All of this data was extracted from the data files created by the data acquisition system. The on-off state of the left and right side drive motors was indicated by a one or a zero in the corresponding column of the file (Table 2). A one indicated the "on" state while a zero indicated the "off" state. These indicators were used to create two additional sets of files. The first was a duplicate of the original files, with the exception that all rows of data in which both drive motors were indicated to be off were excluded. The result was a data set in which the tool-frame was in continuous motion, including turns. A second alternate type of file was generated by copying the first duplicate, while excluding rows of data in which the tool-frame was indicated to be turning. This provided a data set consisting of continuous straight line motion of the tool-frame and implement.

Mean values of draft, theoretical speed, and actual groundspeed were calculated over the full length of the files representing continuous running, inclusive of turning, for each of the data runs for planting and cultivating. This procedure was repeated for the files representing continuous straight line running.

The following procedure was used to determine the fraction of the running time that was spent turning. The number of data loops in which the machine was found to be turning was counted from each of the files representing continuous running, inclusive of turns. This number was divided by the total number of loops in the same file.

The resulting quotient represented the fraction of the total running time for the tool-frame that was devoted to turns. This fraction was a function of the radius at which the tool-frame was operating. Smaller radii required more frequent turns and a larger portion of total run time devoted to them. To determine this relationship, each data run was associated with a pair of radii that were measured along the reference boom. One value was measured at the start of each run, and one at the conclusion. The mean of these two radii was associated with the fraction of time spent turning, for the corresponding data run (Appendix C). An exponential curve was fit to the resulting pairs of data (Figure 23). The relationship between percentage of time spent turning and boom radius, for planting, was given by:

$$Y = 55.94*e^{-.116M}$$

$$r^2 = .983$$

where **Y** is the percentage of tool-frame run time spent turning and **M** is the operating radius in meters. In this case, r is the correlation coefficient for **ln**(% time turning) regressed on **M**. This relationship allows a prediction of the turning time if the radius and total running time are known.

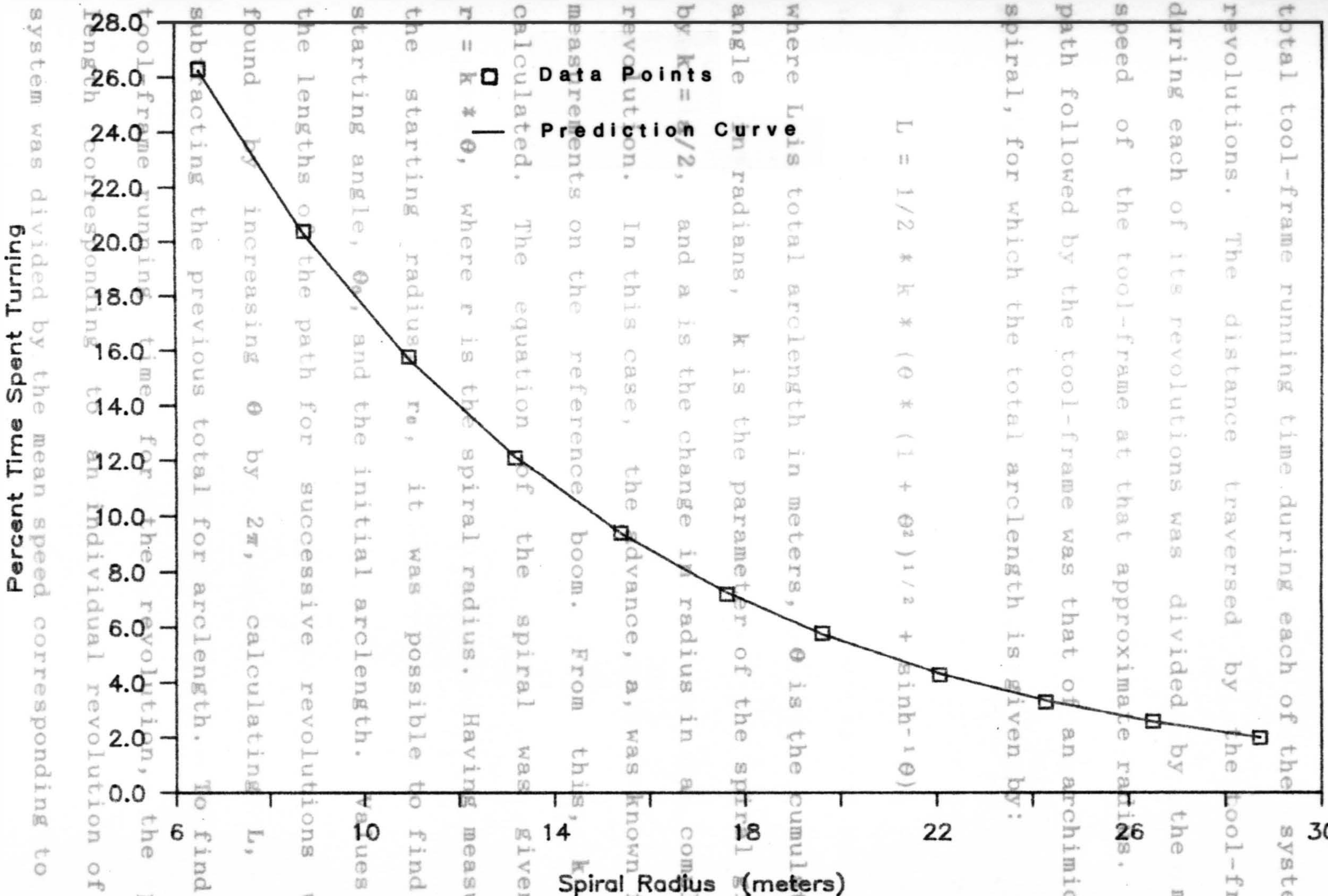

Figure 23. Regression line relating percentage of tool-frame running time spent turning, to the spiral radius.

The following procedure was used to determine the total tool-frame running time during each of the system's revolutions. The distance traversed by the tool-frame during each of its revolutions was divided by the mean speed of the tool-frame at that approximate radius. The path followed by the tool-frame was that of an archimidean spiral, for which the total arclength is given by:

$$L = 1/2 * k * (\theta * (1 + \theta^2)^{1/2} + \sinh^{-1}\theta)$$

where L is total arclength in meters, θ is the cumulative angle in radians, k is the parameter of the spiral given by $k = a/2$, and a is the change in radius in a complete revolution. In this case, the advance, a, was known from measurements on the reference boom. From this, **k** was calculated. The equation of the spiral was given by $r = k * \theta$, where r is the spiral radius. Having measured the starting radius, r_0, it was possible to find the starting angle, θ_0, and the initial arclength. Values for the lengths of the path for successive revolutions were found by increasing θ by 2π, calculating L, and subtracting the previous total for arclength. To find the tool-frame running time for the revolution, the path length corresponding to an individual revolution of the system was divided by the mean speed corresponding to the

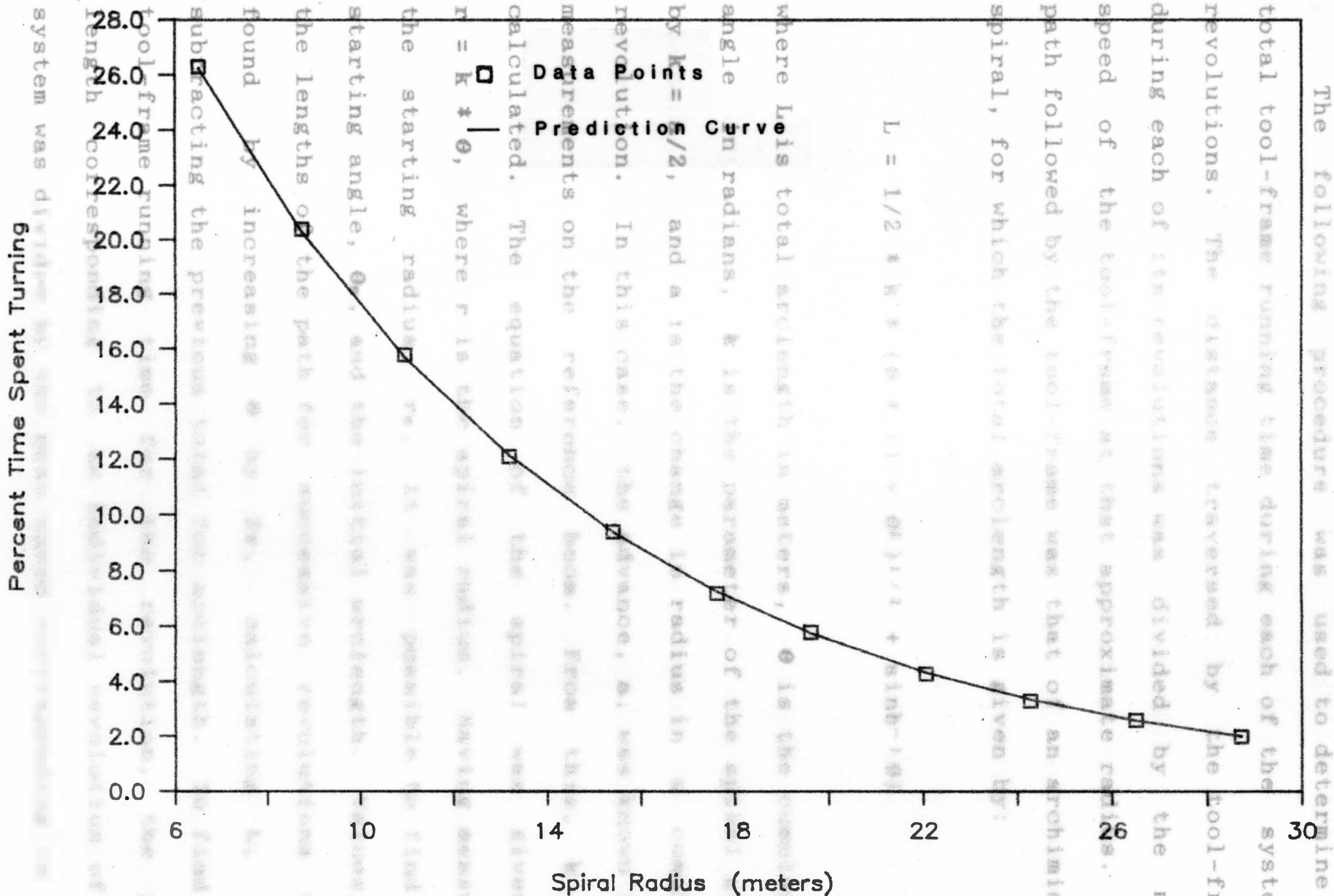

Figure 23. Regression line relating percentage of tool-frame running time spent turning, to the spiral radius.

average radius during the revolution. The average radius during this segment of the spiral was used, with the relationship developed earlier, to determine what portion of the running time was spent turning and what portion was spent moving forward. Values of arclength, running time, and percent time spent turning, are given for both planting and cultivation operations in appendix C.

The total length of time that the reference system spent running, and ultimately, the length of time required to complete an operation, was determined by the speed of the reference boom and the number of revolutions required to cover the plot. Measurements of the time required by the boom to make a complete revolution were taken, and the interval was found to be 75 minutes per revolution.

Near the end of the planting operation it was apparent that the tool-frame speed was no longer sufficient to overtake the reference boom as required. To alleviate this problem, the slightly larger drive wheels of the reference boom tower were exchanged with the drive wheels on the tool-frame. This reduced the speed of the reference system while increasing the speed of the tool frame. The time required for the reference system to complete the eleven revolutions was 13.75 hours and 15.11 hours, respectively, for the large and small wheels. The former value was used as the reference system run time for planting, since that set of wheels was used for nearly the entire planting operation. The larger value was used as the reference system run time for cultivation, since that set of wheels was in place for that operation.

The difference between these times and the summation of the tool-frame running times, summed over the 11 revolutions, represented the total time that the reference system ran alone for each operation.

$$T_{operation} - T_{straight} - T_{turning} = T_{ref\ only}$$

Total energy required for an operation by each power phase was found as the sum of the products of voltage, current, power factor, and time, for each of the three possible motor conditions. Once the times for turning, forward running, and reference system running alone, were known; it was possible to make energy calculations for the planting operation and the cultivation operation by using the appropriate values of voltage and current for the operation (Table E.1).

