AN ABSTRACT OF THE THESIS OF

John Edward Macnab for the degree of Master of Science in Agricultural Engineering presented on March 19, 1976

Title: Modeling the Effects of Tractive Effort on Agricultural Tractor Energy Requirements

Abstract approved: Redacted for Privacy

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A computer model is developed that models the effect of tractive performance on tractor energy requirements. The model is composed of three main segments. The first predicts tractive performance. The variables incorporated in this section include: towed force of wheel (TF), wheel pull (P), wheel torque (Q), dynamic wheel load (W), unloaded tire section width (b), unloaded overall tire diameter (d), wheel rolling radius (r), cone index (CI), and wheel slip (S). Wismer and Luth's (1972) equations for towed and driving wheels are used to separately model, each axle of the tractor. Appropriate tire efficiency terms are derived, transforming drawbar horsepower to axle horsepower for two- and four-wheel drive tractors. A method is also outlined for obtaining dynamic axle weights from the static weight of the tractor.

The second segment of the model deals with predicting tractor fuel consumption. Persson's (1969) method of determining fuel consumption from PTO load and the Nebraska

Tractor Test Reports is the basis for this section.

Segment three of the model is the interface between the tractive performance and fuel consumption portions of the model. Here, axle horsepower is converted to equivalent PTO horsepower; overall gear ratios and required engine speed are determined.

Field tests are conducted for several tractors in order to compare measured and predicted tractor performance. The test procedure and the equipment used are outlined and the results of the tractor tests are shown. Comparisons are made between both measured and predicted tractive performance and fuel consumption.

The example tractor is modeled. The model was then used to show the effect of tractive performance on fuel consumption. To indicate the effect of soil strength on tractor performance, several cone indices are used and the effect on optimum wheel slip is noted.

Modeling the Effects of Tractive Effort on Agricultural Tractor Energy Requirements

by

John Edward Macnab

A THESIS

submitted to

Oregon State University

in partial fulfillment of the requirements for the degree of

Master of Science

Completed March 19, 1976

Commencement June 1976

APPROVED:

Redacted for Privacy

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Date thesis is presented March 19, 1976

Typed by Barb McVicar for John Edward Macnab

ACKNOWLEDGMENTS

I wish to express my sincere thanks to Dr. Robert B. Wensink for his advice, guidance, continuous interest and encouragement during the course of this study. Also to Mr. Dean E. Booster for his suggestions and willing assistance in the preparation of this thesis.

A special thanks to those who so generously allowed the use of their tractors and facilities for test purposes: Mr. Wheeler Calhoun (Hyslop Agronomic Experiment Station), Mr. Clayton Johnson (Farm Services Division of Oregon State University), Mr. Glen Page (Jackson Farm, Oregon State University, and Mr. Donald Macnab (Macnab Company Ranch).

I also wish to thank Mr. Robert D. Wismer, Deere and Company Technical Center, for his assistance in obtaining research material and supplying a cone penetrometer, also Mr. Carl A. Reaves of the National Tillage Laboratory for the use of a cone penetrometer. Appreciation is also expressed to the Linn County Shop for making available their portable truck scales. I would also like to thank Mr. Sid Shannon, Agricultural Engineering Shop, for his excellent work in constructing the fuel flow meter.

Thanks also go to Professor K. W. Domier, University of Alberta, for his help in obtaining research publications. Mr. Michael Kizer and Mr. John Wedman also deserve special recognition for their help in the collection of test data.

A very special thanks is due to my wife Elaine for her understanding, help and motivation.

I wish to express my appreciation to Mr. Pete Lyngstrand for making the drawings and Mrs. Barb McVicar for her excellent typing service.

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I. INTRODUCTION

Problem

An array of factors needs to be considered when modeling the tractive performance and energy requirements of agricultural tractors. Tractor performance depends on the load, operating speed, soil conditions, and the physical design of the tractor. The efficiency at which a tractor converts the fuel energy into usable work is directly affected by the tractor's ability to provide traction when interacting with the soil.

The simplest model of the tractor tire-soil interaction is obtained by assuming a rigid wheel operating on a hard surface; this, however, does not closely approximate field conditions. A pneumatic tire operating on a deformable soil is a very complex model since the geometry of the wheel and soil both change during dynamic operation. An exact model of the pneumatic wheel-soil system has not been accomplished because of its complexity. The best that has been done is to make simplifying assumptions so that a useful solution may be obtained. Though these assumptions allow a workable model for which data may be collected, the accuracy of the results may be limited.

The modeling and subsequent predicting of tractor drawbar performance has long been of interest to both engineers and farmers. The Nebraska Tractor Tests report tractor performance and are a means of comparing different tractor makes and models. The results, however, cannot be directly applied to field conditions to predict performance. It would be impossible to test all tractors under several field conditions and obtain fair, reliable results from which comparisons could be made.

Computer modeling and simulating allow the engineer to do hypothetical field testing. Even though the results obtained from the prediction equations are only estimates, they allow the engineer to determine trends which result from changing the model input parameters. In addition, the effect of tractor and soil parameters on tractive and energy efficiencies can be studied via computer modeling without incurring the cost, time and machinery allocations necessary for field tests.

Purpose and Scope of Study

The study reported herein has several purposes. First, to develop a computer program for modeling tractor tractive and energy requirements. Secondly, to utilize the program to measure the effect of tire efficiency, coefficient of traction and soil strength on energy consumption. The model is designed to use input parameters that are easily obtainable, such as Nebraska test data for fuel consumption, and cone penetrometer readings as a measure of soil strength.

The third objective is to validate the model. Field tests were conducted so that measured and predicted tractor performance could be compared.

Definitions of Terms

The study of terramechanics has many terms which are not found in other disciplines. The meaning of particular terms used in this discipline may differ from the common definition of that term. Even within the discipline a term may have different definitions depending on the particular author. To avert misunderstanding the more important terms are defined in the following list. The majority of the definitions are from Freitag (1965b) and ASAE Recommendation R296.1. ASAE Recommendation R220.3 is also cited for references to the tire selection tables.

Soil Terms

Cone index: A measure of soil strength. The force per unit base area required to push a penetrometer through the soil at a steady rate.

Cone penetrometer: A 30^{o} circular stainless steel cone with driving shaft. The design and test procedure are discussed in ASAE R313.1.

Cohesion (c): The shear strength of a soil at zero normal pressure. It is represented as a parameter in the Coulomb expression, $s = c + p \tan \phi$, relating the shear strength of a soil (s) to the normal pressure (p). (Freitag, 1965b).

Friction angle (ϕ): A parameter in the Coulomb expression, $s = c + p \tan \phi$. It is a measure of the soil shear strength (s) and increases with an increase in pressure (p). (Freitag, 1965b).

Pneumatic Tire Terms

Diameter (d): Unloaded outside tire diameter when inflated to recommended operating pressure. (ASAE R220.3)

Section width (b): Maximum outside width of the inflated, but unloaded, tire cross section. (ASAE R220.3)

Section height (h): The height of the tire, including normal growth caused by inflation, measured from the nominal rim diameter to the highest point on the lug face. (ASAE R296.1)

Loaded section height: Minimum distance from the nominal rim diameter to an unyielding surface for a loaded tire.

Deflection (δ): Change in section height from the unloaded to loaded condition.

Nominal rim diameter: The diameter measured from bead seat to bead seat of the rim.

Static loaded radius: Distance from the center of the axle to the bearing surface for a tire when inflated to recommended pressure, mounted on normal rim and carrying maximum recommended load. (ASAE R296.1, ASAE R220.3)

Rolling radius (r): Forward advance per revolution of the loaded tire when towed on a plane, level, unyielding surface, divided by 2π. It is related to the tire diameter and the deflection. (Freitag, 1965b).

Tire-Soil System Terms

Coefficient of traction: Ratio between drawbar pull and dynamic weight on the traction devices. Also referred to as traction coefficient or dynamic traction ratio. (Some authors use static weight in place of dynamic weight.)

Coefficient of rolling resistance: Ratio between rolling resistance and dynamic weight on the traction devices.

Drawbar pull: Force in the direction of travel produced by the vehicle at the drawbar. (ASAE R296.1)

Wheel pull: Force in the direction of travel produced by the wheel.

Dynamic weight: Total force normal to the plane of the undisturbed supporting surface, exerted by the traction or transport device under operating conditions. (ASAE R296.1)

Static weight: Total force normal to the plane of the undisturbed supporting surface, exerted by the traction or transport device while stationary on level ground with zero pull and zero torque. (ASAE R296.1)

Weight transfer: The change in normal forces on the traction and transport devices of the vehicle under operating conditions, as compared to those for the static vehicle on a level surface. (ASAE R296.1)

Wheel load: The vertical force applied to the tire through the axle. (Freitag, 1965b)

Towed force: The pull required to tow the wheel with zero torque at the axle. (Freitag, 1965b)

Travel ratio: Ratio of the actual rate of wheel advance to the theoretical rate of advance. The theoretical rate of advance is defined as $r\omega$, where r is the rolling radius and ω is the angular velocity of the wheel. (Freitag, 1965b)

Slip: Unity minus the travel ratio. (Freitag, 1965b)

Slip: Relative movement in the direction of travel at the mutual contact surface of the traction device and the surface which supports it. (ASAE R296.1)

Zero conditions: Zero conditions may be those of zero net traction, or zero torque for the traction device, as well as zero drawbar pull for the vehicle. Other zero conditions might also be used. The specified zero conditions should always be stated. (ASAE R296.1)

Sinkage: The depth to which the tire penetrates the soil (measured relative to the original soil surface).

II. REVIEW OF LITERATURE

Studies of Soil

When predicting tractive performance, the most important factor is the soil. Bekker (1956) listed four soil characteristics:

1. Soils generally exhibit a plastic behavior to a degree; that is, they tend to deform permanently without fracture.
2. Soils are generally compressible in the surface region.
3. Agricultural soils vary from almost pure sand to soils very high in clay and/or organic content.
4. Soils, even within a small area, will be far from homogeneous both vertically and laterally.

Freitag (1965a) identified four soil groupings based on the effect of load on soil strength.

1. <u>Nonfrictional</u>. Soils in which the strength does not change with load. They have only a cohesive strength component. An example is a wet, saturated clay.
2. <u>Frictional</u>. Soils in which the strength increases reversibly under load. They have only a frictional strength component. Dry sand is a good example.
3. <u>Sensitive</u>. Soils in which the strength decreases irreversibly under load. They are only found in

undisturbed soils and the strength loss is from the destruction of the natural soil structure by the applied load. This soil type is usually found in very wet silty or clayey soils.

4. Compactible. Soils in which the strength increases irreversibly under load. In general they have cohesive properties, but are not highly saturated. These soils are commonly described as being workable. Soils in this class are usually partially saturated clays and loams.

Compactible soils are not well understood in regard to loading and to acquired strength. They range from frictional to nonfrictional conditions, but fortunately, they are much more trafficable than either group. Compactible soils are most often encountered in agriculture, with the extreme condition being those soils falling in the nonfrictional range.

Much military research has concentrated on the extreme conditions of nonfrictional and frictional soils to improve tractive performance under adverse conditions.

Studies of Tire-Soil Interactions

Several researchers have studied the performance of wheels operating on soil. The rotary energy available at the drive axle was transformed by the wheel to translational energy to produce work. Not only was the efficiency at which the tire accomplished this energy transformation important, but also, was the effect the tire had on the soil and the plant life environment.

Vanden Berg, et al., (1961), analyzed the forces acting on a rigid wheel operating on soil. The performance of the wheel was clearly related to the magnitudes of the forces and the relationships between the different forces. A scheme of forces was developed for a wheel acting as either a transport device (the wheel being towed over the soil) or a traction device.

Persson (1967) defined a set of basic wheel-performance parameters from which the remaining parameters could be derived. The basic set of wheel-performance parameters consisted of one traction parameter, one resistance parameter, one velocity-reduction parameter, and the zero-pull rolling radius.

Many things affected the performance of a tractor tire. The physical properties such as diameter, width, operating pressure, allowable load and tread design all had an effect on the tire performance. In 1938 the Society of Automotive Engineers Co-operative Tractor Tire Testing Committee (1938) concluded that the traction of pneumatic tractor tires was affected as follows:

1. The most important factor affecting the drawbar performance was the soil itself.
2. For a given soil, the most important factor affecting drawbar pull was the weight that the tire carried.
3. Tractors with higher horsepower-to-weight ratios had to travel faster to utilize the available horsepower or use added weights to operate at lower speeds.
4. Inflation pressure had an effect; lower pressures were advantageous on loose, sandy soils. This advantage disappeared on firmer soils.

Even though these same conclusions hold true today, more is known about the relationships and interrelationships of the physical tire properties. Kliefoth (1966) reported from studies of German and French tires that tire treads with open centers had no clearly measurable influence on trafficability on a soil with good plowing conditions. Also, the tire load required to give a certain pull varied with the kind and condition of the soil. The traction-coefficient, the ratio of pull to the load on the tire, decreased when the load on the tire was increased on soils with a poor bearing capacity. On soils with a good bearing capacity the traction-coefficient increased with increased tire load. The traction-coefficient remained nearly constant with tire load on the group of soils between these extremes. Increasing the diameter of the tire increased the traction-coefficient, but the increase was not directly proportional to the diameter. Decreasing tire inflation pressure caused a non-linear decrease in the traction-coefficient. A slight increase in the traction-coefficient was reported by increasing tire width.

Taylor, et al., (1967), tested the effect of tire diameter on the performance of powered tractor wheels at the National Tillage Machinery Laboratory. The tires tested were all 12.4/11 of 24, 36 and 42-inch rim diameters. These tires were first tested with lugs, and then with lugs removed. The tires were tested without the lugs to eliminate the effect of tread wear. Three steel wheels, 12 inches wide, with outside diameters of 40, 50 and 60 inches, fitted with lugs, were also tested to eliminate the effects of inflation pressure and deflection. Measuring the effect of diameter was complex. Tires of differing diameters carrying the same weight must have either varying inflation pressure, or deflection, or both.

The pneumatic tires were tested on concrete and on several soils. In each test, the tire diameter and one additional parameter were varied while the other two parameters were held constant. A series of data points curves were drawn for each soil condition and tire diameter. From their results, Taylor, et al., (1967), concluded that for the same load and inflation pressure, increasing the diameter generally increased the pull and the coefficient of traction for pneumatic tires. The greatest improvement in pull was achieved by increasing the tire diameter while additional load was applied to maintain the same tire deflection, since a larger tire was capable of carrying a greater load. Increasing inflation pressure for constant load and diameter gave a decrease in pull. Finally, the largest deviation in pull arose from the differences in the soil or traction conditions. Söhne (1969) reported that increasing wheel diameter was more advantageous than increases in wheel width. The widening of the tire, without increasing inflation pressure, does not give as consistent results as does increasing the tire diameter.

Model studies were conducted by Clark and Liljedahl (1969) on the performance of single, dual and tandem wheels at the Purdue University traction testing facility. Two tire sizes were tested, 4.00-8 and 6.00-12. The tires were smooth to remove the effect of tread on tire performance. The total vertical load for each of the single, dual and tandem-wheel arrangements were equal. That is, if a single tire was tested at load, x, then each of the wheels in the tandem or dual wheel arrangement were tested at one-half load or x/2. Each tire configuration was tested on three different artificial soil conditions which were classified as loose, medium firm and firm soil. All the soils were frictional in nature.

From their tests they concluded that dual tires performed better than single tires having the same total vertical load in loose soil for travel reductions less than 30 percent. Dual tires did not reach their full advantage unless the inflation pressure was reduced below that of a single tire. Tandem tires out-performed equal-sized single tires for all the soil and loading conditions used in their investigation. The tandem tires did not consistently show an advantage over low-pressure dual tires. Wheel sinkage was reduced with dual and tandem tires for all the soil conditions tested.

Melzer and Knight (1973) studied the effect of duals and their spacing on wheel performance in sand. They observed that the performance of the dual wheel system decreased with increasing wheel spacing until the performance level of a single wheel was obtained. They called this spacing the critical spacing and found that it was a function of the tire width. Maximum performance was obtained at zero spacing with little decrease in performance until the tire spacing to tire width ratio was approximately one-half.

From their tests they discovered that a dual-wheel configuration at zero spacing, considered as one wheel, developed nearly as much pull as a single tire with double the width of a single dual. Also, they found the dual-wheel configuration to be more efficient.

Taylor (1975) conducted tire tests with three different tire tread designs. He considered the standard agricultural tractor tread (R-1), shallow tread (R-3), and industrial tractor or intermediate tread (R-4). In good conditions all three tires performed equally. The R-1, however, was superior in extremely difficult field conditions.

Traction Models

No completely successful method has been developed for predicting wheeled-vehicle performance on soil. Freitag (1965b) derived dimensionless terms for tire performance analysis on soft soils, i.e., wet, frictionless clay and dry, cohesionless sand. His results showed that a cone penetrometer was an acceptable means of determining a single soil parameter to use in a soil-wheel model. Freitag (1965b) defined a mobility number as follows:

$$\beta = \frac{Cbd}{W}\sqrt{\frac{\delta}{h}}\left(\frac{1}{1 + \frac{b}{2d}}\right)$$

where:

β = clay mobility number

C = mean cone penetrometer resistance through the top 250 mm of soil

b = tire section width

d = tire undeflected diameter

W = vertical load carried on the tire

δ = tire deflection under load

h = tire section height

The following relationship between coefficient of traction and the mobility number was later developed to be used at 20 percent slip:

$$\frac{P_{20}}{W} = 0.80 - \frac{1.31}{\beta - 2.45}$$

where:

P_{20} = pull at 20 percent slip

β = mobility number

These equations were derived for nonfrictional soils, i.e., pure clay. Freitag (1965b) also developed a mobility number for frictional soil, i.e., pure sand. The clay mobility number has been studied on agricultural soils by other researchers including Wismer and Luth (1972) and Dwyer and Pearson (1975).

A graphical solution for predicting two-wheel drive tractor drawbar performance was presented by Zoz (1970). The graph was based on average tire performance for single tires on concrete and three selected soil types.

Since the method of attaching the load to the tractor will effect the tractor performance, Zoz (1970) used three average weight transfer coefficients of 0.65 (integral), 0.45 (semi-integral) and 0.25 (towed) to determine the dynamic weight on the drive wheels. Only the following four parameters were required to use the graph: the drawbar horsepower, percent slip on concrete, the gear, and the no-load advertised speed from the Nebraska Tractor Tests Reports. The graph was only applicable for two-wheel drives with single tires and was very general.