Total work produced by the system during an operation was found as the product of the mean draft for the entire operation and the total distance traversed by the tool frame and implement during the operation. That distance was found by summing the arclengths for the 11 revolutions of the system.

Energy consumption and efficiency of a conventional system

An estimate of the energy consumption and efficiency of conventional crop operations was used as a comparative evaluation of the electric tool-frame and spiral mechanization system. Stephens, et al. (1981) studied energy requirements for tillage and planting operations. They found an energy consumption rate of 46 kWh/ha for a John Deere 4640 diesel tractor and John Deere 7000 Max-Emerge planter. The combination was operated at a speed of 2.5 m/s while loaded with seed, fertilizer, and insecticide. Planting depth was 2.5 centimeters and the soil was a silt loam. The planter was an eight row unit with a row spacing of .762 meters. These parameters were not unlike those used for the spiral mechanization system with the exception of the operating speed. Draft measured for the above field speed was 11 kN. A similar analysis of a John Deere 830 row crop cultivator found an energy consumption rate of 45 kWh/ha. The cultivator was operated at a speed of 7.6 km/hr at a depth of 5 centimeters in a sandy loam. Cultivator gangs contained five "S" tines, similar to the configuration used by the electric tool-frame system. Draft force measured was 9 kN.

Each of these implements would have required 1640 meters of linear operation to cover an area of one hectare. The product of that distance and the measured drafts provided an estimate of the energy required at the drawbar by the operation for one hectare. When this value was divided into the per hectare fuel energy consumption rate, the result was an overall drawbar energy efficiency for the operation. This efficiency value was readily comparable to a similar measurement of efficiency for the spiral mechanization system.

The measured energy sequestered in a liter of the diesel fuel used in the Stephens, et al.(1981) study, was 10.84 kWhr, or 39.024 MJ. These values were used in the calulation of per hectare fuel consumption and per hectare fuel costs. For consideration of the total energy required by a system to perform an operation, the per hectare energy requirements found in the Stephens, et al. study were divided by .823 to reflect the energy cost of producing the fuel (Fluck and Baird, 1984).

Draft model predictions

The draft model of McKyes (1985), was to be tested against the measured values of draft for the cultivation operation. The prediction equation for horizontal draft was given by:

$$H = P\sin(\alpha + \delta) + c_a dw\cot\alpha$$

and

$$P = (\gamma g d^2 N_G + cdN_c + qdN_q + c_a dN_{ca})w$$

The values of the soil parameters **c**,**c_a**,**ϕ**, and **δ** were measured with a Cohron sheargraph in the field. Values of ultimate shear strength were used, as indicated on the graphs, to estimate cohesion, **c**, and soil internal friction angle, **ϕ**. The average value of **c** and **ϕ** for the eight shear tests taken was used for the model. The value of **ϕ** found was 42.3 degrees. The cohesion value, **c**, found was 7.17 kPa. The average of the values obtained for soil to metal friction angle, **δ** of 20.8 degrees was used. The indicated values of soil to metal adhesion were in excess of the theoretical limit, **c**, for the soil and measuring instrument. This adhesion value was not in proportion to the other soil parameters. The adhesion values were assumed to be erroneous and an alternate estimate of adhesion was made. A value of adhesion equal to one half of the soil cohesion was assumed for the model. This assumption was used previously by Grisso, et al. (1980), and had been approximated under some circumstances for cohesive soils by McKyes (1985). Soil density was measured with a Euling sampler and the mean value of soil density from 13 samples was used in the draft predictions.

No soil surcharge pressure was present during the draft tests. As a result, the surcharge term was not included in the calculation of predicted draft.

Soil tool widths, **w**, and rake angles, α, were measured for each of the five tines on one cultivator gang. The tines were actually sweeps ranging in width from 6.35 to 17.8 centimeters. The sweeps were curved so that the rake angle actually increased with depth. The static rake angle was measured as a chord from the tip of the tool to a point approximately at the soil surface for the average operating depth. An increment of five degrees was added to the measured rake angles to approximate the true rake angle when the tines were under a draft load and depressed rearward. The average of the rake angles for the forward two tines of the gang measured was found to be 36 degrees. This value was applied to all of the forward tines on the cultivator. The average of the angles for the rear three tines of the gang was found to be 26 degrees and was applied to all of the rearward tines of the implement.

Depths were measured for each of the tines after each draft data run by scraping away the loose soil and measuring the depth from the undisturbed soil surface to the bottom of the tine furrows Depths for the tines varied substantially within each gang, so each tine depth was recorded for each data run. (Appendix G).

One data run was performed with the cultivator tines raised just above the soil surface. Since no soil tillage was performed during this run, the mean value of draft measured during the run was assumed to represent the rolling resistance of the cultivator. This value of 0.98 kN was added to the sum of the drafts predicted by the model for individual tines before comparisons were made to the measured cultivator drafts.

Values for the N factors for each of the terms in the draft prediction equation were obtained from charts provided by McKyes (1985). Families of N factor curves were given for different depth to width ratios and a variety of soil internal friction angles. McKyes' sets of curves were compiled for a soil to metal friction angle assumed to be two thirds that of the soil internal friction angle. A set of curves corresponding to an internal friction angle of 40 degrees and a soil to metal friction angle of 26.7 degrees was chosen for this set of draft predictions (Appendix H). These were chosen because they most closely matched the friction angles of 42.3 and 20.8 degrees that were measured for the test soil.

A computer program was written and used to aid in the repetitive portion of the calculations. A depth to width ratio was calculated for each soil tool, and this ratio was combined with the appropriate rake angle to determine N factors from the McKyes charts. Once the N factors were known, the draft force for each tine was calculated and these individual drafts were summed for all of the implement tines. The rolling resistance of the cultivator was added to this sum. The resulting value could then be compared to the draft actually measured for that data run. This process was repeated for each of the six draft runs evaluated. A sample calculation of draft for a single tine is given in appendix G.

Results and Discussion

Path Accuracy

Results of path related tests can be divided into two parts. First of all, comparisons were made to determine if the position reference system was able to generate a spiral directrix for the tool-frame that was consistent between the three operations performed. Secondly, comparisons were made to determine if the tool-frame performed consistently relative to the directrix for the three operations.

The mean values of the radial advance increments for subsequent revolutions were compared using non paired, two tailed, Student's t tests. The mean of the increments measured along the reference boom during the planting operation was compared to the mean increment generated during cultivation. The mean increment for planting was 2.219 meters while the mean for cultivation was 2.264 meters (Table 3). The comparison indicated a significant ($P \leq .05$) difference in the mean increments for these operations. The difference between these increments represented an error in the path during cultivation that was cummulative with each successive revolution of the system. The physical interpretation of the difference was clear in the field as the cultivator accumulated enough path error to begin cultivating out the corn rows after about four revolutions of the system. Without any mechanism to re-orient the reference system or automatically adjust the increment, the tool frame effectively removed most of the crop it had earlier planted.

Table 3. Mean values of radial increments measured on the reference gantry for planting, cultivation, and operating without an implement.

	Planting	cultivation	No Implement
Mean increment	2.219 (m)	2.264	2.216
Standard Deviation	0.044	0.012	0.012

A similar comparison was made between the increment generated along the boom during planting and one generated by the system running without an implement attached to the tool frame. In this case a Student's t test failed to show any significant difference between the advance increments. In fact, the cumulative difference in radius over eight revolutions was just 5.5 centimeters.

No obvious cause was found for the difference found between the cultivator increments and the planting increments, or the lack of the difference between increments for planting and operation without an implement. Because the tool frame only follows the reference point on the directrix, the demands on the directrix should not change between the operations. The ground drive wheel and its soil-to-wheel interface were the most likely sources of the difference. Change in tire pressure or soil conditions could have changed the effective rolling radius of this wheel and the result would be a change in the radial advance increment.

Comparisons were also made between the radial increments as generated on the ground by the implement for the three operations performed (Table 4).

Table 4. Mean values of radial advance increments as measured on the ground for planting, cultivation, and operation without an implement.

	Planting	cultivation	No Implement
Mean increment	2.275 (m)	2.326	2.293
Standard Deviation	0.094	0.066	0.050

In these cases, the t test failed to show a difference between increments for planting and cultivation, and between planting and operating without an implement. Again, a 95% level of confidence was used with a two tailed test. In the case of the comparison of planting and cultivation increments, the inablilty to show a statistical difference was overshadowed by the fact that the tool frame and cultivator accumulated enough error, relative to the planting path, to cultivate out the crop. This effective difference was born out when a comparison was made between cummulative ground radii. The ground radius after eight revolutions of planting was compared to an estimate[1] of the radius after eight revolutions of cultivation. The cumulative radius for eight revolutions of planting was 23.08 meters. The estimated radius for eight revolutions of cultivation was 24.02 meters, a difference of nearly one meter. In contrast, the difference between the accumulated ground radius for eight revolutions of planting and eight revolutions without an implement was only 7 centimeters. It would have been desireable to replicate these tests to determine if the variability in the advance increment found for these operations would be repeated. However, a single failure of the system to repeat the planting path is sufficient to impact the conclusions of this work.

It was possible that some of the path error that caused the cultivator to remove the crop rows was due to deviation of the tool-frame and implement from the

[1] An estimate was used because of a malfunction in the system during cultivation that eliminated one increment from consideration and precluded the use of the measured cumulative radius. The estimate was found as the product of the mean of the valid increments and the desired number of revolutions. This product was added to the same starting radius as used for planting.

directrix generated rather than from an erroneous directrix. To isolate this type of error, comparisons were made between the ground measured radii and the boom measured ones for the same operation. A Student's t test at a 95% confidence level indicated a difference between the mean advance increment on the ground and the increment on the reference boom for planting. A similar difference was found for operation without an implement. The t test failed to indicate a difference between generated directrix increment and the path increment followed for cultivation. In all cases, the mean radial advance increment measured on the ground appeared to be larger than the increment generated by the directrix. Physically interpreted, the tool-frame gradually moved from the inside toward the outside of the ideal spiral directrix as the path was generated.

The cumulative difference between the generated directrix spiral and the one followed by the tool-frame was plotted against the revolution number for eight revolutions of the system for the three operations (Figure 24). Ground radii were marked at the rear of the implement for planting and cultivation, while they were marked at a point plumb beneath the drawbar for operation without an implement. This may account for the offset of the line representing operation without an implement in

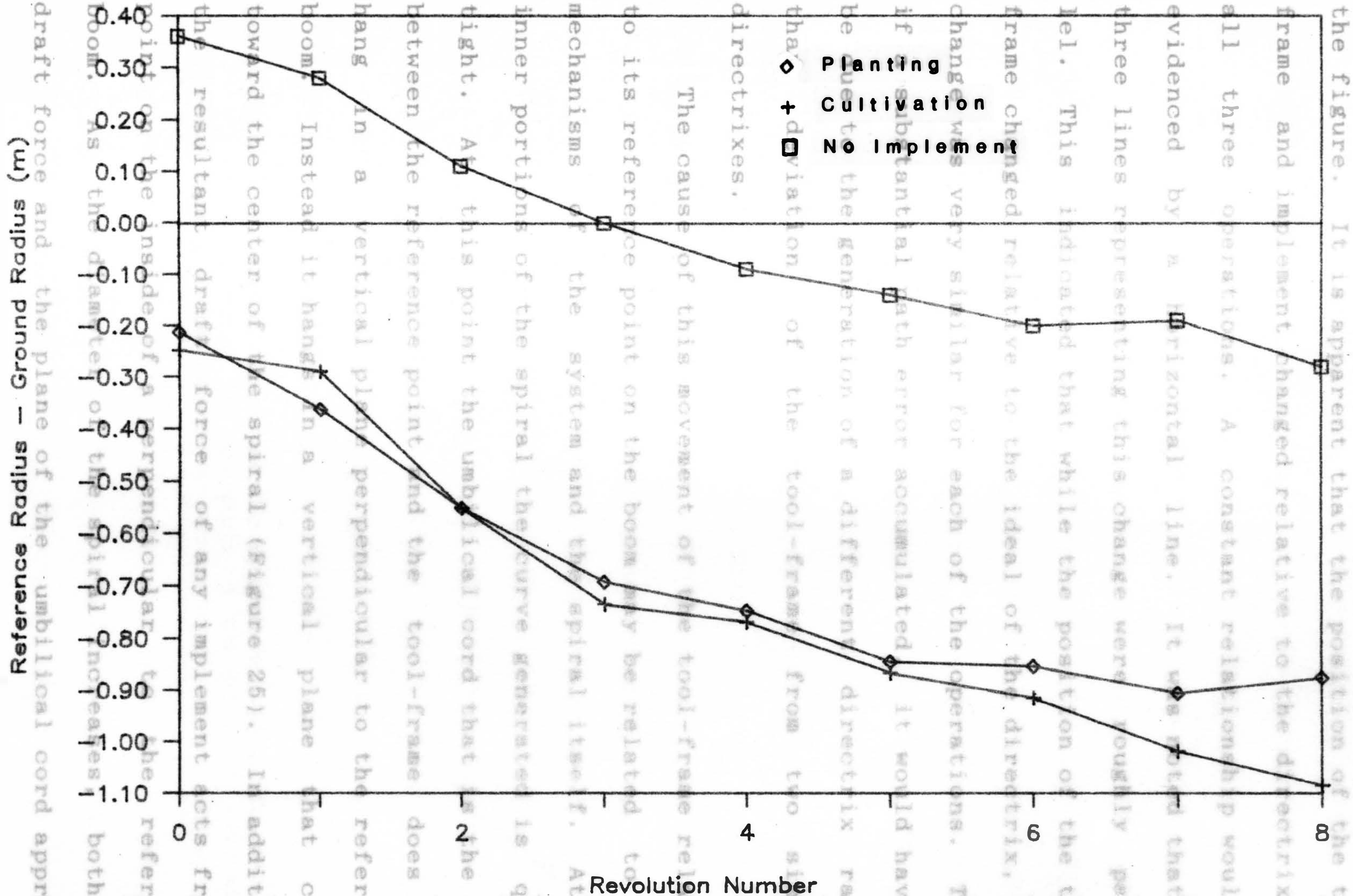

Figure 24. The relative position of the generated row to the reference point on the directrix for eight system revolutions and three operations.

the figure. It is apparent that the position of the tool-frame and implement changed relative to the directrix for all three operations. A constant relationship would be evidenced by a horizontal line. It was noted that the three lines representing this change were roughly parallel. This indicated that while the position of the tool-frame changed relative to the ideal of the directrix, the change was very similar for each of the operations. Thus, if a substantial path error accumulated, it would have to be due to the generation of a different directrix rather than deviation of the tool-frame from two similar directrixes.

The cause of this movement of the tool-frame relative to its reference point on the boom may be related to the mechanisms of the system and the spiral itself. At the inner portions of the spiral the curve generated is quite tight. At this point the umbilical cord that is the link between the reference point and the tool-frame does not hang in a vertical plane perpendicular to the reference boom. Instead it hangs in a vertical plane that cants toward the center of the spiral (Figure 25). In addition, the resultant draft force of any implement acts from a point on the inside of a perpendicular to the reference boom. As the diameter of the spiral increases, both the draft force and the plane of the umbilical cord approach the normal to the reference gantry. This gradual change may account for the gradual re-orientation of the tool-frame to the reference point or directrix. It would be expected that at some point this re-orientation would reach a maximum and the ground and directrix advance increments would be identical from there on.

Regardless of the cause of this relative movement of tool-frame and directrix, its consistency indicated that any large accumulated path errors were due to differences in the directrix generated.

The implications of such a cumulative error (as found with cultivation) are important to the evaluation of appropriate uses for the present system. Obviously, if the system were to be used for planting and cultivating row crops, it would be necessary to modify it in some way to increase the repeatability of the path between operations. If the exact source of the variation between the operations of cultivation and planting could be found, it might be possible to reduce this variation and eliminate the problem.

The fact that the differences [illegible] for cultivation and planting were consistent from revolution to revolution lends credence to the idea that an adjustment might cure the problem. The ability of the system to reproduce the planting path quite accurately

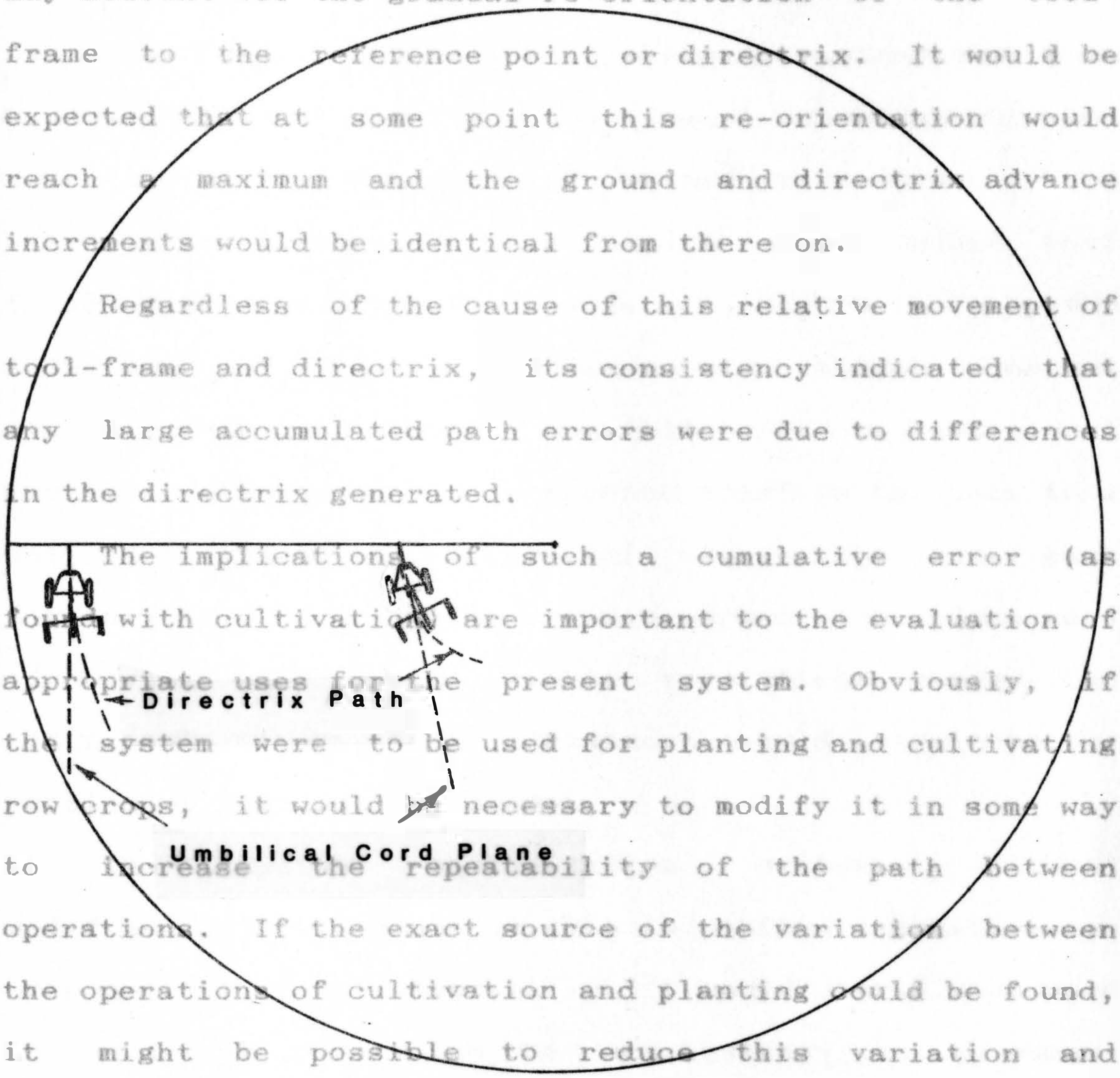

Figure 25. **Plan view schematic of the system depicting the position of the tool-frame and the angle of the umbilical cord at the limits of the spiral.**

the normal to the reference gantry. This gradual change may account for the gradual re-orientation of the tool-frame to the reference point or directrix. It would be expected that at some point this re-orientation would reach a maximum and the ground and directrix advance increments would be identical from there on.

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The fact that the differences in increment for cultivation and planting were consistent from revolution to revolution lends credence to the idea that an adjustment might cure the problem. The ability of the system to reproduce the planting path quite accurately without an implement is puzzling, but does indicate that the potential for a repeatable path exists.

Very little variability of the path, other than the gradual movement of the tool-frame outward across the reference point, was observed within revolutions. This was not unexpected since the amount of allowable tool frame error from the reference path is controllable by the adjustment of the cams and microswitches in the control system. This left open the possibility of an additional mechanism to correct the reference point on the boom to a desired increment at an interval of, perhaps, once each revolution. The existing advance mechanism could be used to advance the system within revolutions while the additional intermittent adjustment would eliminate the problem of cummulative errors.

The spiral mechanization system, without modification, would not be able to perform repetitive row crop operations satisfactorily. It could possibly be used for an operation in which the consequences of a cummulative path error were not as damning as they were for cultivation of a row crop. An example of such a function might be turf mowing in the production of sod. Alcock and Jahns (1983) suggested this as an appropriate use for a spiral mechanization system. The overlap or underlap of the mowing operation between revolutions would not be a particularly serious fault, and could be corrected by subsequent passes over the plot without adverse consequences. Other limitations of the system, such as speed or energy requirements may, however, render the system inappropriate for turf farming.

Energy Efficiency

Overall energy efficiency was calculated as the work done at the drawbar, in kWhr/ha, divided by the heat content of the fuel burned to perform the work. Drawbar work performed by the spiral mechanization system during a planting operation was found to be 1.11 kWhr/ha. Fuel energy required to perform that operation was 136 kWhr/ha (Table 5.). Overall efficiency of the electric system for a planting operation was 0.81%.

Table 5. Energy efficiency for the electric system and a conventional cropping system.

	Work Output	Energy used*	Efficiency
$Planting_{elec}$	1.11 kWhr/ha	137 kWhr/ha	0.81%
$Planting_{conv}$	5.0	55.8	8.9
$Cultivation_{elec}$	2.91	180	1.6
$Cultivation_{conv}$	4.11	54.7	7.5

* Represents energy sequestered in the fuel burned to do the work, including fuel production energy.

The same operation performed with a conventional diesel tractor consumed energy at a rate of 55.8 kWhr/ha (Stephens, et al.,1981). Work output of that system was 5.0 kWhr/ha, yielding an overall efficiency of 8.9%. This indicates a large disadvantage for the electric system in overall energy efficiency for a planting operation.

The drawbar work done by the spiral mechanization system while cultivating was found to be 2.91 kWhr/ha. The energy inputs during that operation were calculated to be 180 kWhr/ha. Overall efficiency for the electric system while cultivating was 1.6%. Similarly, a diesel tractor and cultivator consummed 54.7 kWhr/ha while doing work equivalent to 4.11 kWhr/ha. Overall efficiency for this conventional system was 7.5% during the cultivation operation. Again, the electric system was shown to have a much lower overall efficiency in the conversion of fuel energy to usable field work.

It was not within the objectives of this work to locate the sources of inefficiencies in the electric system. However, it was noted during operation, that the electric motors powering the tool frame were never loaded to a level near their rated capacity. The load applied to the drive motors was a function of the gear reduction and the implement draft. The low speeds and narrow implements used by the test system kept motor loads relatively low and were responsible, in part, for the low system efficiency. A heavier tool frame, a wider implement, and higher operating speeds would allow loading of the drive motors to levels at which they might operate efficiently.

A second factor contributing to the inefficiency in the spiral mechanization system was the powering of the reference system. The motor powering the irrigator boom ran continuously during the field operations. None of the energy consumed by this motor contributed to the mechanical work done at the implement drawbar. Because the reference system motor ran longer than either of the tool frame drive motors, it is possible that one half or more of the energy consumed during the operation was used to provide position reference and guidance, as opposed to providing draft for the implement. This type of inefficiency is inherent in the test system used, although a wider implement would reduce the number of system revolutions required and reduce the energy involved in guidance.