Wismer and Luth (1972) developed traction equations for agricultural soils. They defined a wheel numeric similar to Freitag's clay mobility number, but for constant values of $\delta/h = 0.20$ and $b/d = 0.30$.

$$Cn = \frac{CIbd}{W}$$

where:

Cn = wheel numeric

CI = cone index, mean penetrometer resistance through top 150 mm of soil

W = dynamic wheel load, normal to soil surface

b = unloaded tire section width

d = unloaded overall tire diameter

δ = tire deflection under load

h = tire section height

Wismer and Luth's equations for towed and powered wheels are:

Towed wheel:

$$\frac{TF}{W} = \frac{1.2}{Cn} + 0.04$$

Powered wheel:

$$\frac{P}{W} = 0.75\left(1 - e^{-0.3CnS}\right) - \frac{1.2}{Cn} + 0.04$$

where:

TF = towed force of wheel, parallel to soil surface

P = wheel pull, parallel to soil surface

e = base of natural logarithms

S = wheel slip

Freitag (1965b) and Wismer and Luth (1972) both defined zero slip as the condition when the vehicle was operating on a hard surface with zero drawbar load. The more common definition of zero slip was the condition of zero drawbar pull on the surface where the tests were being made. Requiring the zero slip condition to be measured on a hard surface gave a fixed base from which to compare tractor performance for different soil conditions.

Johnson (1975) derived predictive equations from Freitag's (1965b) wheel performance data. He also took data from Robinson's, et al., (1969), tests with a log skidder to check the actual vs. predicted results.

Fiske (1973) applied Wismer and Luth's (1972) traction equation to log skidders by replacing their wheel numeric, Cn, with Freitag's clay numeric, Cm.

where:

$$Cn = \frac{CIbd}{W} \quad \text{and} \quad Cm = \frac{CIbd}{W}\left(\frac{\delta}{h}\right)^{\frac{1}{2}}$$

This addendum accounted for the change in tire deflection of a loaded and unloaded log skidder.

Tractor Field Tests

Many people have done tractor field testing. Much of the information was of limited value to others because the complete results were not printed and the physical characteristics of the tractor were not given.

Friesen, et al., (1967, 1968, 1969) tested and compared tractors equipped with singles, duals and four-wheel drive. Southwell (1967) field tested conventional, four-wheel drive and tandem tractor arrangements. Dwyer, et al., (1974), used Freitag's mobility number in evaluating tire performance data obtained with the National Institute of Agricultural Engineering MK II Single Wheel Tester.

Dwyer and Pearson (1975) compared the tractive performance of two- and four-wheel drive tractors. They modeled the four driving wheels of a four-wheel drive tractor as two driving wheels, each of width equal to the average width of the front and rear wheels, and the diameter equal to the sum of the diameters of the front and rear wheels.

Fuel Consumption Models

The Nebraska Tractor Tests are the most widely used source for comparing tractor fuel consumption. The test procedures are those given in the agricultural tractor test code, ASAE Standard S209.4. Since the tests are conducted by an impartial organization, the results are accepted by both tractor manufacturers and farmers. To obtain results that can be validated by replication, the varying drawbar and fuel consumption data must be obtained on a hard surface. Since all the tractors are tested on the same unchanging surface, comparisons can be made between tractor makes and models. The data, however, cannot be directly applied to a tractor operating in the field. The important thing to remember when evaluating the fuel consumption data is that the tractor could be operating in the field, on a concrete surface, or on a dynamometer and the output at the axles will be the same. There is some constant relationship between axle horsepower and fuel consumption, but the relationship between fuel consumption and drawbar horsepower is a function of the efficiency of the tractor tire on the surface on which it is operating.

Sulek and Lane (1968) used the Nebraska PTO Varying Power and Fuel Comsumption data to derive fuel consumption equations for diesel, gasoline and propane powered tractors. From their data analysis, they observed that gasoline and propane powered tractors show a continuous increase in fuel economy as the load in increased. Diesel tractors fuel economy peaked at 85 percent load and then dropped. They concluded that the maximum PTO power fuel economy of a tractor was a reasonably good indicator of the heavy-load portion of the PTO varying load test. The variation in the light-load fuel economy of all fuel classes could not be explained by variations in the maximum power fuel economy.

Persson (1969) developed an empirical relationship between fuel consumption and power. He reported that his equation would predict "reasonably well" tractor fuel consumption at varying loads and engine speeds between 60 and 100 percent maximum rpm. The equation for fuel consumption was:

$$F_h = \frac{1}{H\alpha}\left(2544\ bhp_p + \frac{cV_dN^2}{(2)(155600)}\right), \quad lb/hr$$

where:

F_h = fuel consumption, lb/hr

H = net heat value of fuel, btu/lb

bhp_p = engine horsepower as measured at PTO shaft

V_d = engine displacement, in^3

N = engine speed, rpm

The coefficients α and c can be determined from Nebraska Test data and possibly reflect the thermal efficiency of the engine and mechanical power loss, respectively.

III. MODEL DEVELOPMENT

The complete model may be divided into three main parts. The first deals with predicting the tractor tractive performance. The physical characteristics of the tractor, drawbar load and soil conditions all need to be included. Secondly, the model predicts tractor fuel consumption from the engine load, speed and characteristics. The third section deals with combining the tractive performance and fuel consumption models and determining their interrelationships.

Tractive Performance

To predict tractive performance several physical tractor parameters are considered. These parameters are tire size and number, wheel load, axle torque, wheelbase length and drawbar height. Also, a measure of soil strength is included.

Wismer and Luth's (1972) traction equations for pneumatic tires operating on soils with both frictional and cohesive properties are the basis for this part of the computer model. Nine variables are incorporated in these equations: towed force of wheel (TF), wheel pull (P), wheel torque (Q), dynamic wheel load (W), unloaded tire section width (b), unloaded overall tire diameter (d), wheel rolling radius (r), cone index (CI), and wheel slip (S).

The equations predict the forces acting on both towed and driving wheels.

Each front wheel of a conventional two-wheel drive tractor is modeled by the equation of a towed wheel. A wheel located on an unpowered axle is considered a towed wheel. Axle torque is assumed to be zero by neglecting bearing friction. Wheel load, tire size, inflation pressure and soil strength determine the towed force or rolling resistance of the towed wheel. Wismer and Luth's (1972) equation for a towed wheel is limited to tires operated at "nominal tire inflation pressures." "Nominal tire inflation pressure" is defined as that pressure which produces tire deflections of approximately 20 percent of the undeflected section height. The equation for towed force is developed for tires with a tire width/diameter ratio (b/d) of approximately 0.3. The towed force of a front wheel, or its rolling resistance, is predicted from:

$$\frac{TF}{W} = \frac{1.2}{Cn} + 0.04 \tag{1}$$

where:

TF = towed force of wheel, lb

W = dynamic wheel load, lb

Cn = wheel numeric

Multiplying both sides of this equation by the dynamic load (W) on the wheel predicts the rolling resistance of the wheel. The wheel numeric (Cn) varies with the soil cone index, tire section width, overall tire diameter and dynamic load. On a firm soil with a high cone index the value of Cn is very large and the towed force of the wheel is equal to four percent of the dynamic wheel load.

The rolling resistance of the front axle of a conventional two-wheel drive tractor is the sum of the towed forces for the two front wheels. The dynamic wheel load for each wheel is one-half of the dynamic weight on the front axle. The total rolling resistance of the front axle is then written as:

$$FRR = 2 \times \frac{FWD}{2} \left(\frac{1.2}{CNF} + 0.04 \right) \text{, lb} \tag{2}$$

or

$$FRR = FWD \left(\frac{1.2}{CNF} + 0.04 \right) \text{, lb} \tag{3}$$

where:

FRR = rolling resistance of front axle (unpowered), lb

FWD = front axle dynamic weight, lb

CNF = front wheel numeric

$= \dfrac{CIbd}{FWD/2}$

Similarly, the driving wheels of both conventional two-wheel drive and four-wheel drive tractors are modeled by Wismer and Luth's equation for driving wheels:

$$\frac{P}{W} = 0.75 \left(1 - e^{-0.3CnS} \right) - \left(\frac{1.2}{Cn} + 0.04 \right) \tag{4}$$

where:

P = net wheel pull, lb

S = wheel slip (decimal form)

W,Cn = same as equation 1

This equation is derived from tires with $b/d \approx 0.30$ and tire deflection/section height ratio (δ/h) limitation of $\delta/h \approx 0.20$. The wheel pull (P) parallel to the soil is obtained by multiplying both sides of the equation by the dynamic wheel load. The equation is formed by two steps. The first step predicts the gross pull developed by the wheel. The second step is the rolling resistance of the wheel and is subtracted from the gross pull to yield the net pull of the wheel. On firm soil surfaces with a large cone index the gross wheel pull is equal to 75 percent of the dynamic wheel weight. The net wheel pull is then the gross wheel pull minus four percent of the dynamic wheel load, which accounts for the wheel rolling resistance.

The net output of a powered axle is then determined by the sum of the net pulls of its driving wheels. The net pull of a driving axle is then written as:

<u>front wheel drive</u>

$$FP = N\left[\frac{FWD}{N}\, 0.75 \left(1 - e^{-0.3(CNF)(FSLIP)}\right) - \frac{FWD}{N}\left(\frac{1.2}{CNF} + 0.04\right)\right] \text{, lb}$$

or:

$$FP = 0.75\ FWD \left(1 - e^{-0.3(CNF)(FSLIP)}\right) - FWD\left(\frac{1.2}{CNF} + 0.04\right) \text{, lb} \qquad (5)$$

rear wheel drive

$$RP = 0.75\ RWD\left(1 - e^{-0.3(CNR)(RSLIP)}\right) - RWD\left(\frac{1.2}{CNR} + 0.04\right),\ lb \qquad (6)$$

where:

FP = net front axle pull, lb

RP = net rear axle pull, lb

FWD = front axle dynamic weight, lb

RWD = rear axle dynamic weight, lb

CNF = front wheel numeric

CNR = rear wheel numeric

N = number of wheels per axle

FSLIP = front wheel slip (decimal form)

RSLIP = rear wheel slip (decimal form)

A complete tractor is composed of varying arrangements of towed and driven axles. The drawbar pull developed by the tractor is predicted by summing the forces acting on each axle. A conventional two-wheel drive tractor is then modeled as:

$$DBP = RP - FRR,\ lb \qquad (7)$$

where:

DBP = drawbar pull, lb

RP = net pull developed by rear axle, lb

FRR = front axle rolling resistance, lb

Similarly, the drawbar pull for a four-wheel drive tractor is expressed as:

$$DBP = FP + RP,\ lb \qquad (8)$$

where:

FP = net pull developed by front axle, lb

The equations for rolling resistance and wheel pull require that the dynamic wheel load or dynamic axle load be known. The dynamic axle weight varies due to weight transfer when the tractor is pulling. The amount of weight transfer is subtracted from the front static axle weight to determine the dynamic front axle weight. Likewise, the weight transfer is added to the rear axle static weight to yield the dynamic rear axle weight.

This study is limited to tractor drawbar loads that are applied horizontally and parallel to the drawbar. In addition, this study assumed the resultant forces on the tractor wheels act at points directly under the axles. Then the dynamic weight on both front and rear axles can be determined if the static weights, wheelbase length, drawbar height and drawbar pull are known. From Figure 1 the equations for the dynamic front and rear axle weights, respectively, are:

$$FWD = FWS - DBP \left(\frac{DH}{WB}\right) \text{, lb} \tag{9}$$

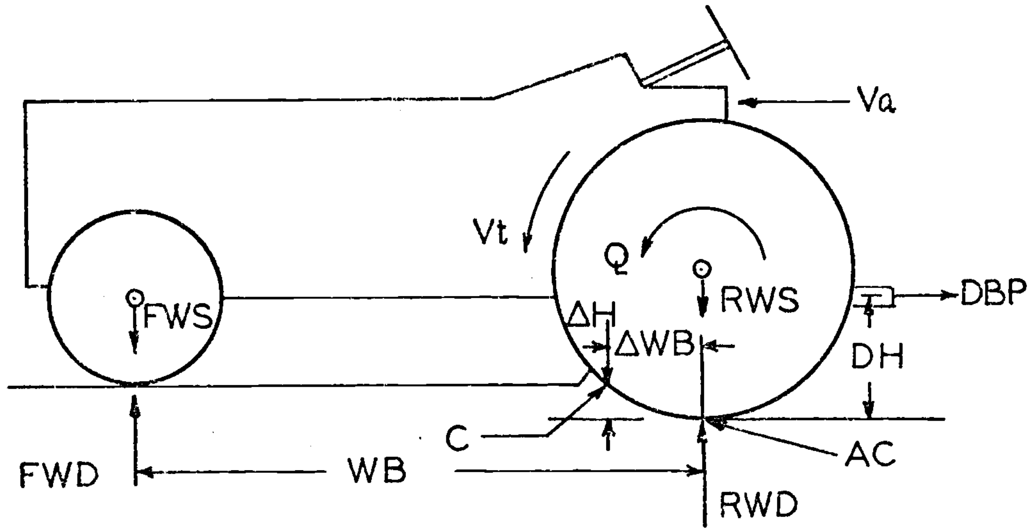
and

$$RWD = RWS + DBP \left(\frac{DH}{WB}\right) \text{, lb} \tag{10}$$

where:

FWS = static front axle weight, lb

FWD = dynamic front axle weight, lb



$$FWD = FWS - DBP\left(\frac{DH}{WB}\right)$$

$$RWD = RWS + DBP\left(\frac{DH}{WB}\right)$$

C = CENTROID OF CONTACT AREA
AC = ASSUMED CENTROID OF CONTACT AREA
ΔH = ERROR IN DRAWBAR HEIGHT
ΔWB = ERROR IN WHEELBASE LENGTH

Figure 1. Assumed tractor weight distribution.

RWS = static rear axle weight, lb

RWD = dynamic rear axle weight, lb

DH = drawbar height, in

WB = wheelbase length, in

For equilibrium the amount of weight transfer, $DBP\left(\frac{DH}{WB}\right)$, must be the same for both axles. The assumption of the location of the resultant forces on the wheels may not be correct, but the error is small. The centroid of the tire contact area will be, in most cases, slightly forward of the axle, thus reducing the effective wheelbase length (see Figure 1). Also, the drawbar height is slightly reduced due to the movement of the centroid of the tire contact area. Since both the numerator and denominator of the weight transfer expression are reduced, the error in weight transfer is expected to be small.

The problem is that the dynamic axle weights and drawbar pull are unknown. Neither can be solved directly without knowing the other. An iterative procedure was developed that allows both unknowns to be solved. The first step is to assume no weight transfer. Dynamic axle weights are replaced with static axle weight and either equation (7) or (8) is solved for drawbar pull. This drawbar pull is then inserted into equations (9) and (10) to compute dynamic axle weights. These dynamic axle weights are used to calculate a new drawbar pull which, in turn, is used to calculate new dynamic axle weights. After several iterations the change in drawbar pull and axle weights becomes negligible. The predicted values of drawbar pull and dynamic axle weights are now known.

The equations for drawbar pull also include the slip of the driving wheels. Wismer and Luth (1972) define slip as:

$$S = \left(1 - \frac{Va}{Vt}\right) \tag{11}$$

where:

S = wheel slip (decimal form)

Va = actual travel speed

Vt = theoretical wheel speed, $r\omega$

r = rolling radius of wheel on hard surface

ω = angular velocity of wheel

The zero slip condition is defined as a self-propelled operation on a hard surface with no drawbar load. For the tractor model, all drive wheels operating on one axle are assumed to have the same slip. The front and rear axles of a four-wheel drive tractor, though, may have differing slips.

Wismer and Luth also define the torque of a driving wheel as the gross pull acting through the moment arm, (r). The driving wheel torque (Q) can be expressed as:

$$Q = 0.75 \left(1 - e^{-0.3CnS}\right) rW, \text{ lb in} \tag{12}$$

The tractive efficiency of the driving wheels is defined as the ratio of the output power to input power. The output power of a wheel is the product of the net pull and actual travel speed. The input power is the product of wheel torque and wheel angular velocity. Wismer and Luth's expression for tractive efficiency of a wheel is:

$$TE = \left\{ 1 - \left[\frac{\frac{1.2}{Cn} + 0.04}{0.75\left(1 - e^{-0.3CnS}\right)} \right] \right\} (1 - S) \qquad (13)$$

where:

TE = tractive efficiency

Similarly, an efficiency term may be defined for the tractor as a unit. Tractor tractive efficiency is the ratio of drawbar horsepower to axle horsepower. For a two-wheel drive tractor the efficiency is:

$$TTE = \left\{ 1 - \left[\frac{\left(\frac{1.2}{CNR} + 0.04\right) + \frac{FWD}{RWD}\left(\frac{1.2}{CNF} + 0.04\right)}{0.75\left(1 - e^{-0.3(CNR)(RSLIP)}\right)} \right] \right\} (1 - RSLIP) \qquad (14)$$

where:

TTE = tractor tractive efficiency

CNR = rear wheel numeric

CNF = front wheel numeric

The tractive efficiency for a four-wheel drive tractor can be expressed as:

$$TTE = \frac{DBP}{\frac{GPF}{1 - FSLIP} + \frac{GPR}{1 - RSLIP}} \qquad (15)$$

where:

DBP = drawbar pull, lb

GPF = gross front axle pull, lb

$= 0.75 \text{ FWD} \left(1 - e^{-0.3(\text{CNF})(\text{FSLIP})}\right)$

GPR = gross rear axle pull, lb

$= 0.75 \text{ RWD} \left(1 - e^{-0.3(\text{CNR})(\text{RSLIP})}\right)$

FSLIP = front wheel slip (decimal form)

RSLIP = rear wheel slip (decimal form)

Wheel tractive efficiency and tractor tractive efficiency are important terms in determining the power the tractor's engine must develop to produce a given amount of drawbar horsepower. For a given set of operating conditions, the axle horsepower a two-wheel drive tractor needs to develop is:

$$\text{AHP} = \frac{\text{RP}}{\text{TE}} \frac{\text{V}}{375} \tag{16}$$

where:

AHP = axle horsepower

RP = net rear axle pull, lb (equation 6)

V = tractor speed, mph

TE = tire efficiency (equation 13)

The axle horsepower required by a four-wheel drive tractor is:

$$\text{AHP} = \frac{\text{DBP}}{\text{TTE}} \frac{\text{V}}{375} \tag{17}$$

where:

DBP = drawbar pull, lb (equation 8)

TTE = tractor tire efficiency (equation 15)

According to ASAE Agricultural Machinery Data D230.2, the ratio of axle horsepower to PTO horsepower is approximately 0.96 to 1. An estimate of PTO horsepower, therefore, is:

$$\text{PTO HP} = \frac{\text{AHP}}{0.96}\ , \text{ hp} \tag{18}$$

Fuel Consumption

Some of the engine parameters important in predicting fuel consumption are engine speed, displacement, thermal and mechanical efficiency and horsepower output. The empirical relationships derived by Persson (1969) are the foundation for this section of the model. Part load and varying speed fuel consumption can be predicted from the following equation:

$$F_h = \frac{1}{H\alpha}\left(2544\ \text{bhp}_p + \frac{cV_dN^2}{(2)(155600)}\right)\ , \text{ lb/hr} \tag{19}$$

where:

F_h = fuel consumption, lb/hr

H = net heat value of fuel, Btu/lb

N = engine speed, rpm

V_d = engine displacement, in^3

bhp_p = engine horsepower as measured at PTO shaft

2 = factor for four-stroke cycle engine

α and c = engine constants determined from Nebraska Test data

Persson using data from Nebraska Varying Power and Fuel Consumption tests shows that a nearly linear relationship existed between PTO mean effective pressure (pmep) and the fuel consumption term β (Figure 2). Where pmep and β are defined as:

$$\beta = \frac{(2)H}{60NV_d} F_h \text{ , Btu/in}^3 \qquad (20)$$

$$\text{pmep} = \frac{396{,}000 \times (2) \times \text{PTOHP}}{V_d \times N} \text{ , psi} \qquad (21)$$

The equation for the line in Figure 2 is expressed as:

$$\beta = \frac{1}{12a} (\text{pmep} + b)\text{, Btu/in}^3 \qquad (22)$$

where:

$\frac{1}{12a}$ = slope of the line, lb ft/Btu

b = constant, psi

Persson suggests the 85 and 21.3 percent load points be used to determine the line of Figure 2 since the points between them were either close to or on the straight line connecting these points. This study utilizes regression analysis of β versus pmep for 10 individual Nebraska tractor tests giving coefficients of determinations (R^2) ranging from 0.995 to 0.999. The 85 and 21.3 percent load points fall extremely close to the regression line. The 100 percent load points are farther from the regression lines than the 85 percent load points in nine of the ten trials

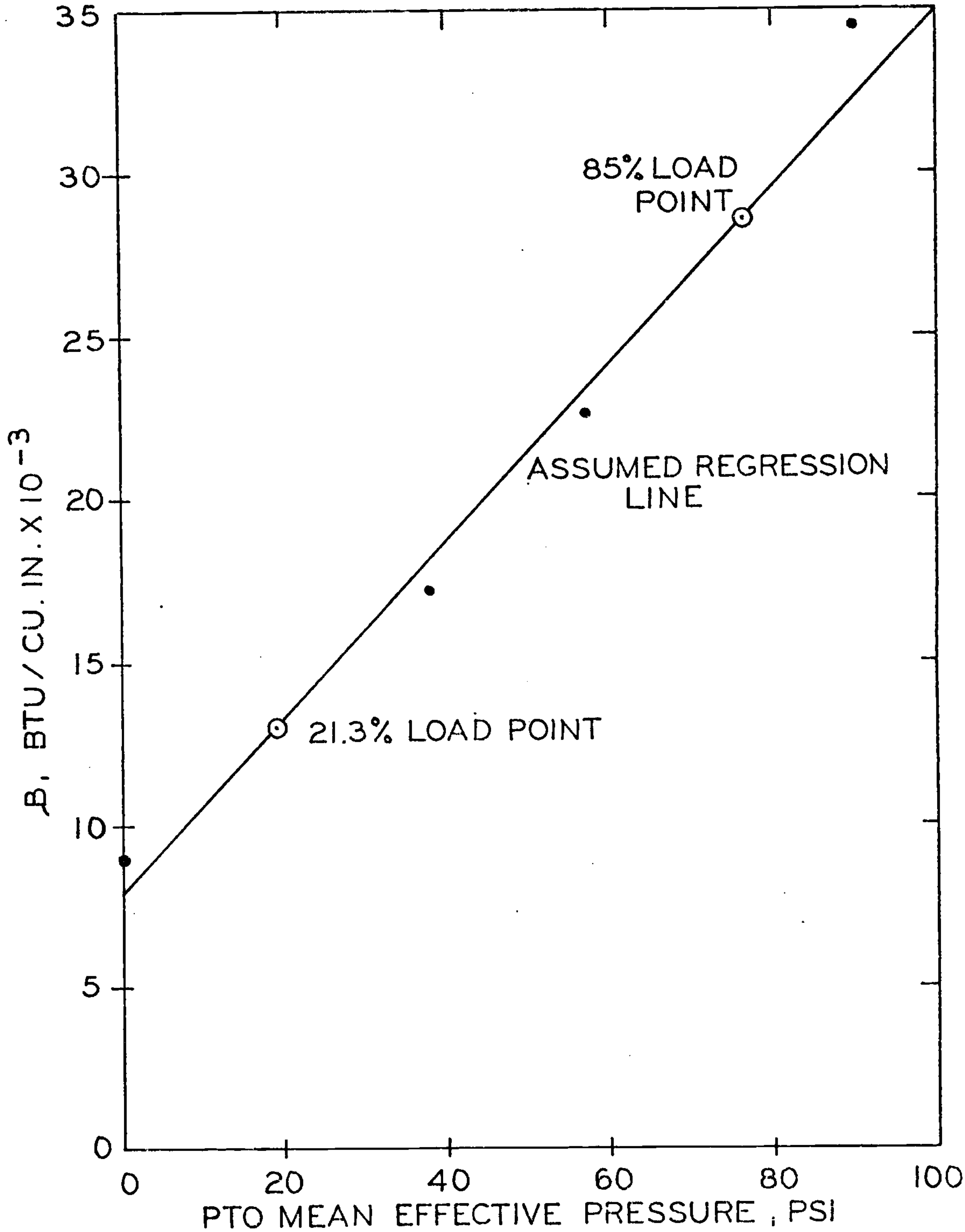


Figure 2. Fuel consumption (β) versus PTO mean effective pressure for Case 870 diesel tractor calculated from Nebraska Test Report No. 1149.

(Table 1). The 100 percent load point is above the regression line for every tractor. This study's data reinforced Persson's findings that diesel tractors appear to be over-fueled to obtain higher horsepower ratings.

Table 1. DIFFERENCE BETWEEN REGRESSION AND MEASURED FUEL CONSUMPTION ($\Delta\beta$) FOR 10 TRACTORS.

Nebraska Tractor Test Report No.	$\Delta\beta = \beta_{REGRESSION} - \beta_{MEASURED}$, Btu/in^3		
	21.3% load	85% load	100% load
1157	-0.0002	0.0001	0.0005
1167	-0.0001	0.000	0.0005
1166	0.000	0.000	0.0004
1160	-0.0003	0.001	0.0006
1148	-0.0001	0.000	0.0003
1140	0.000	0.000	0.0006
1135	0.002	0.000	0.0002
1123	0.000	0.0001	0.0003
1110	0.000	0.0003	0.0001
1100	-0.002	-0.0001	0.0003

The coefficients a and b of equation (22) for the line through the 85 and 21.3 percent load points can then be determined from the following equations:

$$\frac{1}{12a} = \frac{\beta(85\%) - \beta(21.3\%)}{\text{pmep}(85\%) - \text{pmep}(21.3\%)}, \text{ lb ft/Btu} \tag{23}$$

$$b = 12a\ \beta(21.3\%) - \text{pmep}(21.3\%), \text{ psi} \tag{24}$$

where:

(21.3%) and (85%) are the percent loads

During Nebraska Varying Power and Fuel Consumption tests the tractor engine is operated at high-idle. Variations in the engine speed are due to the governor's control as the load is changed. Since the Nebraska Test fuel consumption tests are all conducted at high-idle, it is not possible to use these data to determine the effect of reduced engine speed on fuel consumption. Persson discovered from Swedish and German tractor test reports that the coefficient b varies with rpm. The equation describing this variation is:

$$b = \frac{cN}{1000} \quad , \text{psi} \tag{25}$$

where:

N = engine speed, rpm

c = coefficient in equation 19, psi/1000 rpm

The coefficient c can be determined from the Nebraska fuel consumption data at high-idle where "b" can be evaluated by equation 24. Substituting equation (25) for "b" and αJ for "a" into equation (22) yields the following equation:

$$\beta = \frac{1}{12\alpha J}\left(\text{pmep} + \frac{cN}{1000}\right), \ \text{Btu/in}^3 \tag{26}$$

where:

J = mechanical heat equivalent = 778 lb ft/Btu

$\alpha = a/J$

Combining equations (20) and (26) and replacing pmep with equation (21) gives Persson's fuel consumption equation:

$$F_h = \frac{1}{H\alpha}\left[2544\ bhp_p + \frac{cV_dN^2}{(2)15560}\right], \text{ lb/hr} \tag{19}$$

Persson compared empirical equations 26 and 22 for β to a theoretical equation for β where β was equal to:

$$\beta = \frac{1}{(12)(778)N_i}\ (pmep + lmep),\ Btu/in^3 \tag{27}$$

where:

N_i = indicated thermal efficiency

lmep = mean loss effective pressure

= imep - pmep, psi

imep = indicated mean effective cylinder pressure, psi

pmep = PTO mean effective pressure, psi

This variable α appears in the same place in equation 26 as does N_i in equation (27). Similarly, "b" and "c" occupy the same position in the equations as does lmep. Persson points out that even though α is of the approximate magnitude as N_i for diesel engines, it should not be labeled as "indicated thermal efficiency." A suggested name was "apparent thermal efficiency." Similarly, "b" could be called "apparent loss mean effective pressure."

Tractor models with a high α or "a" value ordinarily have a low fuel consumption at full load. A low "b" or "c" value indicates the tractor should perform better at reduced loads than a tractor with a high value.

Model Completion

A tractor's tractive performance can now be predicted from the following collective equations: the drawbar pull equations (7 and 8), the tractive and tractor tractive efficiency equations (13 and 15), and the axle horsepower equations (16 and 17). The required tractor inputs are tractor type, two or four-wheel drive, the number and size of the wheels, wheelbase length, drawbar height, and front and rear static weights. The cone index of the soil, measured with a cone penetrometer, as defined in ASAE R313.1 is required. Operating conditions must also be stipulated; these include the field speeds, percent wheel slip and the gears to be tested.

Information is not usually available on transmission and final drive gear ratios. The overall gear reduction of a tractor (engine rpm/axle rpm) can be determined for tractors tested in the Nebraska Maximum Power Test. The information required for each gear is travel speed, crankshaft rpm, percent slip of the drive wheels, and rear tire size.

The first step is to determine the zero slip speed for each gear at the engine rpm specified in the Nebraska test.

$$VO(I) = \frac{VS(I)}{1 - \frac{SLIP2(I)}{100}} \text{ , mph} \tag{28}$$

where:

VO(I) = no slip speed in gear I, mph

VS(I) = measured speed at SLIP(I), mph

SLIP2(I) = measured wheel slip, percent

I = gear

The second step is to determine the axle rpm for each of the gears.

$$RPMA2(I) = \frac{VO(I)}{SSLR(NTR1)} \left(\frac{5280 \times 12}{2\pi \times 60} \right) \text{ , rpm} \tag{29}$$

$$RPMA2(I) = 168.07 \frac{VO(I)}{SSLR(NTR1)} \text{ , rpm}$$

where:

RPMA2(I) = axle rpm in gear I

VO(I) = no slip speed in gear I, mph

SSLR(NTR1) = static loaded radius of drive wheel as given in ASAE R220.3

I = gear

NTR1 = tire size used in Nebraska Test

The ratio of engine rpm to axle rpm can now be determined for each gear. RPME2(I) is the engine speed used in the Nebraska Maximum Power Test for gear I.

$$RATIO(I) = \frac{RPME2(I)}{RPMA2(I)} \tag{30}$$

Knowing the engine rpm to axle rpm ratios, the engine rpm required for different field speeds and wheel slips in each gear may be determined.

$$RPM(I) = \frac{RATIO(I) \times V}{1 - \frac{SLIPM1}{100}} \times \frac{168.07}{SSLR(NTR)} \tag{31}$$

where:

RPM(I) = engine rpm at speed V and wheel slip, FS for gear I

V = actual field speed, mph

SLIPM1 = percent wheel slip (field operation)

SSLR(NTR) = static loaded radius, ASAE R220.3

I = gear

NTR = equipped tractor tire size

The rpm determined from equation 31 is used in equation 19 to predict fuel consumption.

Using the drawbar pull predicted from equations 7 and 8 and the field speed set in the operating conditions, the drawbar horsepower can be predicted.

$$DBHP = \frac{DBP \times V}{375} \tag{32}$$

where:

DBHP = drawbar horsepower, hp

DBP = drawbar pull, lb

V = field speed, mph

PTO horsepower is required for equation (19) to determine fuel consumption. The equivalent axle horsepower for a two-wheel drive tractor is the net pull of the drive wheels divided by the tire tractive efficiency as shown in equation (16). For a four-wheel drive tractor the axle horsepower is the drawbar pull divided by the tire tractor tractive performance as given in equation (17). The PTO horsepower used in fuel consumption equation (19) is the appropriate axle horsepower divided by 0.96.

Several checks were installed to prevent the model from selecting gears, drawbar loads and field speeds that would over-rev the engine or demand more power than available. Since only PTO horsepower at rated rpm is available in the Nebraska test reports, it was assumed that rpm and PTO horsepower are strictly linear. Figure 3 illustrates that this is appropriate, since it will give conservation results, especially at lower engine speeds where lugging the engine would be a problem.

The slope of the line in Figure 3 is then the PTO horsepower at rated engine speed divided by the rated engine speed. The first check is to determine if the engine rpm predicted by equation (31) is within reasonable limits of the rated engine rpm. Since the actual high idle speed of the engine is controlled by the engine governor, the maximum allowable speed is set equal to rated speed plus

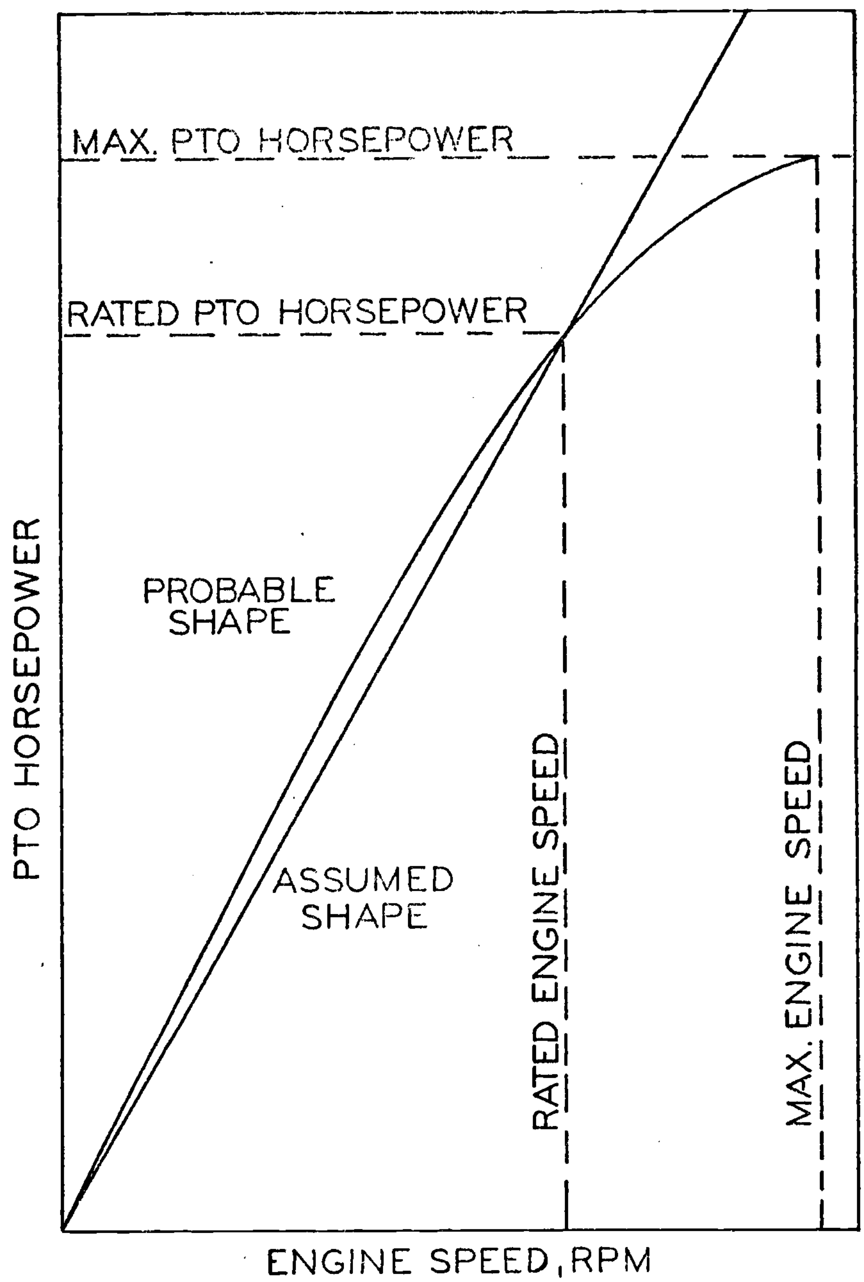


Figure 3. Relationship between probable and assumed curves of PTO horsepower versus engine speed.

10 percent. The first check then is to determine whether predicted engine speed for gear I is less than 110 percent of rated rpm.

If the engine is operating at an acceptable speed, the second check is for available power versus required power. The available power is determined by multiplying the engine rpm by the slope of the horsepower-rpm line from Figure 3.

$$APHP = RPM(I) \times \frac{RHP}{RRPM} \qquad (33)$$

where:

APHP = available PTO horsepower

RPM(I) = engine rpm for gear I

RHP = rated PTO horsepower at rated rpm

RRPM = rated rpm

The tractor is operating at an acceptable power level if the PTO horsepower predicted by equation (18) is less than the available PTO horsepower determined above.