Energy Cost

The cost of electrical energy varies widely with locale. The utility serving the area which included the test site applied the following rates to irrigators. For center pivot irrigators without electric pumps, a rate of

$.055/kWhr for energy used was added to a flat annual fee of $500/pivot. For irrigators using electricity for pumping, the rate schedule included $.068/kWhr for the first 677 kWhr/kW (500 kWhr/hp) and $.055/kWhr for energy used beyond 677 kWhr/kW (500 kWhr/hp). These amounts were added to a $26.81/kW ($20/hp) annual charge. For the calculations of energy cost for the test operations, a rate of $.068/kWhr was used for each kWhr drawn.

The price of diesel fuel, for comparative purposes, was assumed to be $.264/l ($1.00/gal).

The unit costs of the energy sources were applied to the energies actually used in the field. That is, the prices of electricity and diesel fuel were applied to the corresponding energy quantities for which a producer would be required to pay.

The electricity used in the planting operation was found to be 35 kWhr/ha (Table E.1.). The cost of this energy to the producer was $2.39/ha (Table 6). The fuel used by the diesel tractor and conventional planter to plant a hectare was was 4.24 liters. At the fuel cost of $.264/l for diesel fuel, the planting operation had an energy cost of $1.12/ha with the conventional system.

Electricity used to cultivate a crop was found to be 46 kWhr/ha. The corresponding cost was $3.15/ha. The conventional system required 4.15 liters of diesel fuel at a cost of $1.10 to cultivate one hectare.

Table 6. Energy costs for the electric system and a conventional cropping system.

	Electricity*	Diesel Fuel**
Planting	$ 2.39/ha	$ 1.12/ha
Cultivation	$ 3.15/ha	$ 1.10/ha

* $ 0.068/kWhr
** $ 0.264/liter

There appears to be a disparity between the calculations of overall efficiency and energy costs. The overall efficiency of the electric system is higher for cultivating than for planting, yet the cost difference between the electric system and the conventional one is more pronounced for the cultivation operation. The reason is that measured drafts for cultivating with the electric system were much higher than those measured for planting. So, while the higher draft may result in a somewhat higher efficiency for the electric motors in the system, total energy use was also higher. In contrast, the Stephens et al. (1981) study found lower drafts and lower fuel consumption for a cultivation operation than for planting.

These comparisons indicated energy costs for a producer using the spiral mechanization system of 2 to 3 times the fuel cost of a conventional tractor-implement combination. It should be noted that the comparisons made did not include other operating costs such as oil, repairs and maintenance, labor, or fixed ownership costs. A comparison of overall costs of an electric system and a conventional one might indicate a smaller advantage for the conventional system, or perhaps, an advantage in overall costs for the electric system.

Subjective Observations and Evaluation of the Spiral Mechanization System

This was the first test of the spiral mechanization system in an actual field operation, and it was felt that the operation and observation of the system might disclose advantages or disadvantages of the system not otherwise addressed in the objectives of this study.

System Reliability The reliability of the automatic guidance system was evaluated subjectively during field operations. Two problems surfaced during these operations. The first was one that involved the flexible cords supplying power to the tool frame. These would occasionally catch on some facet of the central tower. If not watched, this problem would damage the cords, or the tool frame if the cord should separate. The use of slip rings to provide continuous power to the tool frame and reference system would eliminate this potential problem. The second problem occurred during cultivation when the cable carrier supporting the umbilical cord became lodged on a portion of the track on which it rode. This caused the cable loop advancing the reference point to slip. As a result, the reference system departed from the desired spiral path associated with the existing rows and, with no way to relocate itself short of human intervention, would have continued to deviate from the desired path. This problem could easily be avoided in the future by modifying some small portions of the cable carrier. These malfunctions did point out the need for some form of safety mechanism to switch off the system in the event of a malfunction. With the exception of the aforementioned problems, the automatic guidance system performed very well. With a few minor alterations and the addition of one or more failsafe switches, the system could satisfactorily operate unattended. These observations refer to the ability of the tool frame and reference system to function together without anomalous variations. Any use of the system would still be constrained by the limitations in terms of path repeatability found earlier.

Tool Frame Performance The electric tool frame performed reliably throughout the test field operations. It was a very simple machine and required little adjustment or maintenance. Several limitations were observed, however. The tool frame had drive motors and gear reductions identical to those on the reference gantry. The use of the larger of the two sets of wheels on the tool frame allowed it to overtake the reference boom, as required. However, if the tool frame wheel slip exceeded some critical level the separation of the reference boom and tool frame would gradually increase until the umbilical cord became taut and was damaged. This would have occurred had the system not been controlled manually near the outer limit of the spiral path. Average wheel slip for planting was 19%. Weight in the form of steel and concrete blocks totaling 4.52 kN was added to reduce this slip prior to cultivation of the crop. Average value of slip for the cultivation operation was 14%. Even with the extra weight and a slip value not particularly excessive or unusual, the speed of the tool frame was insufficient to overtake the reference system at its outer limit. Possible solutions to this problem include the use of a much heavier tool frame to decrease slip, and a smaller gear reduction to the drive wheels to increase the speed. The decrease in the gear reduction from motors to axles would have the added benefit of increasing the power required of the motors. Loading these motors more closely to their capacity would undoubtedly increase the overall efficiency of the machine.

<u>**Speed**</u> Tool frame speeds of 2.70 to 3.60 cm/s were measured during field operations. These were, roughly, two orders of magnitude lower than the speeds used by Stephens et al. (1981) for the same operations performed with conventional equipment. The low speeds did not prevent the implements from performing their functions, and, in any case, it would be possible to modify or develop implements to improve their performance at low speeds. The low overall speed of the system does, however, limit its applications. The test system made use of a single span of a center pivot system. With the outer end of this span running continuously, an operation such as planting could be performed in 15.11 hours. The area covered by that single span is .2702 ha. Such a low work rate makes the existing system inappropriate for most crops. The problem of low work rate would be accentuated as tower spans were added to the system. The speed of the system is limited by the speed of the outer tower and its drive mechanism. If an irrigator was used without alteration to the drive system, the addition of a span

would simply increase the time required to complete a revolution. The speed problem could be reduced if modifications were made to the center pivot drives to increase the angular speed of the system. The work rate of the tool frame would still be low near the center of the pivot, where the tangential speed of the irrigator is low, but it might be able to complete an operation in a reasonably timely manner. Such modifications of the irrigator may not be compatable, however, with its role as a water applicator.

Alternative Systems A possible alternative system using a linear move irrigation system came to light during testing of the spiral mechanization system. The alternative concept may be able to solve some of the problems inherent to the center pivot centered guidance system. An existing linear move irrigation machine could be used as a position reference device and mobile power source. An electric tool frame similar to the one in use by the spiral mechanization system could be used to propel implements. Instead of following perpendicular to the irrigator, the tool frame could be made to run parallel to the irrigator gantry. The tool frame would operate back and forth between the ends of the irrigator, while the irrigator incremented one implement width forward after each pass. Such a system would have at least three distinct advantages. 1) The reference system would only cover the field once to perform an operation, instead of many times. As a result, the energy devoted to tool frame reference could be greatly reduced. 2) The tool frame speed could be independent of the reference system speed. Because the tool frame would follow along the gantry instead of behind it, the tool frame could be built to go much faster and make more efficient use of the electric power used by the drive motors by keeping them properly loaded. 3) The tool frame could run nearly continuously at a more conventional speed greatly reducing the time spent to complete an operation.

Problems with such a system would include a more complex steering system to guide the tool frame at higher speeds and reverse it at the system ends. Also, some mechanism other than flexible cords, would probably be needed to transfer electric power to the tool frame. Such a system would have greater potential to perform operations in a timely, efficient manner; a necessity if an automatic cropping system is to have a commercial application.

Draft Model Tests

The values of draft predicted by the McKyes model for the cultivator were compared to the measured values of draft for the corresponding tests (Table 7). In each comparison, the predicted value of draft exceeded the measured value. Values of predicted draft ranged from 172% to 101% of measured draft. A Student's t test of matched pairs was used to compare the means of the predicted and measured values. A two tailed test with a 95% confidence level was used. This comparison indicated a significant difference between the mean of the predicted drafts and the mean of the measured drafts. Both measured and predicted values were plotted against weighted average cultivator depths for the corresponding data runs (Figure 26). Weighted average depths were used because three different width tines were used and there was considerable variation in depth between tines. The weighted depth was calculated as:

$$D = (\sum w_i * d_i)/\sum w_i$$

where D is the weighted average depth, w is the individual tine width, d is the individual tine depth, and i is the subscript for the 16 tines of the cultivator.

Table 7. Predicted and measured drafts, and weighted average tool depths.

Depth	Predicted Draft	Measured Draft
4.44 (cm)	2.55 (kN)	1.72 (kN)
5.13	2.96	2.28
6.82	3.91	2.27
6.39	3.99	3.58
8.18	5.07	5.01
8.34	6.02	5.91

The measured and predicted drafts tended to converge as the depth increased. Several conditions were observed that may have contributed to higher values of predicted draft at these shallow depths. A layer of dry soil 1 to 2 centimeters thick was present on the surface of the soil. Cohesive and adhesive properties of the soil were measured beneath this dry layer and reflected the properties of the deeper, moist soil. Depth of tine measurements were made from the soil surface to the bottom of the tine furrow. So, while the measured depth was treated as having uniform properties, the dry, loose layer of soil on the surface would have had little cohesion or adhesion and would have contributed less to actual draft than indicated by the model. Any exaggeration of draft by the model due to this layer would become less pronounced as the depth increased. Also, while the draft model does not have any stated limits to its application, the depths used for these tests were relatively shallow, and may be near the lower limit of applicable depths for the model.

Other factors that must be considered in evaluating the accuracy of the predictions include; the accuracy of measurements of soil properties, appropriate rake ang[illegible] and tine geometry. Soil tests, in this case, were d[illegible] with the use of a Cohron Sheargraph. The values for s[illegible] internal friction angle and soil-to-metal friction a[illegible] appeared to have good repeatability. These, and the valu[illegible] for soil cohesion, were evaluated assuming the ulti[illegible] cohesion to be the appropriate soil parameters, as oppose[illegible] to the peak values of cohesion. The Sheargraph indicate[illegible] high peak values of cohesion, points at which the soi[illegible] initially fractures. If these peak values had been use[illegible] to determine the friction angle and cohesion, the mode[illegible] would have predicted much higher drafts.

The assumption that the soil-to-metal adhesion [illegible] equal to one half of the soil cohesion was made becau[illegible] reliable estimates of soil to metal adhesion we[illegible] unavailable. While probably not grossly inappropria[illegible] this value for adhesion does represent an approximati[illegible] Higher or lower estimates of this parameter would yield correspondingly higher or lower predictions of draft.

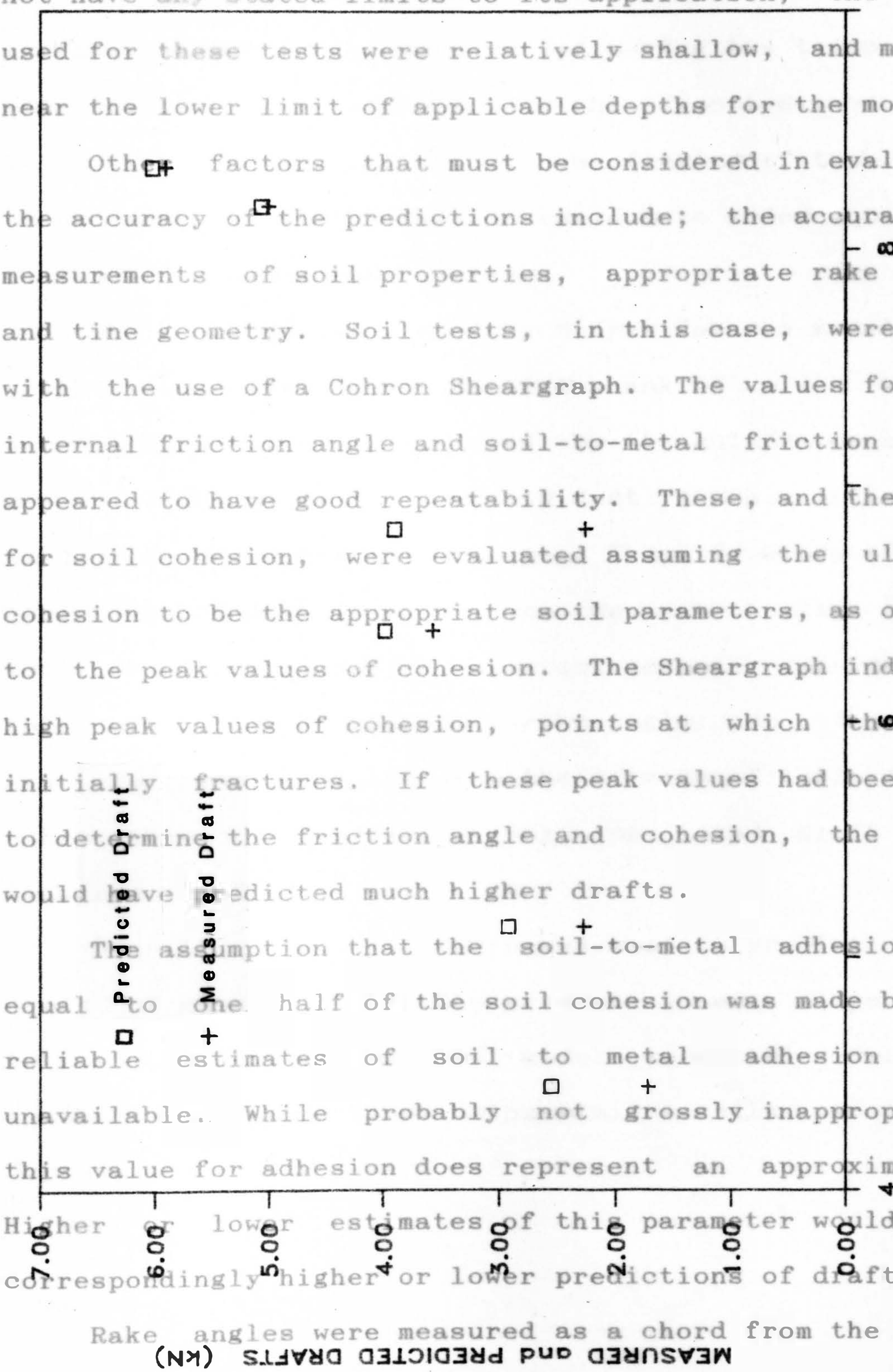

Figure 26. Measured and predicted cultivator drafts versus weighted average depth.

Rake angles were measured as a chord from the tip of not have any stated limits to its application, the depths used for these tests were relatively shallow, and may be near the lower limit of applicable depths for the model.

Other factors that must be considered in evaluating the accuracy of the predictions include; the accuracy of measurements of soil properties, appropriate rake angle, and tine geometry. Soil tests, in this case, were done with the use of a Cohron Sheargraph. The values for soil internal friction angle and soil-to-metal friction angle appeared to have good repeatability. These, and the value for soil cohesion, were evaluated assuming the ultimate cohesion to be the appropriate soil parameters, as opposed to the peak values of cohesion. The Sheargraph indicated high peak values of cohesion, points at which the soil initially fractures. If these peak values had been used to determine the friction angle and cohesion, the model would have predicted much higher drafts.

The assumption that the soil-to-metal adhesion was equal to one half of the soil cohesion was made because reliable estimates of soil to metal adhesion were unavailable. While probably not grossly inappropriate, this value for adhesion does represent an approximation. Higher or lower estimates of this parameter would yield correspondingly higher or lower predictions of draft.

Rake angles were measured as a chord from the tip of the curved tine to a point corresponding to the soil surface. Since depths were varied during the tests, there was probably some variation in the effective rake angle. In addition to this variation, the flexing of the spring teeth supporting the tines would have added some small variability to the rake angles.

The tool geometry used by McKyes for the model being tested was a simple straight flat shank of width **w**. The narrowest of the tines used by the cultivator closely resembled the model tine. The widest tines, or sweeps, used were triangular in shape. The side wings of these sweeps sloped down and away from the central axis of the tool. The only aspects of tool geometry considered in this work were tool width and rake angle. No attempt was made to isolate or evaluate the effects of tool geometry other than to compare overall predicted draft with measured draft.

Overall, the model performed well in predicting the draft of a cultivator. The predictions were close enough to actual drafts to make the model acceptable for many agricultural applications using similar soils, tools, and depths.

Conclusions

The spiral mechanization system was able to generate and Archimidean spiral path while planting a row crop. The system was not able to repeat that pattern with sufficient accuracy to cultivate the crop. The guidance system used by the tool-frame to follow the generated spiral directrix performed consistently between operations. The inability of the overall system to repeat the planting path was traced to the ground driven radial advance mechanism. This mechanism generated a spiral with a larger radial advance increment during cultivation than was generated for the planting operation. The source of the variation was most probably the ground drive wheel and its soil-wheel interface. If the system were to be used for row crop work, some modification or addition to the system would be required to eliminate cummulative path errors. Applications not requiring a close tolerance to the previous pass or operation could be performed with the system, as is.

The overall efficiency of the present spiral mechanization system from fuel combustion to drawbar work was found to be 0.81% for planting. Similarly, efficiency for cultivation of the crop was found to be 1.6%. These values compare to efficiencies of 8.9% and 7.5% for planting and cultivation with a conventional diesel system. The present spiral mechanization system represents an inefficient use of energy resources compared to a conventional system. The inefficiency of the electric system can be attributed to insufficiently loaded drive motors, and to the relatively large amount of energy required by the system to provide a constantly moving directrix.

The cost of the energy used by the spiral system to plant a crop was $2.39/ha compared to $1.12/ha for a conventional cropping system. Cost of cultivation energy was $3.15/ha for the electric system and $1.10/ha for the conventional one. Economic choices made solely on the basis of energy cost would strongly favor the conventional system over the present spiral mechanization system.

The following modifications to the spiral mechanization system would improve its practicability. 1) A mechanism to correct the directrix path to prevent cummulative errors. 2) The speed of the present system is low. If crop operations were to be performed in a timely manner, an increase in the operating speed of the reference system and tool-frame is highly recommended. 3) The drive motors of the tool-frame and reference system need to be sized to allow complete and efficient use of the energy inputs while performing field operations.

Many of the modifications recommended in the existing spiral mechanization system may be more easily adapted to a system that utilizes a linear move irrigator instead of a center pivot. If a system were designed to allow the tool-frame to operate parallel to the reference gantry, the tool-frame speed, and motor loading, could be independant of the irrigator speed. Energy usage by the reference system could be greatly reduced as the number of passes by the irrigator required to complete an operation would be reduced to one. Timely performance of operations would be more easily achievable with higher tool-frame speeds. Energy efficiency of the tool-frame motors could be much higher since they could be sized for optimum loading. It is recommended that future research in the use of irrigation machines as tool-frame reference devices be directed at the use of a linear move system before further resources are employed to improve the spiral mechanization system.

It was apparent in the course of the review of liturature pertaining to automatic guidance in agriculture, that very little work has been done regarding the economic viability of automatic guidance in agricultural machines. Some future research is warranted in the area of automatic guidence system costs, benefits, reliability, and the tolerance of field operations to system malfunctions.

The draft values predicted by the McKyes model of soil forces for the cultivator were found to be significantly different than the measured drafts. Even though the drafts were found to be different, the model did a reasonably good job of predicting the soil forces on a cultivator. Predicted drafts were 101 to 172% of measured drafts. Predictions were thought close enough to make the model useful for predicting the drafts of other soil tine implements in applications in which speeds are low and inertial forces are minimal.

Further validation of this model is recommended over a broader range of tool depths, soil conditions, and tool geometries. The electric tool-frame was a useful power source and instrument platform for these low velocity tests, and could be used again to obtain additional data to verify the McKyes draft model.

References

Agricultural Engineers Standards. 1984. Agricultural machinery data (D230.3). ASAE, St. Joseph, MI 49085.

Alcock, R. 1982. Center pivot mechanization - going round in ever increasing circles. ASAE Paper No. 82-113, ASAE, St. Joseph, MI 49085.

Alcock R. and G. Jahns. 1983. Spiral mechanization via a center pivot system. ASAE Paper No. 83-403, ASAE St. Joseph, MI 49085.

Ambler, B., G. O. Harries, and S. W. R. Cox. 1980. Optical ranging for tractor guidance. ASAE Paper No. 80-1558, ASAE, St. Joseph, MI 49085.

Anon. 1980. The technical and economic feasibility of electric tractors in U.S. agriculture. An institutional and national summary report by researchers at Iowa State Univ., South Dakota State Univ., Rutgers Univ., Virginia Polytechnic Institute, and the University of Georgia.

Brandt, J.A., and B.C. French. 1983. Mechanical harvesting and the California tomato industry: a simulation analysis. American Jounal of Agricultural Economics. 65(2): 265-272.

Brooke, D.W.I. 1968. Off-the wire guidance for leader cable vehicles. ASAE Paper No. 68-819, ASAE, St. Joseph, MI 49085.

Buck, N.L. and H.A. Hughs. 1981. A final report: The feasibility of electric farm vehicles: Grant No. USDA/SEA/CR 59-2511-0-5-005-0. Agricultural Engineering Dept., Virginia Polytechnic Institute and State University, Blacksburg, VA.

Busse, W., H. Coenenberg, F. Feldmann, and T.F. Crusinberry. 1978. The first serial produced automatic steering system for corn combines and forage harvesters. In: GRAIN AND FORAGE HARVESTING. Proceedings of the first international grain and forage conference. ASAE Publication No. 1-78, ASAE, St. Joseph, MI 49085.

Cooper, M.R., G.T. Barton, and A.P. Brodell. 1947. Progress of farm mechanization. USDA Misc. Publ. 630.

Elamin, M. A. 1981. A comparative evaluation of electric- and gasoline - powered garden tractors. Unpublished M.S. Thesis, Iowa State University, Ames, Iowa, 50011.

Fabian, Z. and G. Jahns. 1976. Traktorok automatikus kormanyzasanak lehetosegei szoloultetrenyeken. Mezogazdasagi gepeszet es Epiteszet. (1976) 2:47-49

Fluck, R.C., and C.D. Baird. 1984. Thermodynamic considerations of electric power in agriculture. Transactions of the ASAE 27(4) :1195-1199.

Ford, H. 1922. MY LIFE AND WORK. Garden City Publishing Co. Garden City, New York. pp. 289.

Gilles, E.D., Lenkhardt, P. and Liebmann, M. 1980. Automatic guidance of ships along an inductive pilot cable. AISM IALA, Tokio. No. 12.3.1.

Godwin, R.J. 1974. An investigation into the mechanics of narrow tines in frictional soils. Unpublished Ph.D. dissertation. University of Reading, U.K.: pp.248.

Godwin, R.J. and G. Spoor. 1977. Soil failure with narrow tines. Journal of Agricultural Engineering Research. 22:213-228.

Grisso, R.D., J.V. Perumpral, and C.S. Desai. 1980. A soil-tool interaction model for narrow tillage tools. ASAE Paper No. 80-1518, ASAE, St. Joseph, MI 49085.

Hesse, H. 1974. An automatic impulse controlled implement control system for plant rows. The Journal of Agricultural Engineering Research 19:101-109.

Hettiaratchi, D.R.P. and A.R. Reece. 1967. Symmetrical three-dimensional soil failure. Journal of Terramechanics 4(3):45-67.

Hettiaratchi, D.R.P. 1969. The calculation of passive earthpressure. Unpublished Ph.D. dissertation. University of Newcastle-upon-Tyne.

Hettiaratchi, D.R.P. and A.R. Reece. 1974. The calculation of passive soil resistance. Geotechnique, 24(3)289-310.