If both the engine speed and engine load requirements are met, a prediction of tractive performance and fuel consumption can be made for the tractor at the operating conditions specified.

IV. COMPUTER MODEL

The tractive performance and fuel consumption models are not difficult to solve by hand. The calculations will take considerable amounts of time, though, if field conditions, operating conditions or tractor parameters are varied to determine the effects on tractor performance. The computer's speed enables many variations to be tested once the model has been programmed.

The FØRTRAN IV computer language was used to program the modeling formulation. Program execution was performed on the Oregon State University time sharing CDC-3300 computer. All program outputs and Central Processing Unit (CPU) times correspond to this machine's hardware. For use with the Oregon State Open Shop Operating System (OS-3), the program is designed to be run from a remote teletype terminal and to be conversational. All inputs needed for the model are entered through the teletype. Output may either be obtained from the teletype or a high speed line printer. Changes may be made in the input parameters from the teletype and new output generated, if several sets of input data are required.

To use the program, the operator simply answers a series of questions that the computer program asks him via the teletype. Several alternatives are available to the operator depending upon the area of tractor performance he wishes to study or analyze. They include predicting fuel consumption only, predicting tractive performance only, and finally, predicting both tractive performance and fuel consumption. The "fuel consumption only" section requires a given PTO horsepower level and engine speed, while the "tractive performance - fuel consumption" section predicts a PTO horsepower and engine speed to use in determining fuel consumption. The two fuel consumption predictions will be equal only if the predicted PTO horsepower and engine speed match the given values. The "tractive performance only" and "tractive performance - fuel consumption" sections predict equivalent tractive performances given the same input parameters.

The information the computer program requires depends upon the area of tractor performance which is of interest. The "tractive performance - fuel consumption" option shall be discussed further since it is the basis for the other two options.

The first information required deals with the tractor's physical and geometric characteristics. These parameters are listed below along with their corresponding model variable names:

1. Two-wheel or four-wheel drive, (ITYPE)
2. Single or dual drive wheels, (INTIRE)
3. Tractor wheelbase, inches, (WB)

4. Drawbar height, inches, (DH)
5. Rear axle static weight, lb., (RWS)
6. Front axle static weight, lb., (FWS)
7. Rear tire size, Ex. 18.4-34, (TIRER)
8. Front tire size, Ex. 10.00-16, (TIREF)

Next, information is required on soil strength. Values for the front tire cone index (CIF) and the rear tire cone index (CIR) are entered at this time. Units for the cone indices should be in psi. The tractor operating conditions also need to be entered; these include:

1. Percent rear wheel slip, (RSLIP)
2. Percent front wheel slip (four-wheel drive only), (FSLIP)
3. Range of field speeds, mph, (VMAX), (VMIN)
4. Number of speeds between VMAX and VMIN at which to determine fuel consumption, (ITER)

The tractor is modeled at the maximum (VMAX), and minimum (VMIN) speed and also at intermediate speeds. The value of ITER determines the intermediate speeds. To calculate engine rpm the ratio of engine rpm to axle rpm (RATIO(I)) for each gear to be simulated must be determined. The program contains two methods of obtaining the overall gear ratios (RATIO(I)). If data on the tractor being modeled are available, the overall gear ratios could be calculated and entered into the program directly. Two measurements are necessary for these calculations. First, measure the distance the tractor travels on a hard surface during one revolution of the drive wheel. Secondly, measure engine rpm and zero slip tractor velocity for each gear of interest. The tractor's zero slip velocity is calculated on a hard surface with no drawbar load. The overall gear ratio is then calculated using the following equation:

$$RATIO(I) = \frac{J}{VO(I,J)} X \left(\frac{60}{12 \times 5280} \right)$$

or

$$RATIO(I) = \frac{J}{VO(I,J)} X \; (9.47 \times 10^{-4}) \qquad (34)$$

where:

I = gear

RATIO(I) = overall gear ratio for gear I

VO(I,J) = zero slip speed at the measured rpm for gear I, mph

J = measured engine rpm in gear I

X = distance traveled per revolution of rear wheel, inches

The actual static loaded radius of the tractor drive wheels is also inputed at this time. The static loaded radius is the distance the tractor travels on a hard surface during one revolution of the drive wheel divided by 2π. If the second method of determining the overall gear ratios is used, the static loaded radius is assumed to be equal to the value listed in ASAE Recommendation R220.3 and does not need to be inputed.

The second method entails using the Nebraska Maximum Power Test data for the specific tractor being modeled. The gear ratios can only be calculated for the gears that are listed in the Nebraska Test. The computer model determines the gear ratios from equation (30). If the Nebraska test data are used, the number of gears to test (NOG) and the gears (KGEAR(I)) are inputed at this time. The Nebraska Test data needed to compute the gear ratios are inputed later in the program.

The program at this time makes tractive performance calculations. Then information about the tractor from its Nebraska Test Report is required for fuel consumption predictions. First, the engine displacement in cubic inches (DISP) is entered. Next, the 85 percent and 21 percent load data from the Varying Power and Fuel Consumption test are required. From these data the engine parameters α, b and c are calculated. The required information is:

PTOHP(K) = PTO horsepower

RPME1(K) = engine rpm

GPH(K) = fuel consumption, gal/hr

K = 85 and 21 percent load points

To determine the PTO horsepower versus rpm curve, the PTO horsepower (EPTOHP) at rated engine speed (ERPM) from the Power Take-off Performance Test is entered. If the overall gear ratios were earlier entered directly, the computer model now has all the information required to complete the calculations. If not, the size of drive wheels (TIRENR) used in the Nebraska test is inputed along with the following Maximum Power Test data:

JGEAR(K) = gear

VS(K) = speed, mph

RPME2(K) = engine speed, rpm

SLIP2(K) = wheel slip, percent

A set of Maximum Power Test data are inserted for every gear for which tractor performance is to be predicted.

The computer model then completes the calculations and outputs to the teletype the predictions for tractive performance and fuel consumption. Several options are available to make further tests of the tractor by changing any or all of the following variables:

CIF = front tire cone index, psi

CIR = rear tire cone index, psi

RSLIP = rear wheel slip, percent

FSLIP = front wheel slip, percent

VMAX = max field speed, mph

VMIN = min field speed, mph

DIV = number of speeds between maximum and minimum

KGEAR(I) = gears to be tested

After completion of all the desired variations the program may be started over at the beginning to model a different tractor, or program execution may be terminated.

V. COLLECTION OF FIELD TEST DATA

Field tests were conducted with several tractors to produce data for comparison with the model's predicted tractor performance. Six tractor models were tested. Five of the six tractors were diesel powered. All of the tractors, except one, were relatively low horsepower (less than 55), conventional two-wheel drives. One large four-wheel drive with dual tires was tested.

The tractors were obtained from several sources. The Hyslop Agronomic Experiment Station loaned the International 130. Oregon State University's Farm Services Division supplied the Massey-Ferguson 235, Allis-Chalmers 170 and Allis-Chalmers 6040 tractors. The Ford 3000 was borrowed from Oregon State University's Jackson Farm. Macnab Company Ranch donated the use of the Case 2470 tractor.

Five test sites were selected. The Hyslop Farm Experiment Station provided two field plots; one in summer fallow, the other in grass stubble. The summer fallow field was used to test the International 130 and Ford 3000. The International 130 was also tested on the grass stubble plot. Farm Services supplied a summer fallow and a pasture field for use while testing their tractors. The Case 2470 was tested on a wheat stubble field provided by Macnab Company Ranch.

Tractor Test Equipment

Drawbar performance and fuel consumption were the variables of greatest interest. Drawbar pull was measured with a pull meter connected between the tractor drawbar and the load (Figure 4). Two pull meters were available: a Dillion Dynamometer with a maximum load rating of 5,000 lbs. and a Bourdon tube hydraulic pull meter with a load limit of 10,000 lbs. The hydraulic pull meter was used for all the tractors because of its higher load rating.

The tractor load was supplied by towing another tractor backwards in gear. A long chain was used to minimize the effect of any differences in drawbar heights. For laying out the test course, a steel tape and range poles were used. Tractor speed was determined by measuring, with a stop watch, the time required to travel the length of the test course.

An electronic flow meter for measuring fuel consumption was not available. A volume flow meter was designed and built in the Agricultural Engineering shop. It consisted of a small fuel tank from which fuel was supplied to the engine during the test runs. The amount of fuel consumed was a function of the change in fuel depth in the tank from the initial to final points of the run. The main components of the flow meter are shown in Figure 5.

Figure 4. Bourdon tube hydraulic pull meter used during field tests to measure drawbar load.

FLOW METER ASSEMBLY

A. FUEL RETURN VALVE
B. FUEL INLET FROM ENGINE
C. FUEL OUTLET TO TRACTOR FUEL TANK
D. FUEL LEVEL SIGHT TUBE
E. FRAME
F. FUEL INLET FROM TRACTOR FUEL TANK
G. FUEL OUTLET TO ENGINE
H. LEGS
I. MAIN FUEL CONTROL VALVE
J. SCALE
K. FUEL TANK
L. FUEL TANK CAP
M. BUBBLE LEVEL

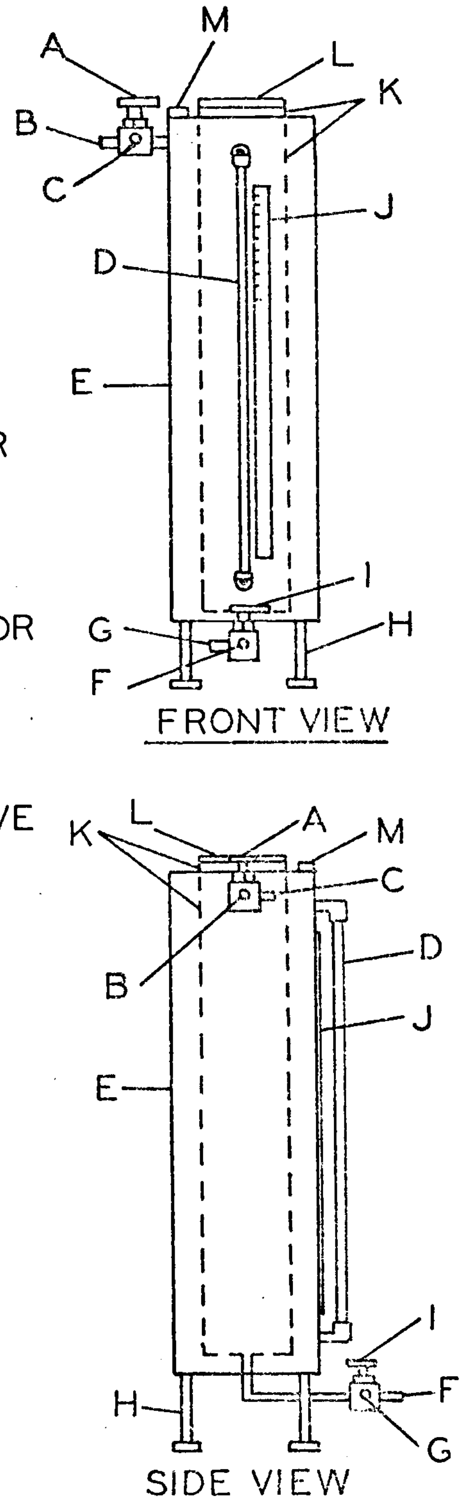
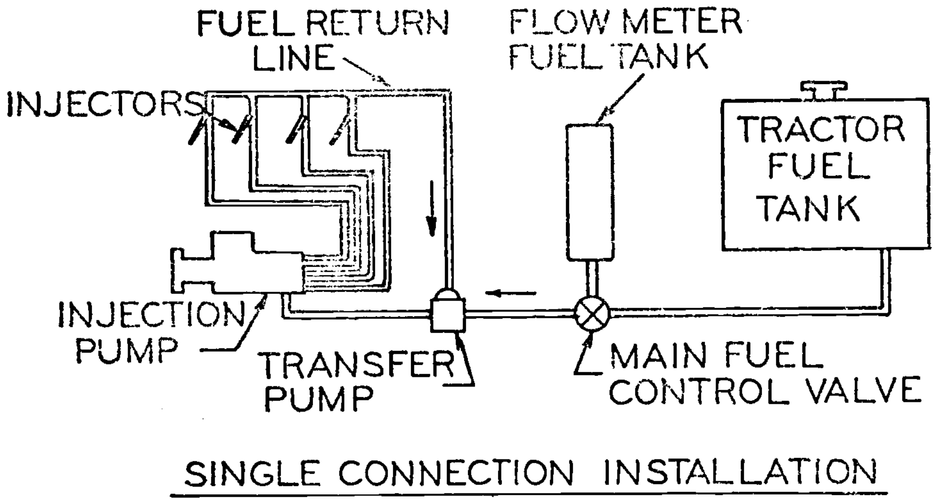


Figure 5. Volume flow meter used during field tests to measure fuel consumption.

The cylindrical fuel tank and sight tube were initially designed so one centimeter in height would contain 0.5 deciliters of fuel. An aluminum pipe with 3.5 inch outside diameter (O.D.) and 0.216 inch wall thickness formed the main body of the fuel tank. The sight tube was 1/2 inch O.D. and 1/4 inch I.D. plexiglass tubing. An extra plexiglass tube of 3/4 inch O.D. and 5/8 inch I.D. was required to obtain the correct tank volume to height ratio. During the initial field test problems arose with breakage of the extra plexiglass tube. The tube was removed and a factor of 0.96 was needed to correct the volume of the flow meter. Therefore, each centimeter of height in the flow meter contained 0.048 liter of fuel. A metal centimeter scale was fastened to the flow meter frame beside the sight tube so the level of fuel could be measured.

Figure 6 shows two types of flow meter installations: the single connection installation and the twin connection installation. The main fuel control value on the bottom of the flow meter was connected to the low pressure portion of the fuel line for both installations. The fuel control valve had two operating positions. In position one, fuel was supplied to the engine from the tractor fuel tank and the flow meter was bypassed. Position two was the test position where fuel from the main tank was shut off and the small flow meter cylinder supplied the engine.



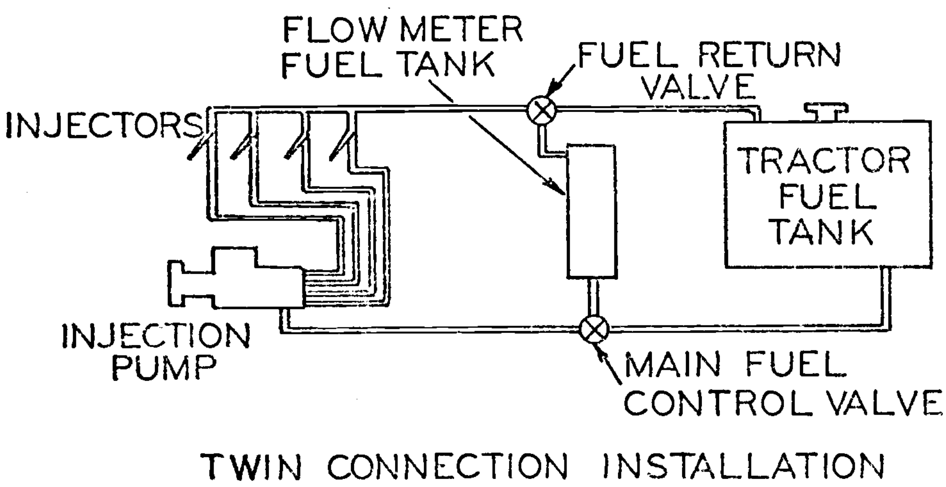


Figure 6. Two flow meter installations in a diesel tractor fuel system.

Diesel engines normally supply excess fuel to the injectors. This fuel is then either returned to the transfer pump to be sent through the system again or returned to the fuel tank. The single connection installation of the flow meter was used for systems where the return fuel line went to the transfer pump. Systems requiring a return line to the fuel tank used the twin connection installation. The top valve on the flow meter, the fuel return valve, was connected into the fuel return line. The fuel return valve was similar to the fuel control valve in that it had two positions. In the first position the fuel bypassed the flow meter and the return flowed back to the tractor fuel tank. Position two was the test position and the return fuel flowed into the top of the flow meter fuel cylinder. When both the fuel control and fuel return valves were in the test position, the net fuel consumed could be determined. The single connection installation determined net fuel consumption directly, since the return fuel was automatically returned to the engine side of the flow meter. This reduced the fuel necessary for the flow meter to supply. Tygon flexible tubing was used for connecting the flow meter to the tractor fuel system.

Of the five diesel tractors tested, three had fuel return lines that went back to the transfer pump. The Case 2470 and Ford 3000 had return lines that went to the fuel tank.

The International Harvester 130 was gasoline powered. The flow meter installation was similar to the single connection installation shown in Figure 6. The difference being that the fuel line out of the flow meter was connected directly to the carburetor and a fuel return line was not required.

Tractor front and rear axle static weights were necessary for predicting tractor performance. Portable truck scales (Figure 7) were borrowed from the Linn County Shop. Each tractor was weighed complete with fuel and driver before testing.

The cone index of the soil in the tractor test area was required for prediction of tractor performance. Cone penetrometers (Figure 8) were borrowed from the Deere and Company Technical Center and also the National Tillage Laboratory.

Tractor Test Procedure

The test procedure was similar for each tractor. Initially, a location for mounting the flow meter was secured. The location was chosen so that the tractor operator could operate the flow meter control valves. Also the flow meter needed to be located above the engine so fuel would flow by gravity. Next, the flow meter fuel control valve was connected into the fuel line. The fuel line

Figure 7. One of a set of portable scales used to weigh tractors.

Figure 8. Cone penetrometer being used to measure cone index of test plot.

between the fuel tank and the transfer pump was removed. A tygon tube was connected to the fuel tank outlet and the inlet side of the flow meter. Another tube was connected from the outlet side of the flow meter to the inlet of the transfer pump. If the fuel return line went to the transfer pump, no further connections were necessary. For the twin connection system the fuel return line was routed through the flow meter fuel return valve and then to the tractor fuel tank.

The next step was to complete a data sheet for the tractor. This included general information such as tractor make, model and year. Other information required was:

1. two or four-wheel drive
2. single or dual drive wheels
3. rear tire size and inflation pressure, psi
4. front tire size and inflation pressure, psi
5. rear static weight, lb
6. front static weight, lb
7. drawbar height, in
8. wheelbase length, in
9. engine displacement, in^3
10. engine manufacturer
11. fuel type (gas or diesel)
12. number of cylinders
13. rated engine speed, rpm

Also the condition of the tires (if they were worn) was noted.