Humburg, D., R. Alcock, and J.L. Julson. 1986. A mechanical system for the production and harvesting of green asparagus. ASAE Paper No. 86-503, ASAE, St. Joseph, MI 49085.

Irrigation Jounal. 1987. 1986 Irrigation survey. Irrigation Journal 37(1): 19-26.

Jahns, G. 1976. Automatic guidance of farm vehicles: A monograph. Agricultural Engineering Departmental Series No.1, Agricultural Experiment Station, Auburn University, Auburn Alabama.

Jahns, G. 1983. Automatic guidance in agriculture: A review. ASAE Paper No. 83-404, ASAE, St. Joseph, MI 49085.

Jordan, H.E. 1983. ENERGY EFFICIENT ELECTRIC MOTORS AND THEIR APPLICATIONS. Van Nostrand Reinhold Co., New York. pp 164.

Kanetoh, Y. 1978. Driverless combine harvester. In: GRAIN AND FORAGE HARVESTING. Proceedings of the first international grain and forage conference. ASAE Publication No. 1-78, ASAE, St. Joseph, MI 49085.

Kirk, T.G., G.C. Zoerb, and J.N. Wilson. 1976. A furrow following tractor guidance system. ASAE Paper No. 76-001, ASAE, St. Joseph, MI 49085.

Lawson, R. et al. 1963. Automatic vehicle guidance system. British Patent No. 932 266.

Le Pori, W. A., A. Mizrach, C.R. Harrison, M.D. Evans, I. Shmulevich, and D.B. Carney. 1986. Evaluation of a cable towing system for field machinery operations. Transactions of the ASAE 29(2): 345-350.

Luth, H.J. and R.D. Wismer. 1971. Performance of plane soil cutting blades in sand. Transactions of the ASAE 14(2):255-259,262.

Marchant, W.T.B. and E.T. Chittey. 1966. Automatic control of sugar beet harvester shares. Journal of Agricultural Engineering Research 11(3):188-200.

Martin, M.A., and J. Havlicek Jr. 1977. Some welfare implications of the adoption of mechanical cotton harvesters in the United States. American Journal of Agicultural Economics. 59: 739-744.

McKyes, E. and O.S. Ali. 1977. The cutting of soil by narrow blades. Journal of Terramechanics 14(2):43-58.

McKyes, E. 1985. SOIL CUTTING AND TILLAGE. Elsevier Science Publishers B.V. Amsterdam, The Netherlands. pp.217.

Nielsen, H., Sv. Aa. Christiansen, and S.S. Kofoed. 1977. A split power approach: the M&S tractor system. Jordbrugsteknisk Inst. Royal Veterinary and Agricultural University Tastrup-Denmark, Meddelelse 31.

Parish, R.L. and C.E. Goering. 1970. Developing an automatic steering system for a hydrostatic vehicle. Transactions of the ASAE 13(4): 523-527.

Payne, P.C.J. 1956. The relationship between the mechanical properties of soils and the performance of simple cultivation implements. Journal of Agricultural Engineering Research 1(1):23-50.

Payne, P.C.J. and D.W. Tanner. 1959. The relationship between rake angle and the performance of simple cultivation instruments. Journal of Agricultural Engineering Research 4(4):312-325.

Perry, C.C. and H.R. Lissner. 1955. THE STRAIN GUAGE PRIMER. McGraw-Hill Book Company, Inc., New York. pp 61.

Reece, A.R. 1965. The fundamental equation of earthmoving mechanics. Symposium on Earthmoving Machinery, Institute of Mechanical Engineers, 179,3F, London.

Reece, A.R. 1968. An automatic electric arable farm. Farm Mechanization and Buildings 20(226): 18-20.

Rushing, D. and J.C. Allen Jr. 1969. Sensing device for use with a vehicle adapted to trace a path defined by a conductor. U.S. Patent No. 3 468 391.

Schmitz, A. and D. Seckler. 1970. Mechanized agriculture and social welfare: the case of the tomato harvester. American Journal of Agricultural Economics 52: 569-577.

Stephens, L.E., A.D. Spencer, V.G. Floyd, and W.W. Brixius. 1981. Energy requirements for tillage and planting. ASAE Paper No. 81-1512, ASAE, St. Joseph, MI 49085.

Telle, M.G., and U.D. Perdok. 1979. Field experiments with a leader cable tractor guidance. ASAE Paper No. 79-1069, ASAE, St. Joseph, MI 49085.

USDA 1986. Agricultural statistics. United States Department of Agriculture. United States Government Printing Office. Washington, DC, 20402.

Vik, B. 1985. The electric choremaster: design and performance of an agricultural electric tractor. Unpublished M.S. Thesis, South Dakota State University, Brookings, SD 57007.

Werner, H. 1986. Irrigated crop budgets. Unpublished crop budgets prepared for East Central Irrigators. South Dakota State University, Brookings, SD 57007.

Wismer, R.D. and H.J. Luth. 1972. Performance of plane soil cutting blades in a clay. Transactions of the ASAE 15(2): 211-216.

Young, S.C., C.E. Johnson, and R.L. Schafer. 1981. Evaluation of a microcomputer-based vehicle guidance controller. ASAE Paper No. 81-1604, ASAE, St. Joseph, MI 49085.

Appendix A

LIST OF SYMBOLS

Table A.1. List of symbols and variables used and the associated physical quantity.

Symbol	Corresponding quantity
a	The change in radius associated with an angular increment of 2π radians for an archimedian spiral.
c	The value of soil cohesion, in kPa.
c_a	The value of soil to metal adhesion, in kPa.
d	The depth of operation of the soil tool beneath the soil surface, in meters.
D	The weighted average soil tool depth.
E	The energy used by the spiral mechanization system during an operation. Also, the strain guage bridge excitation voltage.
g	The acceleration due to gravity, in m/s^2.
H	The value of horizontal draft, in kN.
k	The parameter used to describe an archimedian spiral.
L	Used to represent the arclength of a portion of an archimedian spiral.
M	The radius used to calculate percentage of tool-fram time spent turning.
P	The resultant force of the soil on the tillage tool.
R	The resistance across any of several potentiometers used as transducers, in ohms.

Table A.1. (continued)

Symbol	Corresponding quantity
r	The forward distance of soil failure in the Godwin and Spoor draft model. Also, represents the correlation coefficient.
S	The signal from any of several instruments, measured by the data aquisition system in volts.
V	The vertical component of the draft force, in kN.
w	The width of the tillage tool tine, in meters.
Y	The percentage of tool frame running time that was devoted to turns.
α	The soil tool rake angle, in degrees.
γ	The soil bulk density in tons/m^3.
ϕ	The soil internal friction angle, in degrees.
δ	The soil to metal friction angle, in degrees.
θ	The angle associated with some portion of an archimedian spiral.

APPENDIX B

GROUND DRIVE INCREMENT DATA

Table B.1. Radial increments of the ground drive reference system for consecutive revolutions during a planting operation.

Revolution number	Boom radius*	Change	Ground radius**	Change
0	5.41 (m)	-----	5.62 (m)	----
1	7.50	2.09 (m)	7.86	2.45 (m)
2	9.77	2.26	10.32	2.45
3	11.98	2.22	12.67	2.36
4	14.20	2.22	14.95	2.28
5	16.43	2.23	17.23	2.32
6	18.63	2.20	19.49	2.21
7	20.85	2.22	21.76	2.27
8	23.08	2.23	23.95	2.20
9	25.33	2.25	26.26	2.30
10	27.57	2.24	28.58	2.32
11	29.83	2.26	30.64	2.07
mean		2.219		2.275
standard deviation		0.044		0.094

* Radius measured from the pivot center to the descending umbilical cord along the boom.

** Radius measured from the pivot center to the center of the middle plant row.

Table B.2. Radial increments of the ground drive reference system for consecutive revolutions during a cultivation operation.

Revolution number	Boom radius*	Change	Ground radius**	Change
0	5.41 (m)	-----	5.66 (m)	-----
1	7.68	2.27 (m)	7.97	2.31 (m)
2	9.91	2.24	10.46	2.50
3	12.18	2.27	12.92	2.45
4	14.12	1.94*	14.89	1.97*
5	16.39	2.28	19.58	2.37
6	18.67	2.28	19.58	2.33
7	20.93	2.26	21.94	2.36
8	23.20	2.27	24.28	2.34
mean		2.264		2.326
standard deviation		0.012		0.066

* Because of a system malfunction, this increment is not representative of a normal revolution, and was not used in statistical calculations.

Table B.3 Radial increments of the ground drive reference system for consecutive revolutions during operation without an implement.

Revolution number	Boom radius*	Change	Ground radius**	Change
0	5.42 (m)	----	5.06 (m)	----
1	7.64	2.22 (m)	7.36	2.30 (m)
2	9.85	2.22	9.74	2.37
3	12.07	2.22	12.07	2.33
4	14.26	2.19	14.35	2.28
5	16.47	2.21	16.61	2.26
6	18.69	2.22	18.89	2.29
7	20.91	2.22	21.10	2.20
8	23.13	2.23	23.41	2.31
mean		2.216		2.293
standard deviation		0.012		0.050

* Radius measured from the pivot center to a point on the ground beneath the drawbar.

APPENDIX C

DATA USED FOR CALCULATION OF TOOL FRAME RUNNING TIMES, TURNING TIMES, AND REFERENCE SYSTEM RUNNING TIMES

Table C.1 Arclengths, speeds, tool frame running times, and percentage of time spent turning, by revolution, for a planting operation.

Rev.	$Rad._{ave}$	Arclength	Speed	$Time_{tot}$	% Turning
1	6.52 (m)	41.03 (m)	2.35(cm/s)	1746 (s)	26.3 %
2	8.74	54.95	2.60	2114	20.4
3	10.96	68.89	2.62	2629	15.8
4	13.18	82.82	2.78	2979	12.1
5	15.40	96.76	2.65	3651	9.4
6	17.62	110.70	2.67	4146	7.2
7	19.61	124.64	2.63	4740	5.8
8	22.06	138.58	2.69	5152	4.3
9	24.28	152.52	2.66 *	5734	3.3
10	26.50	166.46	2.66 *	6258	2.6
11	28.72	180.40	2.66 *	6782	2.0

Total running time = 45,930 seconds
Time spent turning = 3,290 seconds
Forward running time = 42,640 seconds

* Reliable speed values were not available here. The values used represent the average of the three previous revolutions.

Table C.2 Arclengths, speeds, tool frame running times, and percentage of time spent turning, by revolution, for a cultivation operation.

Rev.	$Rad._{ave}$	Arclength	Speed	$Time_{tot}$	% Turning
1	6.52 (m)	41.03 (m)	2.60(cm/s)	1578 (s)	26.3(%)
2	8.74	54.95	2.84	1935	20.4
3	10.96	68.89	2.95	2335	15.8
4	13.18	82.82	3.44	2408	12.2
5	15.40	96.76	3.08	3142	9.4
6	17.62	110.70	3.17	3492	7.3
7	19.61	124.64	3.20	3895	5.8
8	22.06	138.58	3.24	4277	4.4
9	24.28	152.52	3.31	4608	3.4
10	26.50	166.46	3.00	5549	2.6
11	28.72	180.40	3.18 *	5673	2.0

Total running time = 38,890 seconds
Time spent turning = 2,850 seconds
Forward running time = 36,040 seconds

* Reliable speed values were not available here. The value used represents the average of the three previous revolutions.

APPENDIX D

MEAN VALUES OF VOLTAGE AND CURRENT MEASURED DURING PLANTING AND CULTIVATION OPERATIONS

Table D.1. Mean values of voltage measured during a planting operation.

	Phase 1	Phase 2	Phase 3
all motor combinations	278 v	280	270
standard deviation	0.74 v	0.60	2.52
sample size	23	23	23

Table D.2. Mean values of current measured during a planting operation.

	Phase 1	Phase 2	Phase 3
1 motor*	1.38 Amps	1.27	1.80
std deviation	0.139	0.090	0.250
sample size	8	9	11
2 motors**	3.05	2.46	3.35
std deviation	0.184	0.646	0.254
sample size	9	8	11
3 motors***	3.88	3.16	4.49
std deviation	0.251	0.627	0.278
sample size	11	11	11

* Indicates reference system only, running.
** Indicates tool frame turning.
*** Indicates tool frame running forward.