The portable scales were set on a concrete slab when weighing the tractors. Each axle was weighed separately by positioning a scale under the left and right wheel. The wheels on the remaining axle were blocked up so that the tractor remained level and no weight transfer would occur. The total static weight of the axle was obtained by summing the weights registered on the left and right scales. The tractor was fueled and the operator seated for all weighings.

Information about the tractor's engine such as displacement and rated engine rpm was obtained from the tractor's Nebraska Test Report. The distance the tractor travels on a hard surface during one revolution of the rear wheel was also measured. This information was required to determine the overall engine to axle speed gear ratios.

The first tests run were the zero slip runs. These were conducted on a hard surface such as a road or firm soil surface. The object of these runs was to determine the relationship between engine speed and tractor velocity at zero drawbar load and zero wheel slip for each of the gears to be field tested. A test strip 30 meters long was staked out. The tractor was then driven through the strip at a constant speed. The time required, engine speed and tractor gear were recorded for the run. Several runs were normally made for each gear at different engine speeds. A zero slip velocity was determined for each run from the following equation:

$$VO(I,J) = \frac{D}{T(I,J)}\left(\frac{60}{88}\right), \text{ mph} \tag{35}$$

where:

I = gear

J = engine speed, rpm

VO(I,J) = zero slip speed in gear I with engine speed J

D = length of test run, 30 m ≈ 98.43 ft

T(I,J) = time, sec

The overall gear ratio was then determined for each run by:

$$RATIO(I) = \frac{J}{VO(I,J)} \; 9.47 \times 10^{-4} \tag{34}$$

If several runs are made in one gear, the overall gear ratios should be the same. Errors in measuring time and engine speed will cause small variations in the calculated gear ratios. An average overall gear ratio was calculated for these cases. Rearranging equation (34), the zero slip speed for each gear at any rpm can be determined.

$$VO(I,J) = 9.47 \times 10^{-4} \; \frac{J}{RATIO(I)} \tag{36}$$

Upon completion of the zero slip runs, a test site was selected for the tractive performance and fuel consumption tests. The optimum test site would be a long level plot with uniform soil characteristics throughout. Also the plot should be wide enough so that all runs could be made on fresh soil. Unfortunately, such a site was not always available. The Farm Services Division summer fallow field where the Massey-Ferguson 235, Allis-Chalmers 170 and Allis-Chalmers 6040 were tested was not a good location. The field was located on a small hill. The tests were conducted on a short semi-level swale across the top of the hill. A 30 meter run was staked out in the levelest portion. Run lengths were also set at 30 meters for the Farm Services Division pasture plot. The summer fallow tests conducted at the Hyslop Farm were 30 meters long for the International 130 and 60 meters long for the Ford 3000. The Hyslop Farm test area was level, facilitating better test plots. The test plot for Case 2470 tested at Macnab Company Ranch was 60 meters long. There was a slight slope on this plot.

The plots were marked by first measuring out the required test distances. Then two range poles were driven into the ground approximately 10 feet apart at both the start and end lines (Figure 9).

Before any tractive performance and fuel consumption runs were conducted, the cone index of the soil was measured. Normally 10 cone penetrometer readings were randomly taken throughout the length of the test course. Figure 9 shows a possible arrangement of penetrometer test locations.

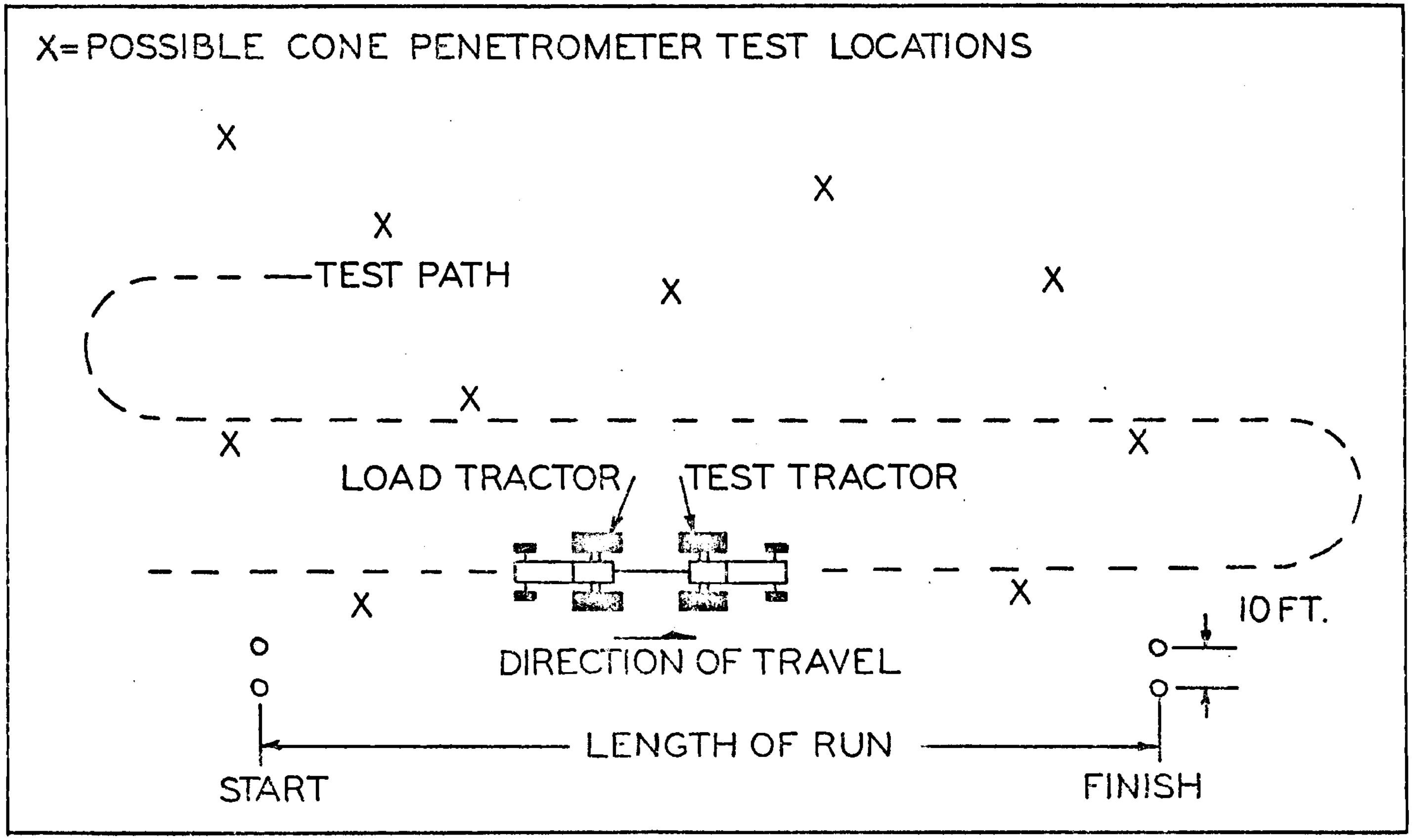


Figure 9. Typical layout of tractor test plot showing possible cone penetrometer test locations.

After completion of the cone penetrometer tests, the actual tractor tests were begun. One end of the pull meter was directly connected to the drawbar of the tractor being tested. The other end of the pull meter was connected by a chain to the drawbar of a second tractor. The function of the second tractor was to supply a load. The load was supplied by towing the second tractor backwards with its engine running, transmission in reverse, and the clutch engaged. A load would be supplied only if the second tractor was operating at a slower speed than the tractor towing it. The amount of load depended on the difference in zero slip speeds of the two tractors. The load could be changed by varying the speed of the load tractor, either by shifting gears and/or changing engine rpm. In some instances the brakes of the towed tractor were applied to increase the load. The load could be increased only to the point where 100 percent wheel slip of the towed tractor was impending.

Four people were required to perform the tests, two of whom were tractor operators. The driver of the test tractor also was responsible for operating the flow meter. Another person operated the stop watch to determine the time required for the run. The fourth person was the data keeper. His function was to record the initial and ending flow meter readings, and observe and record the pull meter readings during the test. Data that was collected for each run included:

1. gear
2. engine speed, rpm
3. drawbar pull, lbs
4. time, sec
5. initial and final fuel flow meter readings, cm

Before a test began, the flow meter fuel cylinder was filled and both control valves were in the normal run position. The fuel at that time was being delivered from the tractor fuel tank. If the return line was necessary, the excess fuel bypassed the flow meter and returned to the tractor fuel tank. The amount of fuel in the flow meter was determined by observing the height of fuel in the sight tube and reading the scale. For accurate readings it was necessary to level the flow meter. A bubble level on top of the flow meter was included for this purpose.

A test was now ready to begin. A desired gear and engine rpm were chosen for the test tractor. The test tractor was required to start pulling the load (second tractor) well ahead of the start of the test plot so equilibrium could be achieved. The operator of the second tractor tried to apply a constant load throughout the run. As the test tractor passed the initial set of range poles, marking the beginning of the run, the operator switched the flow meter control valves to the test position. The tractor was then supplied with fuel from the flow meter. Also, as the tractor passed the initial range poles the timer started the stop watch. The data keeper walked behind the test tractor observing the drawbar pull meter.

When the tractor passed the set of end range poles the flow meter was switched back to the run position and the stop watch was observed. The flow meter was again leveled and the height of fuel in the sight tube recorded. The difference in the initial and final levels determined the amount of fuel (V) used:

$$V = (IR - FR)\,\frac{0.04801}{3.78531}\quad,\ \text{gal}$$

$$FH = \frac{V}{T}\,3600\quad,\ \text{gal/hr}$$

where:

V = volume of fuel, gal

IR = initial fuel level, cm

FR = final fuel level, cm

0.04801 = conversion to liters (volume correction .050 x .96)

3.78531 = conversion from liters to gallons

FH = fuel consumption, gal/hr

T = elapsed time for run, sec

The tractors were then turned around and a test was conducted in the opposite direction. The flow meter fuel cylinder held enough fuel so that several runs could be made before necessitating a refill. The test tractor was operated on fresh soil during each run until either the virgin strip had been exhausted or the distance from the range poles to the tractor became too great for accurate measurements.

Another set of cone penetrometer readings were then taken in the tractor tracks of the runs already completed. The test surface was then harrowed and additional cone petrometers readings recorded so that more runs could be conducted. Tests were normally conducted for each tractor in the first four or five gears. Each gear was also tested at several different engine speeds. Cone penetrometer readings were taken before and after each set of test runs.

From the information gathered from the zero slip tests and field tests the drawbar horsepower, wheel slip and fuel consumption could be determined for each run. The data acquired during the field test also enabled tractor performance to be predicted via the computer model.

Tractor Test Data

General information about the tractors tested plus the test results are given in Appendix A. Table A.1 of Appendix A lists the physical and geometric information that was gathered for each tractor before any field tests were conducted. Figures A.1 through A.6 report the zero condition test results for each tractor. From the zero condition graphs the tractor's zero slip speed versus engine speed for each gear tested can be determined. The zero slip speed is required for determining the percent of drive wheel slip in the test runs. Field test results are reported in Tables A.2 through A.11. The information reported includes: the tractor tested, the field used and its soil cone indices before and after the test runs, and the data collected for each individual run.

VI. VERIFICATION OF MODEL

Model verification is similar to model development; the model's parts can be verified separately and then again as a whole. The accuracy of the fuel consumption portion is directly affected by the ability of the tractive performance equations to predict drawbar pull and tire efficiencies. No matter how well fuel consumption may be predicted using known PTO loads, the fuel consumption obtained from predicted PTO loads will be of little value if the tractive efficiency equations do not correctly describe drawbar performance.

For this reason, the study separated the model into the above two separate entities. The result would, therefore, indicate each model component's ability to predict its physical system.

Predicted Versus Measured Tractive Performance

The accuracy of the tractive performance portion of the model was measured by comparing the actual drawbar performance of each tractor to its predicted performance. In Figures 10 through 19 the measured values of drawbar pull and wheel slip were plotted by tractor and field plot. The model was then used to generate predicted values of drawbar pull for each tractor at varying wheel slips for several different cone indices.

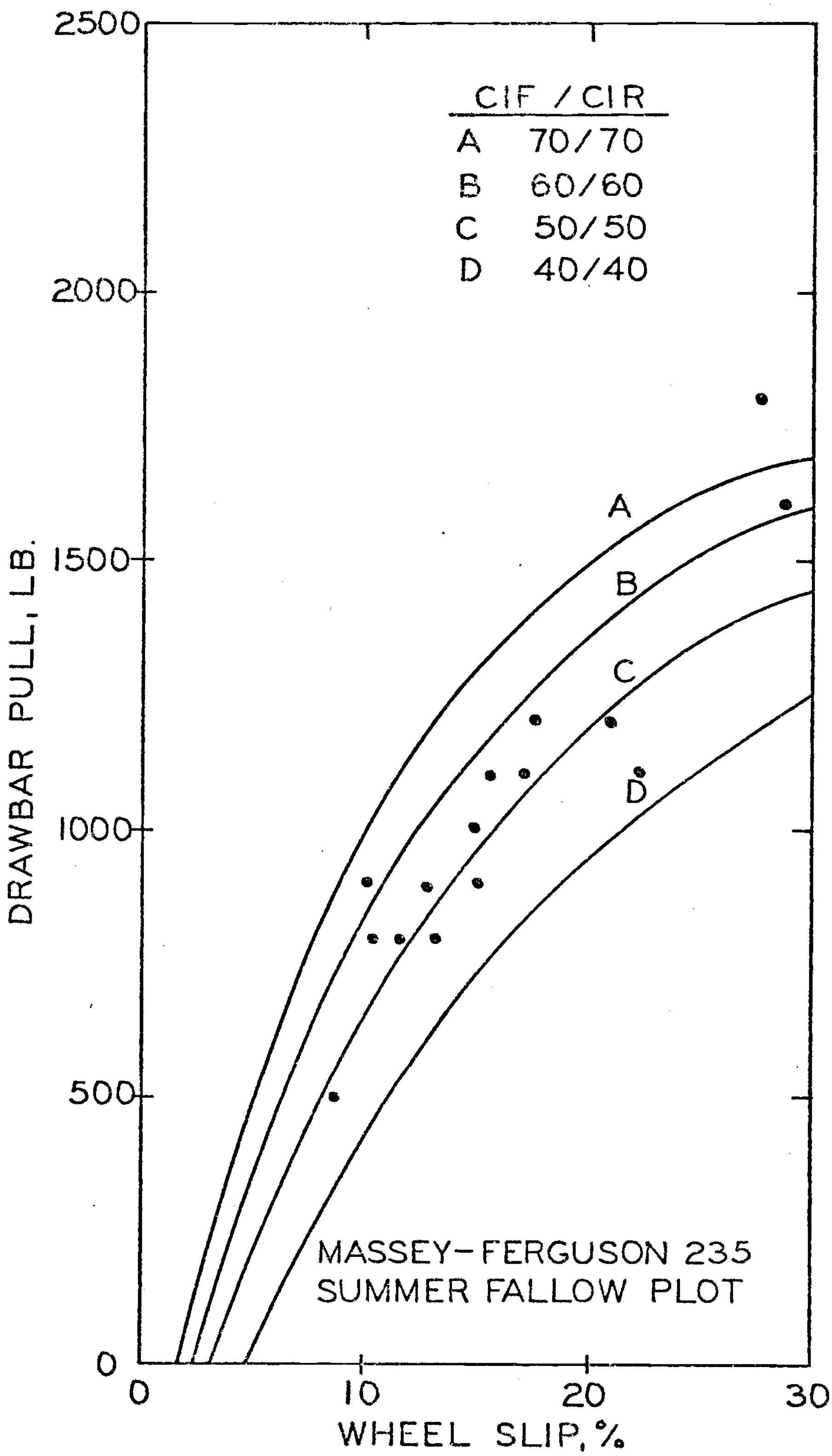


Figure 10. Predicted and measured drawbar pull and wheel slip for the Massey-Ferguson 235 on summer fallow plot with various cone indices.

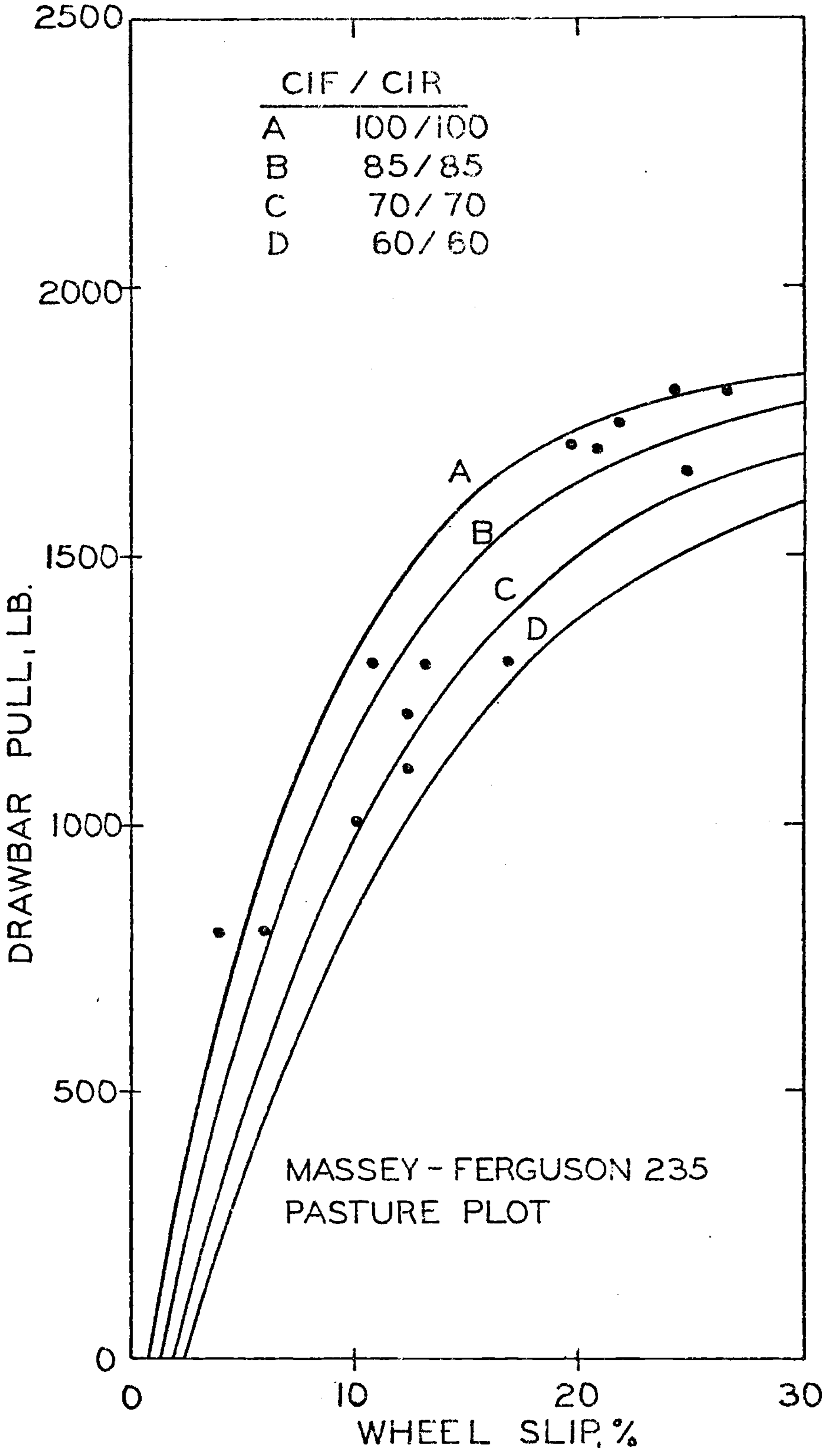


Figure 11. Predicted and measured drawbar pull and wheel slip for the Massey-Ferguson 235 on pasture plot with various cone indices.