Table D.3. Mean values of voltage measured during a cultivation operation.

	Phase 1	Phase 2	Phase 3
all motor combinations	276 v	278	276

Table D.4. Mean values of current measured during a cultivation operation.

	Phase 1	Phase 2	Phase 3
1 motor	2.10 Amps	2.06	2.39
std deviation*	0.129	0.184	0.083
2 motors	3.61	3.56	3.82
std deviation	0.126	0.162	0.081
3 motors	5.08	5.14	5.29
std deviation	0.138	0.245	0.118

* Sample size in all of these cases was 9.

Table D.5. Power factors by phase and motor combination.

	Phase 1	Phase 2	Phase 3
1 motor	0.407	0.242	0.156
2 motors	0.309	0.225	0.174
3 motors	0.292	0.208	0.191

APPENDIX E

SAMPLE CALCULATIONS FOR ENERGY EFFICIENCY OF A PLANTING OPERATION

Table E.1. Calculations for energy efficiency of a planting operation.

1. Calculation of energy for each motor combination and phase.

$$E_i = T_i * V_1 * I_{1i} * PF_{1i} + T_i * V_2 * I_{2i} * PF_{2i} + T_i * V_3 * I_{3i} * PF_{3i}$$

Where : E = energy in kWhr.
T = the total time for the system running in the modes subscripted by i, in hours, (from appendix C).
i = the subscript for each of the three possible motor combinations.
1,2,3 = the subscript corresponding to the three phases of the power system.
V = the phase voltage in kvolts, from appendix D.
I = the phase current in Amps, from appendix D.
PF = the power factor, from appendix D.

$$E_1 = .992*.278*1.38*.407 + .992*.280*1.27*.242 + .992*.270*1.80*.156$$

$$E_1 = 0.315 \text{ kWhr}$$

$$E_2 = .913*.278*3.05*.309 + .913*.280*2.46*.225 + .913*.270*3.35*.174$$

$$E_2 = 0.524 \text{ kWhr}$$

$$E_3 = 11.85*.278*3.88*.292 + 11.85*.280*3.16*.208 + 11.85*.270*4.49*.191$$

$$E_3 = 8.657 \text{ kWhr}$$

Table E.1. (continued)

2. <u>Calculation of total energy consumed in the field, and total field energy required per hectare.</u>

   $E_{total} = E_1 + E_2 + E_3 = 9.497$ kWhr

   Energy per hectare = E_{total} / test plot area

   E/ha = 9.497 kWhr/ .2702 ha = 35.1 kWhr/ha

3. <u>Calculation of net heat energy consummed at the power plant, to perform the planting operation.</u>

   E_{net}/ha = E/ha / net power plant efficiency

   E_{net}/ha = 35.1 kWhr/ha / .257 = 137 kWhr/ha

4. <u>Calculation of the work output of the system during the planting operation.</u>

   W/ha = D * L / A

   Where: W/ha = work done per hectare in Joules.

   D = draft force in Newtons.

   L = the path length covered in the plot in meters.

   A = the area of the plot in ha.

   W/ha = 885 * 1217.7 / .2702 = 3.99 MJ / ha

   3.99 MJ / ha = 1.11 kWhr / ha

5. <u>Calulation of overall efficiency for planting.</u>

   Efficiency = E_{out} / E_{in}

   Efficiency = 1.11 kWhr/ha / 137 kWhr/ha = 0.0081

APPENDIX F

SUMMARY DATA FOR DRAFT, SPEEDS, AND TURNING TIMES, FROM PLANTING AND CULTIVATION DATA ACQUISITION FILES

Table F.1. Summary data for a planting operation.

Run Name	Run Radius	Mean Theoretical Speed*	Mean Actual Speed**	Mean Draft	% Time Turning
P2R1	6.28 (m)	3.38 (cm/s)	2.35	764 (N)	30.00%
P2R2	8.60	3.39	2.59	875	21.30
P2R3	10.87	3.39	2.62	912	13.36
P2R4	13.19	3.39	2.64	897	11.64
P2R5	15.40	3.39	2.65	847	9.66
P2R6	17.49	3.39	2.67	883	7.72
P2R7	19.57	3.39	2.63	956	5.83
P2R8	21.86	3.39	2.69	945	4.56

* Theoretical speed does not include stops or turns.
** Actual speed includes turns, but no stops.

Table F.2. Summary data for a cultivation operation.

Run Name	Run Radius	Mean Theoretical Speed*	Mean Actual Speed**	Mean Draft	% Time Turning
C1R1	6.58	3.75	2.60	2275	29.40%
C1R2	8.62	3.75	2.84	2310	16.28
C1R3	11.06	3.75	2.95	____	15.69
C1R4	12.93	3.75	3.44	2352	11.22
C1R5	15.28	3.75	3.08	2462	9.05
C1R6	17.50	3.75	3.17	2501	7.87
C1R7	20.09	3.75	3.20	2171	10.74
C1R8	22.16	3.76	3.24	2279	3.56
C1R9	24.32	3.76	3.31	2257	4.36

* Theoretical speed does not include stops or turns.
** Actual speed includes turns, but no stops.

Table F.3. Summary data for draft model test runs.

Run Name	Mean Theoretical Speed*	Mean Actual Speed	Mean Draft	sample size
C1R12**	3.75 (cm/s)	3.38	980 (N)	83
C1R13	3.75	3.27	1724	92
C1R14	3.75	3.24	2269	95
C1R15	3.74	2.96	3580	97
C1R16	3.72	2.63	5005	95
C1R17	3.71	2.47	5907	86

* Speeds reflect continuous straight-line running.

** Draft for this run was measured with the cultivator tines above the soil surface, and the draft value measured was used as rolling resistance.

APPENDIX G

CULTIVATOR TINE DEPTH, WIDTH, AND DRAFT DATA AND A SAMPLE CALCULATION OF DRAFT

Table G.1. Cultivator tine widths and depths for draft model test runs.

Tine	Width	C1R8	C1R13	C1R14	C1R15	C1R16	C1R17
1	6.35*	6.04	4.45	7.62	7.94	10.48	10.80
2	10.80	4.92	4.45	6.35	2.54	6.99	3.81
3	6.35	6.19	3.18	7.62	6.67	8.57	8.57
4	6.35	3.81	4.13	6.35	9.53	9.53	9.53
5	10.80	5.24	4.13	6.99	7.94	9.53	10.80
6	17.78	4.29	4.13	6.99	4.45	7.94	6.99
7	10.80	4.45	5.08	7.62	6.35	9.53	8.89
8	6.35	5.88	5.72	7.62	9.53	9.53	12.70
9	6.35	3.97	4.13	4.45	7.62	8.89	12.70
10	10.80	4.77	4.13	6.35	6.67	8.89	10.48
11	17.78	5.40	4.13	6.35	5.08	6.99	3.18
12	10.80	5.24	4.76	7.62	7.62	8.89	9.21
13	6.35	5.24	5.08	6.99	7.62	9.53	12.38
14	6.35	6.04	5.08	7.30	6.99	6.99	8.89
15	10.80	5.72	5.08	6.03	4.45	4.45	3.18
16	6.35	6.19	3.81	7.30	8.89	8.89	11.43
Weighted Average Depth		5.13	4.44	6.82	6.39	8.18	8.34

* All measurements given in centimeters.

Table G.2. Cultivator tine drafts for draft model tests.

Tine	C1R8	C1R13	C1R14	C1R15	C1R16	C1R17
1	187 *	113	267	287	460	484
2	110	955	156	47	178	77
3	141	54	196	159	237	237
4	67	75	149	282	285	551
5	118	87	175	212	280	340
6	137	127	253	141	303	253
7	96	114	200	155	280	254
8	132	124	196	285	285	487
9	72	75	84	196	249	487
10	104	87	155	167	251	328
11	179	127	220	166	249	95
12	117	104	200	199	251	269
13	110	105	172	196	272	468
14	136	105	182	172	170	252
15	135	114	142	95	95	61
16	143	68	183	252	249	394

* All draft values given in Newtons.

Table G.3. Sample calculation of draft using the Mckyes model for tine number one of run C1R17.

$$P = (\gamma g d^2 N_G + c d N_c + q d N_q + c_a d N_{ca})w$$

$w = 0.0635$ m
$d = 0.108$ m
$d/w = 1.70$
$\gamma = 1.545$ t/m^3
$g = 9.81$ m/s^2
$c = 7.17$ kPa
$c_a = 3.59$ kPa
$q = 0$
$\phi = 42$ degrees
$\delta = 21$ degrees
$\alpha = 36$ degrees
$N_G = 6.8$
$N_c = 8.5$
$N_{ca} = 1.7$

$P = 0.54$ kN

$$H = P\sin(\alpha + \delta) + c_a d w(\cot\alpha)$$

$H = .48$ kN $= 480$ N

APPENDIX H

CHARTS YIELDING N VALUES FOR THE UNIVERSAL EARTHMOVING EQUATION

(McKyes, 1985)

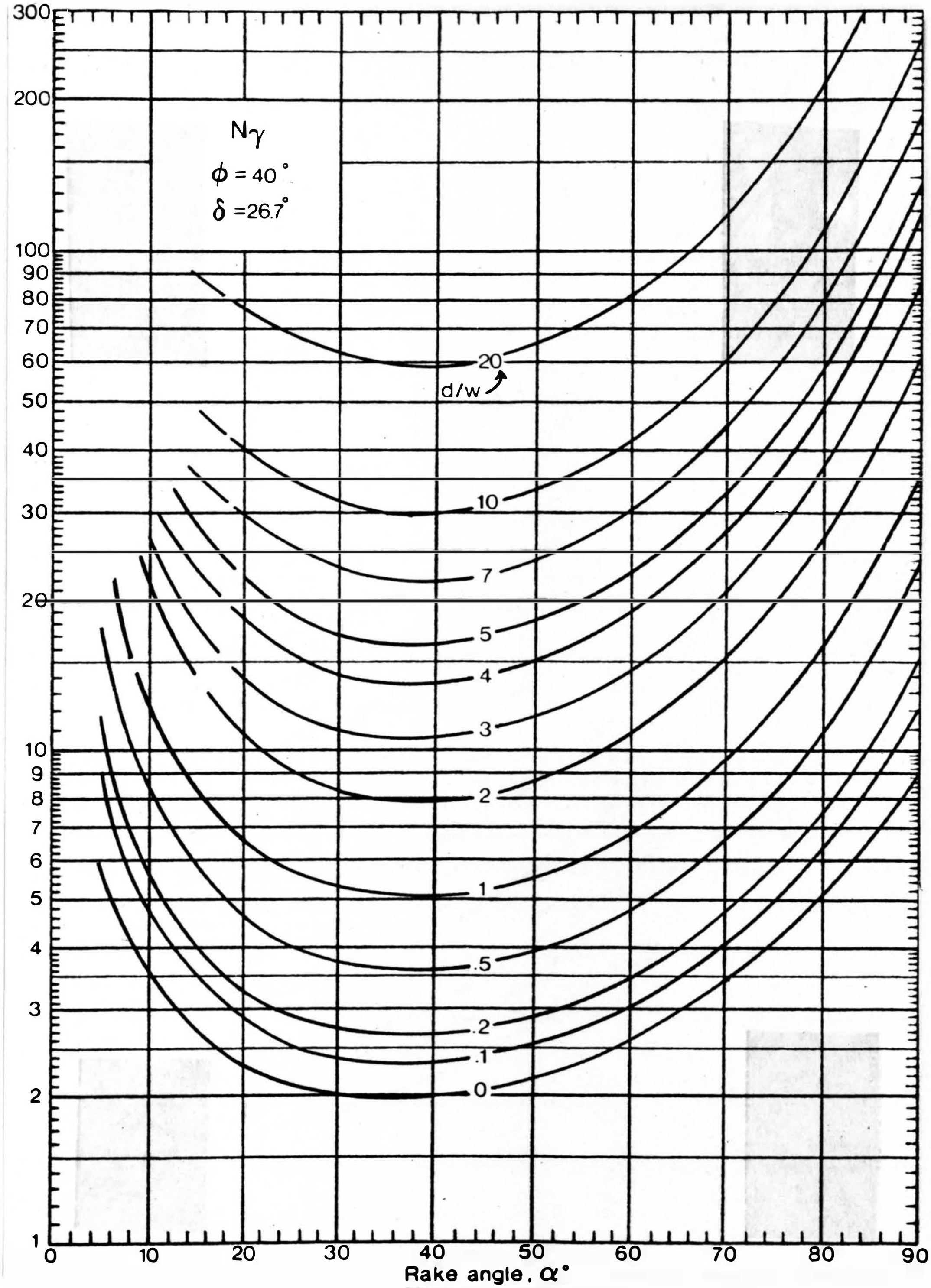

Figure H.1. Plot of N_G for a range of tool aspect ratios and rake angles (McKyes, 1985).