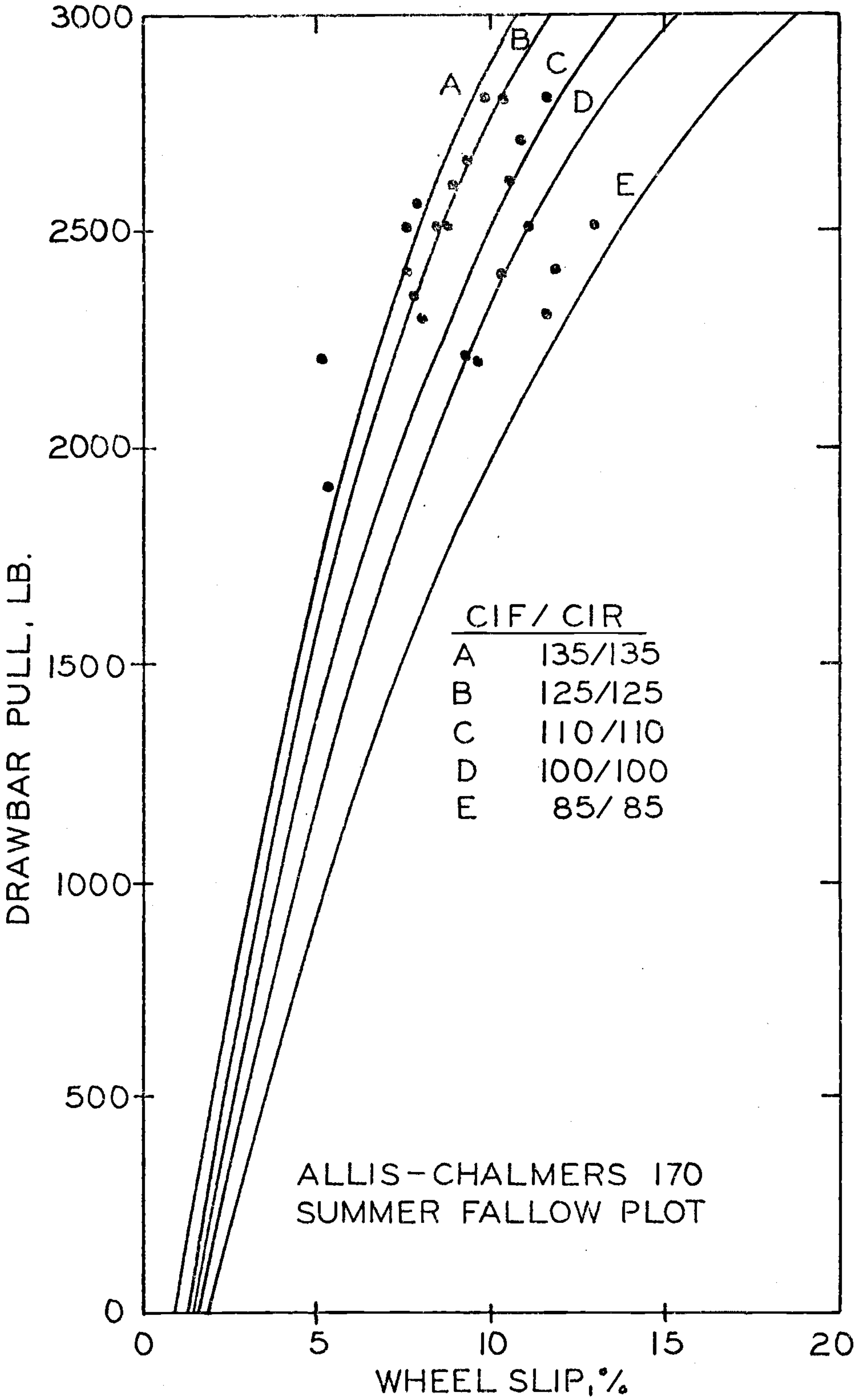


Figure 12. Predicted and measured drawbar pull and wheel slip for the Allis-Chalmers 170 on summer fallow plot with various cone indices.

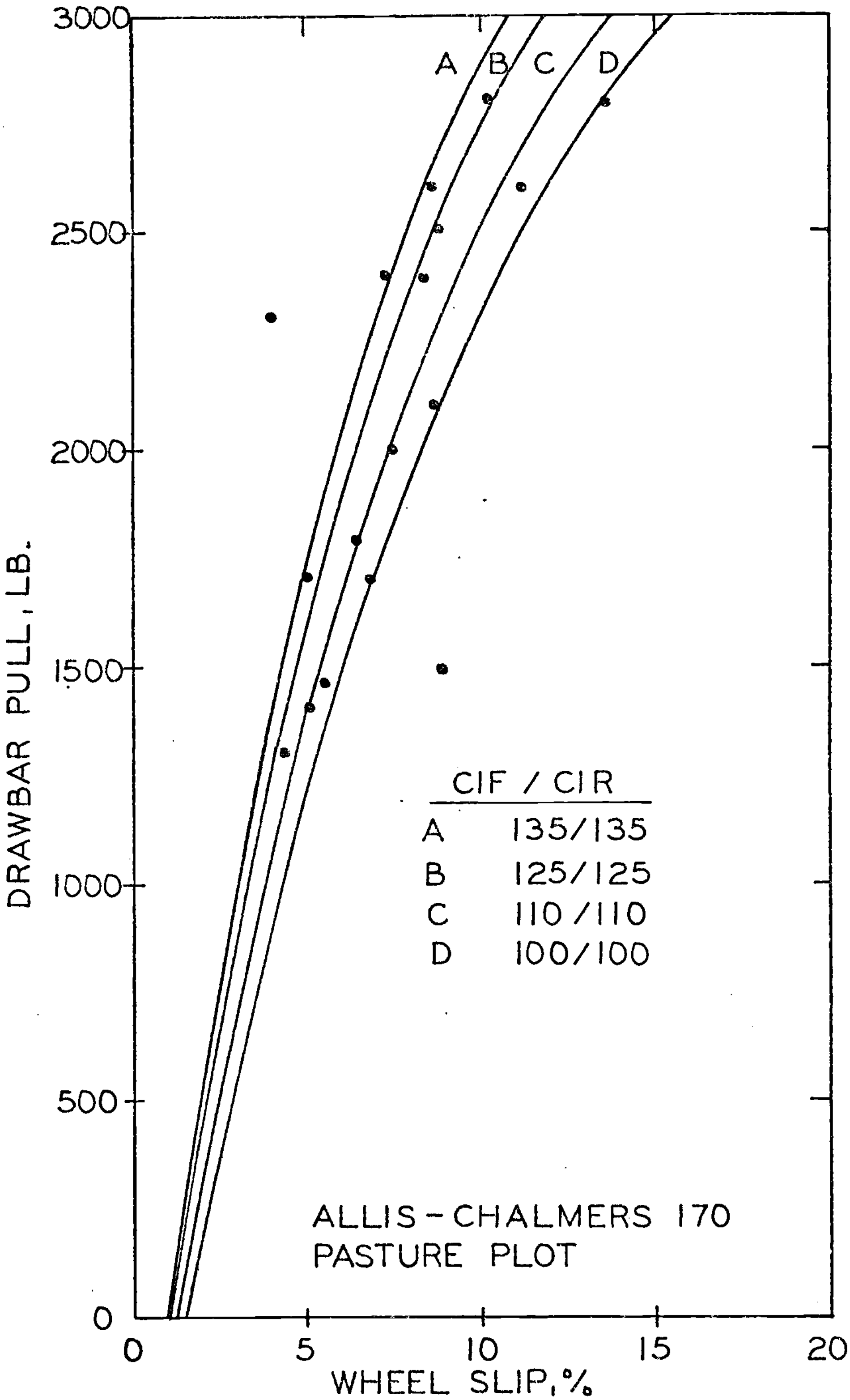


Figure 13. Predicted and measured drawbar pull and wheel slip for the Allis-Chalmers 170 on pasture plot with various cone indices.

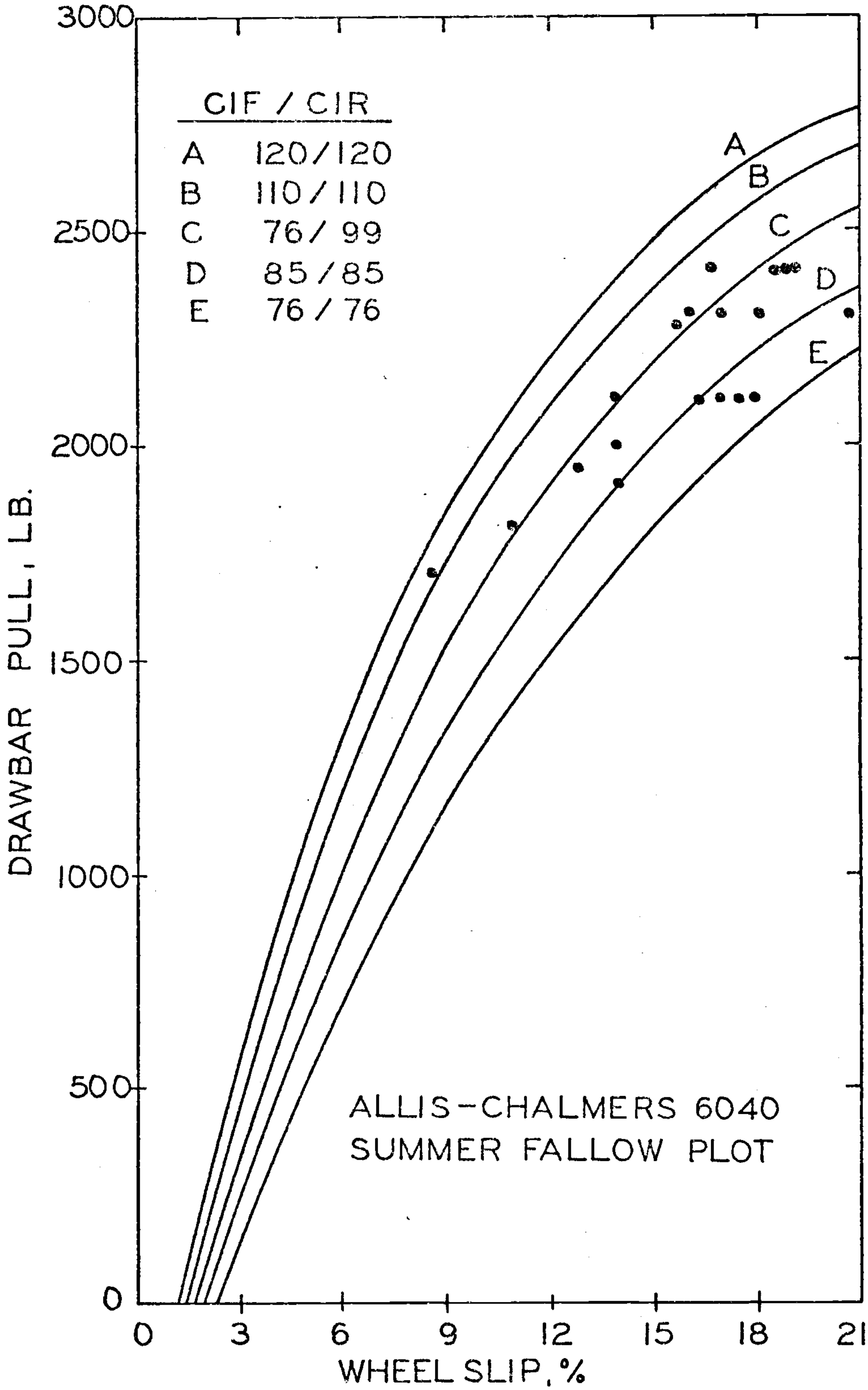


Figure 14. Predicted and measured drawbar pull and wheel slip for the Allis-Chalmers 6040 on summer fallow plot with various cone indices.

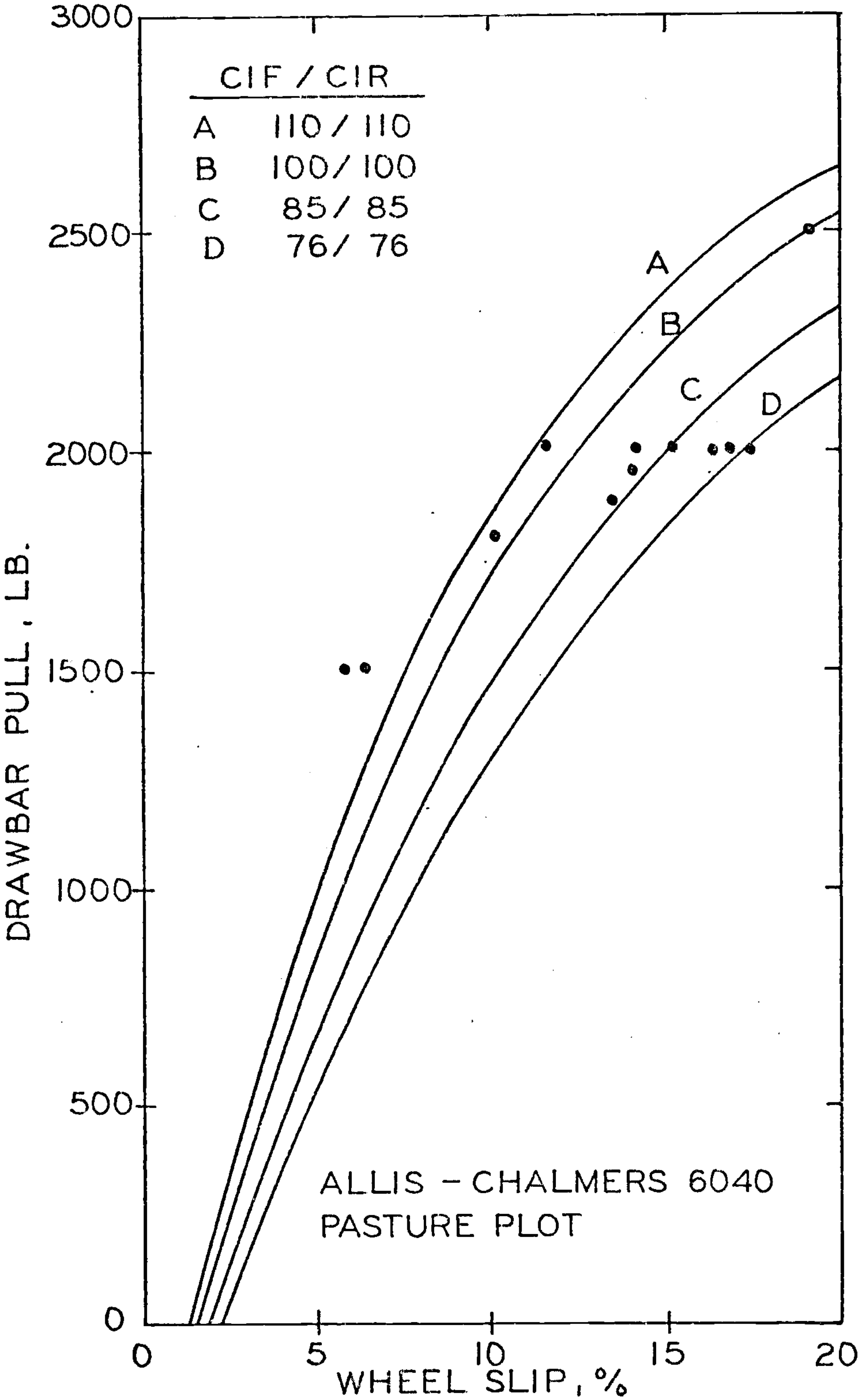


Figure 15. Predicted and measured drawbar pull and wheel slip for the Allis-Chalmers 6040 on pasture plot with various cone indices.