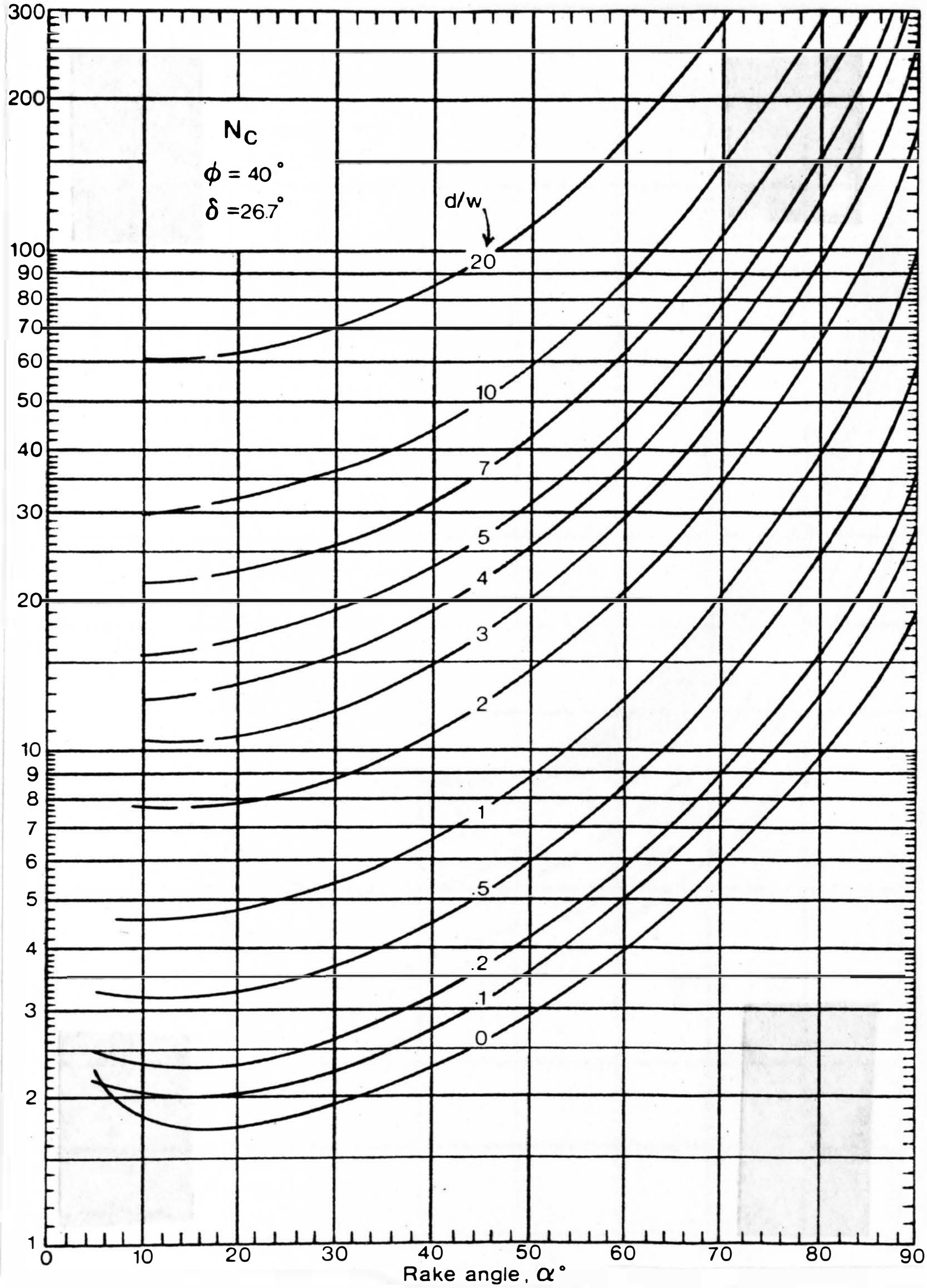

Figure H.2. Plot of N_c for a range of tool aspect ratios and rake angles (McKyes, 1985).

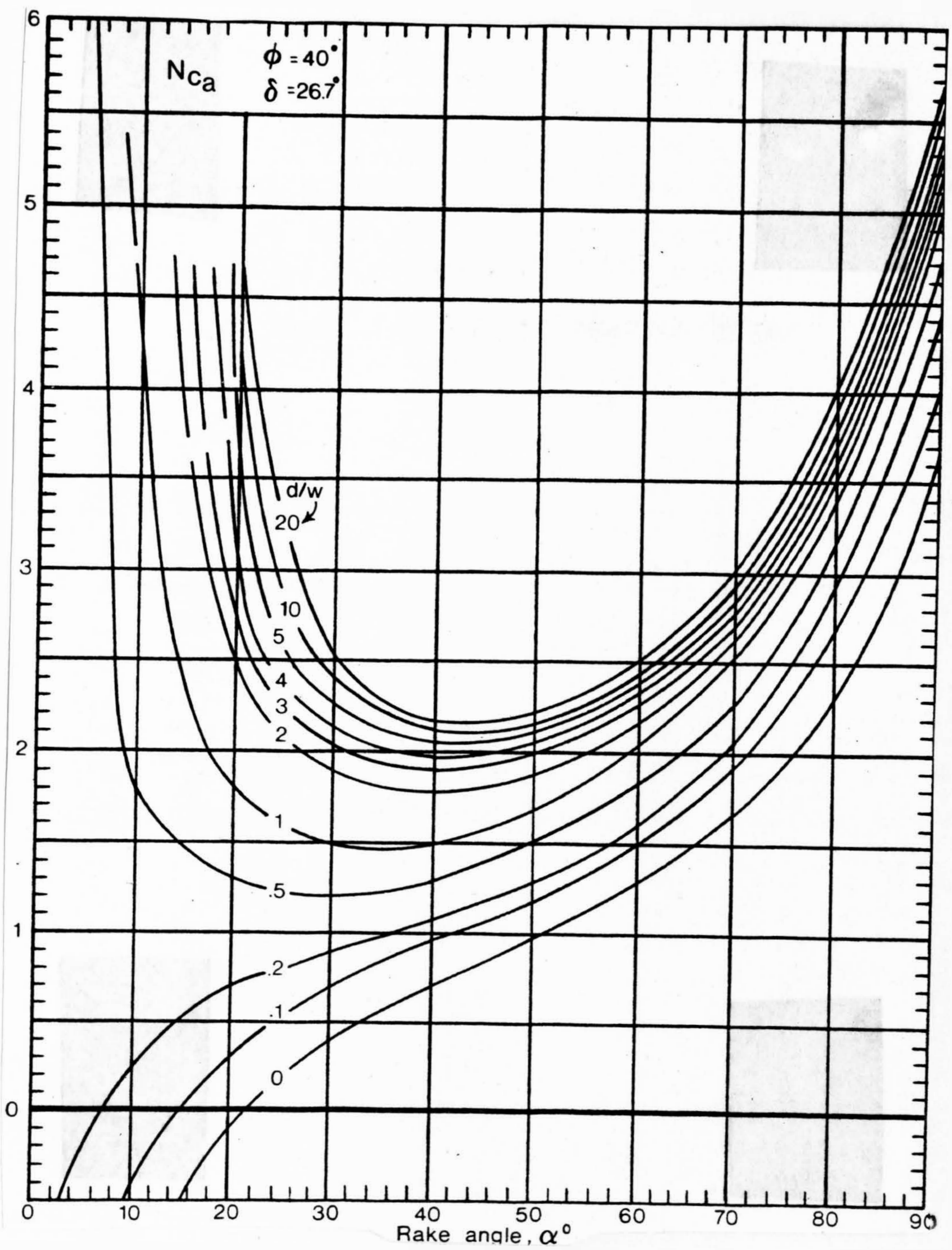

Figure H.3. Plot of N_{ca} for a range of tool aspect ratios and rake angles (McKyes, 1985).

APPENDIX I

INSTRUMENTATION CALIBRATION DATA

Table I.1. Drawbar calibration data.

Draft	Signal Ratio	Draft	Signal Ratio
293 (N)	2.93 (mv/v)	2709	27.98
405	4.12	2958	30.38
556	5.67	3136	32.47
685	6.87	3372	34.87
890	9.26	3598	37.56
1183	11.93	3856	40.37
1383	14.38	3478	35.70
1583	16.54	4199	43.15
1819	19.06	4959	49.87
2202	21.00	7642	80.23
2220	23.00	8362	87.56
2468	25.52	8989	94.14
		9821	101.95

Table I.2 Drive motor speed calibration data and theoretical speed relationship.

Motor Speed	Voltage Signal	Motor Speed	Voltage Signal
1780 (rpm)	4.960 (v)	1780	4.966
1770	4.957	1770	4.946
1770	4.945	1750	4.883
1750	4.901	1740	4.823
1740	4.858	1720	4.807
1720	4.832	1720	4.817
1700	4.758	1700	4.747
1730	4.839	1700	4.741
1720	4.806	1710	4.780
1720	4.815	1790	5.003
1780	4.971	1790	5.003

Groundspeed = (Motor Speed / Gear Ratio) * Rolling Circumference.

Groundspeed = (Motor Speed / 3016) * 342.6 cm

Table I.3. Calibration data for the speed of the press-wheel axle, used to determine actual ground-speed.

Frequency	Voltage	Frequency	Voltage
19 (Hz)	0.486 (v)	19	0.504
47	1.169	48	1.172
75	1.850	76	1.826
88	2.187	89	2.193
105	2.539	104	2.553
117	2.916	118	2.970
131	3.361	132	3.321
146	3.747	145	3.679
169	3.983	167	3.988
179	4.319	181	4.339
193	4.759	196	4.795
210	5.188	207	5.099
227	5.477	224	5.489
242	5.856	243	5.878
246	6.155	245	5.903
230	5.641	231	5.506
215	5.230	215	5.254
198	4.857	198	4.840
187	4.654	183	4.450
166	4.131	165	4.081
153	3.738	154	3.940
136	3.534	137	3.522
122	3.103	123	3.058
111	2.662	110	2.628
95	2.322	92	2.382
81	1.955	81	2.026
64	1.505	64	1.616
50	1.204	50	1.274
37	0.961	38	0.943
24	0.529	26	0.554
11	0.286	10	0.256

Groundspeed = Freq. /60/Gear Ratio * Wheel Circumference.
$Groundspeed_{planting}$ = Freq. /60/26 * 140.2 cm
$Groundspeed_{cultivation}$ = Freq. /60/26 * 125.6 cm

Table I.4. Tool-frame to boom separation calibration data.

Separation*	Resistance	Separation	Resistance
5 (cm)	13.81 (ohms)	35	19.72
10	14.79	40	20.73
15	15.79	45	21.70
20	16.77	50	22.71
25	17.77	55	23.70
30	18.75	60	24.68

* A connecting cable 281.9 cm in length was added to the connect the sensor to the gantry. This length was added as a constant to the equation found from the above data.

Table I.5. Control conduit angle calibration data.

Angle	Resistance	Angle	Resistance
-10 (degrees)	5462 (ohms)	0	3899
-9	5327	1	3744
-8	5173	2	3531
-7	5000	3	3415
-6	4846	4	3257
-5	4652	5	3102
-4	4536	6	2947
-3	4363	7	2812
-2	4208	8	2639
-1	4054	9	2485