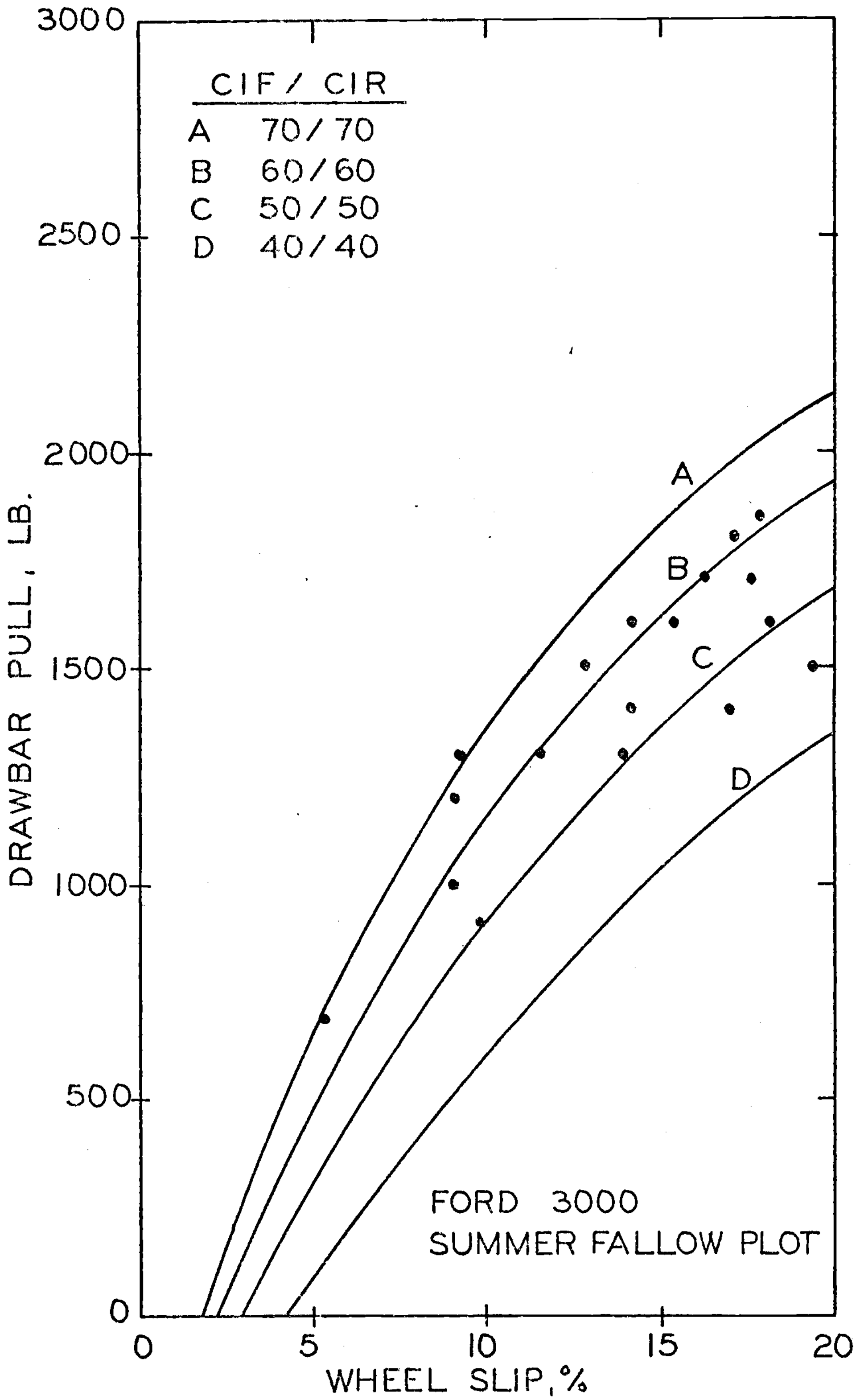


Figure 16. Predicted and measured drawbar pull and wheel slip for the Ford 3000 on summer fallow plot with various cone indices.

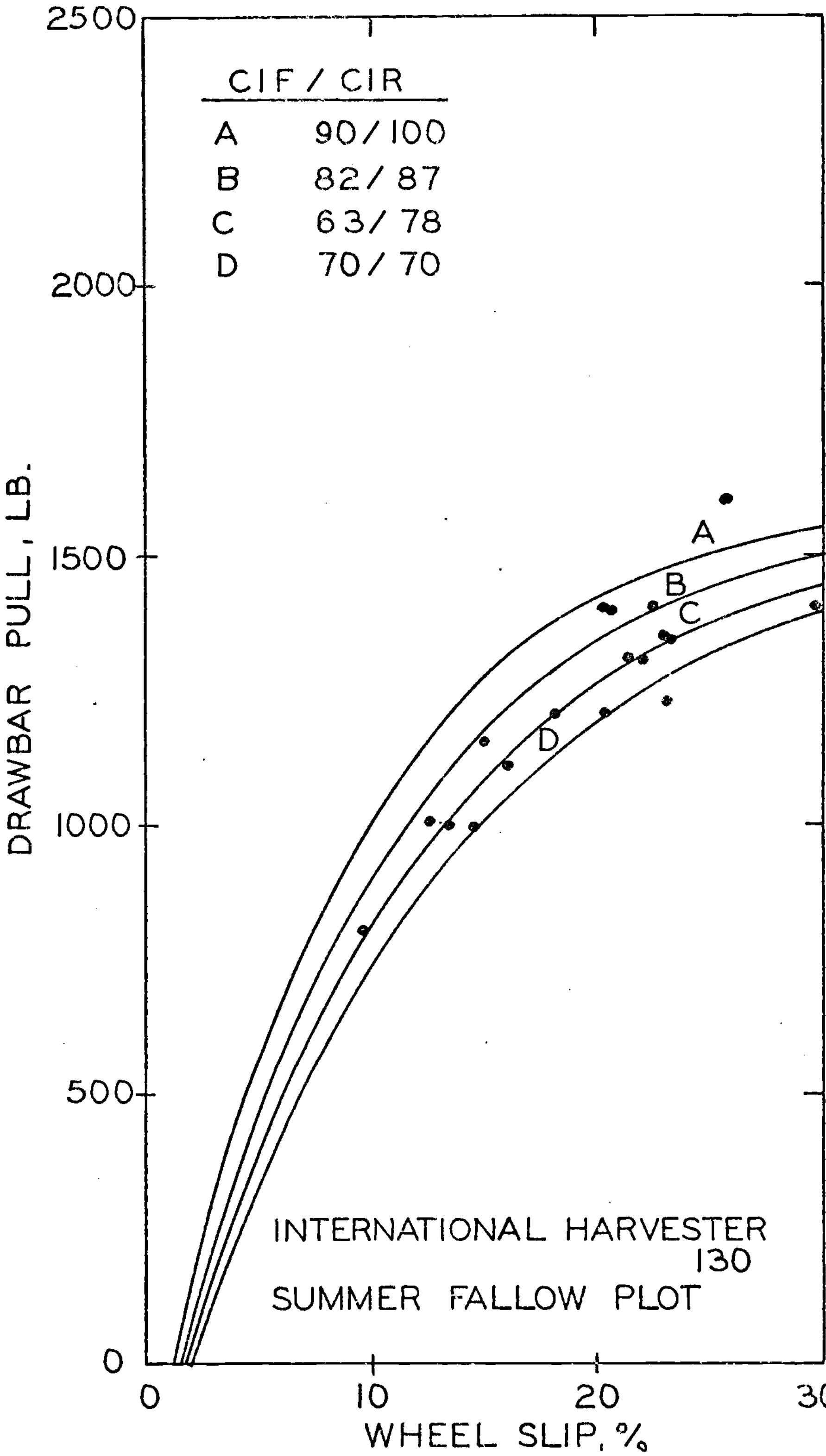


Figure 17. Predicted and measured drawbar pull and wheel slip for the International Harvester 130 on summer fallow plot with various cone indices.

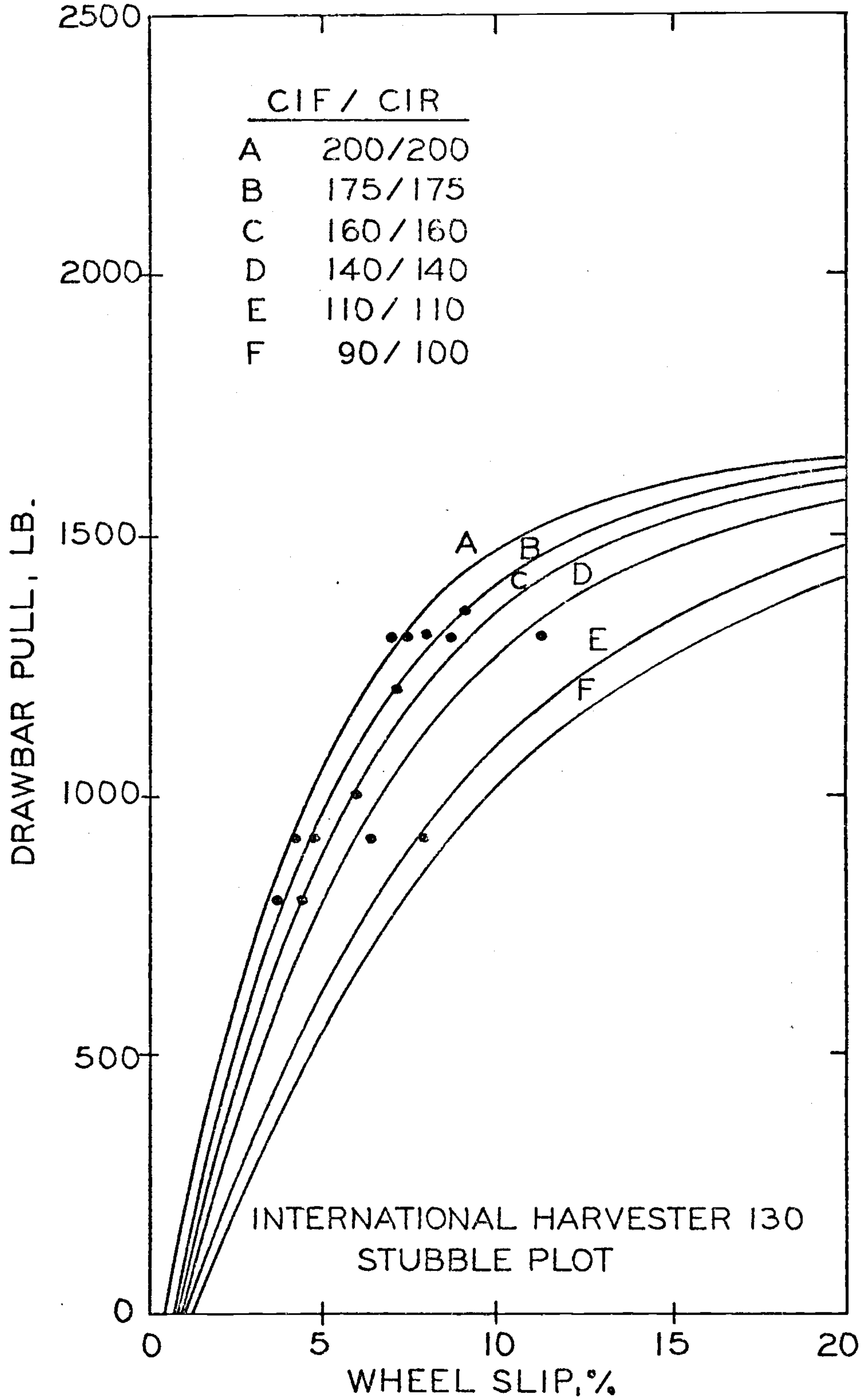


Figure 18. Predicted and measured drawbar pull and wheel slip for the International Harvester 130 on stubble plot with various cone indices.

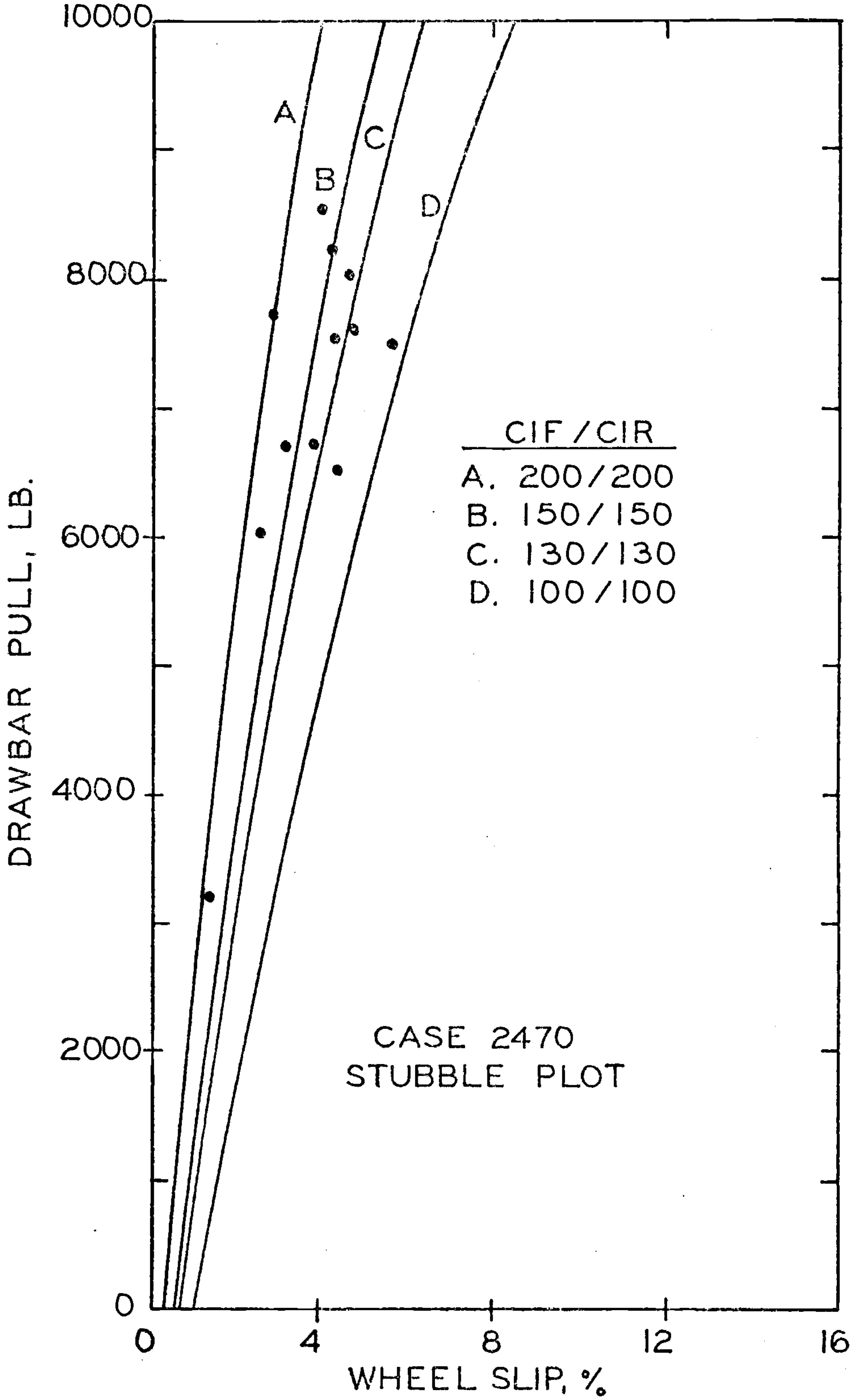


Figure 19. Predicted and measured drawbar pull and wheel slip for the Case 2470 on stubble plot with various cone indices.

Wismer and Luth (1972) reported that the 0- to 6-inch average cone index produced the best correlations in the traction equations for machines with tire sinkages of less than three inches. Also they stated that generally the cone index should be measured before the soil is subjected to wheel traffic. Soils that were highly compactible, though, tended to increase in strength under heavy wheel loads. After being subjected to traffic, these soils may have cone indices several times larger than the initial before traffic values.

The problem of selecting the appropriate cone indices to use in the traction equations is one which requires further study. The model required cone index values for both the front and rear wheels. If the rear wheels of the tractor followed in the tracks of the front wheels, the cone index of the rear wheels could change. The rear wheel cone index of a large tractor working on a freshly tilled soil can be greatly affected by the compaction caused by the front wheels.

Table 2 shows the measured cone index for each tractor and soil with the cone index range that best fitted the measured data. These were obtained by observing the range of cone indices in Figures 10-19 which included the majority of measured data points for each tractor. The measured and best-fit cone indices of the summer fallow plots were within reasonable agreement. The measured cone indices of the

Table 2. COMPARISON OF MEASURED AND BEST-FIT CONE INDEX VALUES.

Tractor and Field Type	Cone Index, psi				
	Measured				Range that best fits data
	Before 0-4 in	After 0-4 in	Before 0-6 in	After 0-6 in	
MF 235 Summer Fallow	50	71	65	80	45-60
MF 235 Pasture	455	457	464	461	60-100
AC 170 Summer Fallow	66	74	90	120	85-140
AC 170 Pasture	279	300	356	350	100-135
AC 6040 Summer Fallow	72	85	97	137	75-120
AC 6040 Pasture	370	329	444	392	75-110
Ford 3000 Summer Fallow	60	80	71	87	50-70
IH 130 Summer Fallow	63	78	82	87	70-100
IH 130 Stubble	250	261	301	312	140-200
Case 2470 Stubble	246	357	391	485	100-200

pasture and stubble plots were normally three to four times larger than the cone indices that best fit the measured results. This could be explained in part by the fact that the pasture plot was very rough and had a crop of grass approximately 18 inches tall initially before any tests were conducted. The grass, when flattened, made a very slick test surface. The grass stubble plot on which the International Harvester 130 was tested also had a slick test surface. The Case 2470 was tested on a wheat stubble plot which was both rough and slick. The plot was rough due to deep furrow seeding (14 inch spacing). A considerable amount of straw was left on the ground from the harvesting operation causing the plot to be slick. The slick test surfaces contributed to higher wheel slips and, thus, lower drawbar pulls than would normally be expected.

Experimental errors and assumptions that were not totally correct explained the variations in the measured tractive performance data of each tractor. Some of the possible sources of error were in misreading the pull meter during the runs, supplying a nonuniform load to the test tractor during a test run, not having a level or long enough test site and incorrectly measuring the time to complete the run. Another source of possible error was using the tachometer with which the tractors were equipped to determine engine rpm. Assuming that the soil in the test strip was homogeneous in both lateral and vertical directions was necessary, but not correct. Variations in soil strength for the test strip could account for a large percentage of the variations in recorded drawbar pull.

Overall, the predicted tractive performance of the tractors, when operating on the summer fallow plots, was reasonably accurate. The tractive performance for the pasture and stubble fields could also be predicted if reduced values of cone index were used to describe soil strength. In each set of draw bar tests all but one or two data points laid within a fairly narrow family of predicted drawbar pull versus wheel slip curves (Figures 10 to 19).

Predicted Versus Measured Fuel Consumption

Comparisons were made between the measured and predicted values of tractor fuel consumption. PTO fuel consumption tests were not conducted, so the drawbar fuel consumption data was used as the basis for comparison. To eliminate as much error as possible in predicting fuel consumption, the cone indices used in the model were not necessarily those measured in the field but, rather, those which would predict most closely the measured drawbar load. This was done since the objective was only to determine the accuracy of the fuel consumption portion of the model. As noted above, for the pasture and stubble test plots the measured and desired cone indices vary greatly. The ability of the tractive performance portion of the model to predict tractor tire efficiency, PTO horsepower and engine speed still affected the fuel consumption predictions. It was not possible to completely eliminate the errors introduced by the tractive performance section of the model since PTO horsepower had not been measured.

Tables 3 through 12 compare measured and predicted fuel consumption for each tractor. The cone index listed was used to predict tractive performance and, thus, fuel consumption. For these runs, the overall gear ratios and static loaded radius of the drive wheels were determined from the zero condition tests (Equation 34), rather than the Nebraska Test Reports.

Table 13 shows the mean difference ($\bar{d}$) in predicted minus measured fuel consumption, the standard deviation of the differences ($s_{\bar{d}}$), and the 95 percent confidence interval estimate of the true mean difference in fuel consumption ($u_{\bar{d}}$) for each tractor. The predicted fuel consumption accurately described the measured fuel consumption for only three tractors. The Ford 3000, International Harvester 130 and Case 2470 each had a low mean difference between predicted and measured fuel consumption. Only one tractor, the Allis-Chalmers 170, tested on the summer fallow plot, had the standard deviation of the mean differences between measured and predicted fuel consumption large enough to cause an unreasonably wide 95 percent confidence interval estimate. The larger variance in the Allis-Chalmers 170 summer fallow runs was probably caused by not having the

Table 3. COMPARISON OF PREDICTED AND MEASURED PERFORMANCE OF MASSEY-FERGUSON 235 ON SUMMER FALLOW PLOT.

Run No.	Gear	Cone Index psi	Engine Speed rpm		Drawbar Horsepower		Fuel Consumption gal/hr	
			Measured	Predicted	Measured	Predicted	Measured	Predicted
1	2	48	2250	2246	5.00	5.10	2.06	1.01
2	2	56	2250	2255	6.06	6.19	1.69	1.07
3	2	53	2000	1996	4.94	5.04	1.89	.86
4	2	52	2000	2002	4.11	4.10	1.71	.81
5	2	48	1800	1797	3.63	3.70	1.62	.69
6	2	58	1800	1803	3.75	3.89	1.61	.69
7	3	43	2250	2249	7.69	7.74	2.14	1.26
8	3	49	2250	2248	8.52	8.53	2.17	1.28
9	3	49	2000	1999	6.12	6.33	1.82	.98
10	3	52	2000	2004	7.29	7.34	1.44	1.04
11	3	50	1800	1802	3.29	3.45	2.01	.71
12	3	56	1800	1798	7.11	7.13	1.74	.92
13	4	61	2250	2253	13.64	13.57	2.50	1.69
14	4	90	2250	2249	15.56	15.43	2.43	1.76
15	4	49	2000	2003	10.13	10.08	1.94	1.28
16	4	49	2000	2003	10.13	10.08	1.83	1.28
17	4	54	1800	1802	7.53	7.63	1.71	.97
18	4	64	1800	1802	7.74	7.83	1.98	.96

Table 4. COMPARISON OF PREDICTED AND MEASURED PERFORMANCE OF MASSEY-FERGUSON 235 ON PASTURE PLOT.

Run No.	Gear	Cone Index psi	Engine Speed rpm		Drawbar Horsepower		Fuel Consumption gal/hr	
			Measured	Predicted	Measured	Predicted	Measured	Predicted
1	1	70	2250	2257	4.01	4.03	1.95	.90
2	1	66	2000	1995	3.82	3.84	1.69	.75
3	2	100	2250	2251	8.87	8.87	2.20	1.24
4	2	74	2250	2245	8.11	8.09	2.26	1.20
5	2	73	2000	2004	6.10	6.14	1.95	.90
6	2	115	1800	1805	4.02	3.54	1.15	.63
7	2	87	1600	1603	3.49	3.51	1.11	.55
8	3	92	2250	2248	12.07	12.11	2.08	1.44
9	3	80	2000	1999	9.02	9.20	2.12	1.10
10	3	64	1800	1802	7.76	7.82	1.90	.95
11	4	95	2250	2250	15.80	15.77	2.35	1.77
12	4	95	2000	2002	14.49	14.54	2.17	1.47
13	4	91	1800	1802	11.08	11.17	1.74	1.10

Table 5. COMPARISON OF PREDICTED AND MEASURED PERFORMANCE OF ALLIS-CHALMERS 170 ON SUMMER FALLOW PLOT.

Run No.	Gear	Cone Index psi	Engine Speed rpm		Drawbar Horsepower		Fuel Consumption gal/hr	
			Measured	Predicted	Measured	Predicted	Measured	Predicted
1	2	134	1900	1903	18.59	18.81	3.66	1.99
2	2	120	1800	1801	16.80	16.84	4.10	1.81
3	2	1200	1800	1801	16.80	16.84	2.05	1.81
4	2	125	1600	1600	15.29	15.23	6.64	1.54
5	2	135	1600	1597	12.69	12.53	3.19	1.38
6	2	170	1400	1400	12.91	12.79	4.19	1.22
7	3	100	1925	1925	21.20	21.19	3.46	2.24
8	3	101	1925	1924	20.55	20.41	5.46	2.18
9	3	98	1800	1798	17.73	17.91	3.29	1.93
10	3	87	1800	1802	18.13	18.13	4.83	1.98
11	3	100	1600	1603	15.81	15.96	3.12	1.64
12	3	91	1600	1599	16.78	16.90	4.64	1.72
13	4	110	1900	1899	31.51	28.92	5.74	2.67
14	4	130	1875	1873	31.32	31.14	3.14	2.72
15	4	127	1800	1801	28.20	28.06	4.15	2.48
16	4	125	1800	1798	28.57	28.55	5.50	2.51
17	4	139	1600	1601	24.45	24.40	3.74	2.08
18	4	125	1600	1601	24.18	24.19	4.69	2.09
19	5	110	1875	1877	30.41	30.23	4.92	2.73
20	5	114	1875	1875	32.33	32.49	3.39	2.86
21	5	136	1800	1800	29.44	29.17	6.19	2.53
22	5	114	1800	1798	30.20	30.18	3.71	2.65
23	5	89	1600	1614	24.45	24.63	4.99	2.25
24	5	123	1600	1599	25.42	25.46	3.63	2.17

Table 6. COMPARISON OF PREDICTED AND MEASURED PERFORMANCE OF ALLIS-CHALMERS 170 ON PASTURE PLOT.

Run No.	Gear	Cone Index psi	Engine Speed rpm		Drawbar Horsepower		Fuel Consumption gal/hr	
			Measured	Predicted	Measured	Predicted	Measured	Predicted
1	1	106	1950	1955	14.19	14.24	3.90	1.80
2	1	125	1950	1955	15.42	15.37	3.65	1.84
3	1	123	1800	1805	12.45	12.65	3.71	1.54
4	1	133	1800	1800	13.45	13.58	3.43	1.58
5*	1	---	1600	----	11.12	-----	3.58	----
6	2	100	1950	1953	20.79	21.01	3.69	2.26
7	2	124	1950	1948	19.62	19.72	4.31	2.10
8	2	126	1800	1798	17.68	16.40	3.95	1.77
9	2	110	1600	1601	13.06	13.21	3.50	1.45
10	3	110	1950	1950	16.27	16.32	3.92	1.96
11	3	101	1800	1799	14.15	14.12	3.82	1.71
12	3	101	1600	1602	15.22	15.26	3.70	1.59
13	4	123	1800	1798	14.82	15.44	4.07	1.81
14	4	110	1600	1600	14.08	14.00	3.59	1.57
15	4	133	1600	1600	17.09	17.37	3.85	1.71
16*	5	---	1600	----	15.25	-----	3.89	----
17	5	105	1400	1400	13.38	13.26	3.30	1.38

*Predictive tests were not conducted due to errors in measured results.

Table 7. COMPARISON OF PREDICTED AND MEASURED PERFORMANCE OF ALLIS-CHALMERS 6040 ON SUMMER FALLOW PLOT.

Run No.	Gear	Cone Index psi	Engine Speed rpm		Drawbar Horsepower		Fuel Consumption gal/hr	
			Measured	Predicted	Measured	Predicted	Measured	Predicted
1	3 Lo	100	2200	2206	5.46	5.57	3.83	0.98
2	4 Lo	108	2200	2208	9.90	10.16	4.12	1.35
3	1 Hi	100	2200	2197	11.22	11.50	4.09	1.38
4	1 Hi	92	2200	2201	10.72	10.77	4.24	1.35
5	1 Hi	101	2000	2000	10.92	11.02	3.95	1.25
6	1 Hi	93	2000	1999	11.01	10.95	4.14	1.28
7	1 Hi	98	1800	1806	9.85	9.74	3.50	1.08
8	1 Hi	113	1800	1797	7.90	7.97	3.91	0.93
9	2 Hi	85	2200	2199	15.88	15.67	4.46	1.71
10	2 Hi	101	2200	2200	18.51	18.73	4.36	1.88
11	2 Hi	98	2000	2001	16.73	17.15	4.36	1.69
12	2 Hi	81	2000	1999	16.08	15.96	4.55	1.68
13	2 Hi	97	1800	1799	13.42	13.74	4.22	1.34
14	2 Hi	71	1800	1800	13.86	13.64	4.03	1.47
15	3 Hi	83	2200	2200	24.25	24.28	4.71	2.35
16	3 Hi	98	2200	2201	27.18	27.68	5.20	2.55
17	3 Hi	105	2000	2001	25.26	25.50	5.10	2.24
18	3 Hi	95	2000	2003	24.54	24.91	4.96	2.27
19	3 Hi	85	1800	1800	19.99	19.91	4.13	1.82
20	3 Hi	89	1800	1803	21.44	21.29	4.52	1.92
21	4 Hi	79	1825	1824	27.23	27.19	4.96	2.40
22	4 Hi	81	1825	1826	27.43	27.38	5.00	2.40

Table 8. COMPARISON OF PREDICTED AND MEASURED PERFORMANCE OF ALLIS-CHALMERS 6040 ON PASTURE PLOT.

Run No.	Gear	Cone Index psi	Engine Speed rpm		Drawbar Horsepower		Fuel Consumption gal/hr	
			Measured	Predicted	Measured	Predicted	Measured	Predicted
1	3 Lo	125	2200	2193	4.78	4.74	3.33	0.91
2	4 Lo	87	2200	2201	8.00	7.97	4.03	1.16
3	1 Hi	103	2200	2203	10.07	9.94	4.28	1.27
4	1 Hi	87	2200	2199	10.24	10.10	4.13	1.31
5	1 Hi	130	2000	2003	7.99	7.57	3.94	1.00
6	1 Hi	85	1800	1799	8.65	8.60	3.75	1.01
7	2 Hi	80	2200	2203	17.13	17.59	4.28	1.92
8	2 Hi	79	2000	2001	14.61	14.73	4.29	1.57
9	2 Hi	89	1800	1799	13.56	13.48	4.15	1.34
10	3 Hi	75	2200	2199	22.94	22.76	4.68	2.29
11	3 Hi	108	2000	1999	22.37	22.57	4.28	1.99
12	3 Hi	79	1800	1800	19.04	18.84	4.37	1.77
13	4 Hi	100	1850	1849	32.42	32.37	5.29	2.70

Table 9. COMPARISON OF PREDICTED AND MEASURED PERFORMANCE OF FORD 3000 ON SUMMER FALLOW PLOT.

Run No.	Gear	Cone Index psi	Engine Speed rpm		Drawbar Horsepower		Fuel Consumption gal/hr	
			Measured	Predicted	Measured	Predicted	Measured	Predicted
1	1	51	2200	2199	5.80	5.80	1.20	1.16
2	2	59	2200	2207	7.48	7.49	1.10	1.27
3	2	50	2200	2205	5.28	5.28	1.05	1.16
4	3	45	2150	2152	13.70	13.60	1.40	1.77
5	3	54	2000	2005	12.63	12.72	1.38	1.53
6	3	68	1500	1506	8.62	8.78	1.01	0.95
7	3	69	1000	1003	3.49	3.57	0.45	0.43
9	4	58	2000	2005	19.15	19.09	1.68	1.96
10	4	63	1500	1501	14.05	13.07	1.55	1.25
11	4	58	1000	1004	6.41	6.66	0.70	0.65
12	5	62	2100	2103	27.25	27.30	2.35	2.59
13	5	62	2000	2006	25.46	25.64	2.17	2.41
14	5	62	1500	1504	17.57	17.63	1.33	1.57
15	5	71	1000	1004	10.07	10.09	0.74	0.84
16	6	57	1600	1604	24.15	24.59	2.45	2.16
17	6	51	1500	1503	21.21	21.14	1.86	1.92
18	6	47	1000	1004	12.53	12.63	1.14	1.13

Table 10. COMPARISON OF PREDICTED AND MEASURED PERFORMANCE OF INTERNATIONAL HARVESTER 130 ON SUMMER FALLOW PLOT.

Run No.	Gear	Cone Index psi	Engine Speed rpm Measured	Engine Speed rpm Predicted	Drawbar Horsepower Measured	Drawbar Horsepower Predicted	Fuel Consumption gal/hr Measured	Fuel Consumption gal/hr Predicted
1	1	78	1200	1202	4.26	4.26	1.09	0.89
2	1	98	1200	1199	6.58	6.64	1.32	1.09
3	1	87	1200	1198	6.39	6.36	1.05	1.08
4	1	96	1400	1402	7.64	7.71	1.39	1.37
5	1	130	1400	1402	8.18	8.16	1.44	1.44
6	1	77	1400	1399	7.01	6.99	1.51	1.33
7	1	77	1400	1400	6.97	7.02	1.50	1.34
8	1	79	1550	1547	7.90	7.88	1.64	1.58
9	1	72	1550	1547	7.48	7.50	1.64	1.61
10	2	77	1200	1201	8.10	8.24	1.55	1.22
11	2	80	1200	1198	8.17	8.23	1.25	1.21
12	2	76	1400	1399	10.10	10.23	1.41	1.59
13	2	71	1400	1402	9.32	9.36	1.67	1.53
14	2	77	1500	1500	11.48	11.54	1.95	1.82
15	2	85	1500	1500	11.43	11.56	1.53	1.78
16	2	70	1550	1550	11.55	11.56	1.79	1.91
17	2	65	1550	1550	11.13	11.34	1.73	1.94

Table 11. COMPARISON OF PREDICTED AND MEASURED PERFORMANCE OF INTERNATIONAL HARVESTER 130 ON STUBBLE PLOT.

Run No.	Gear	Cone Index psi	Engine Speed rpm		Drawbar Horsepower		Fuel Consumption gal/hr	
			Measured	Predicted	Measured	Predicted	Measured	Predicted
1	1	175	1050	1050	3.97	3.96	0.63	0.70
2	1	159	1050	1050	3.94	4.01	0.76	0.70
3	1	195	1200	1201	5.08	5.27	0.94	0.90
4	1	170	1200	1197	5.05	5.16	1.07	0.90
5	1	108	1400	1399	5.69	5.89	1.13	1.17
6	1	132	1400	1397	5.80	5.96	1.31	1.16
7	1	158	1400	1398	6.46	6.59	1.15	1.19
8	1	198	1400	1401	8.28	8.35	1.30	1.31
9	1	205	1400	1400	8.31	8.34	1.14	1.31
10	1	133	1550	1548	8.78	8.78	1.38	1.53
11	1	133	1550	1548	8.78	8.78	1.21	1.53
12	2	173	1200	1200	10.43	10.53	1.55	1.29
13	2	173	1200	1200	10.43	10.53	1.33	1.29
14	2	175	1400	1400	13.42	13.42	1.52	1.70
15	2	181	1400	1400	13.07	13.09	1.54	1.66
16	2	170	1550	1549	14.36	14.47	1.83	1.93
17	2	205	1550	1549	14.63	14.63	1.72	1.92

Table 12. COMPARISON OF PREDICTED AND MEASURED PERFORMANCE OF CASE 2470 ON STUBBLE PLOT.

Run No.	Gear	Cone Index psi	Engine Speed rpm		Drawbar Horsepower		Fuel Consumption gal/hr	
			Measured	Predicted	Measured	Predicted	Measured	Predicted
1	2 Lo	136	2200	2202	54.62	54.81	6.14	6.51
2	2 Lo	158	2000	2003	49.96	49.82	5.61	5.62
3	2 Lo	175	1800	1798	40.52	40.87	4.22	4.60
4	2 Int	195	2175	2174	35.35	38.30	5.92	5.65
5	2 Int	115	2150	2149	68.83	68.71	7.43	7.40
6	2 Int	118	2000	2000	95.44	98.41	9.01	8.65
7	2 Int	129	1800	1798	67.16	68.57	6.26	6.36
8	3 Lo	151	2150	2143	91.71	92.77	8.70	8.58
9	3 Lo	131	2000	2000	82.99	82.55	7.68	7.69
10	3 Lo	201	1800	1800	73.30	73.16	6.80	6.46
11	3 Lo	105	1800	1801	69.36	69.15	6.37	6.56
12	2 Hi	165	1800	1798	95.97	96.60	7.78	7.92
13	2 Hi	131	1600	1600	74.98	74.49	6.25	6.35

Table 13. STATISTICAL COMPARISON OF MEASURED AND PREDICTED FUEL CONSUMPTION, GAL/HR.

Tractor and Field Type	Mean Difference $\bar{d}$	Standard Deviation $s_{\bar{d}}$	95% Confidence Interval Estimate of True Mean Difference, $\mu_{\bar{d}}$
MF 235 Summer fallow	-0.84	0.05	$-0.94 \leq \mu_{\bar{d}} \leq -0.73$
MF 235 Pasture	-0.82	0.06	$-0.95 \leq \mu_{\bar{d}} \leq -0.69$
AC 170 Summer fallow	-2.14	0.23	$-2.61 \leq \mu_{\bar{d}} \leq -1.65$
AC 170 Pasture	-2.02	0.05	$-2.13 \leq \mu_{\bar{d}} \leq -1.91$
AC 6040 Summer fallow	-2.68	0.04	$-2.76 \leq \mu_{\bar{d}} \leq -2.60$
AC 6040 Pasture	-2.66	0.06	$-2.80 \leq \mu_{\bar{d}} \leq -2.52$
Ford 3000 Summer fallow	0.05	0.05	$-0.04 \leq \mu_{\bar{d}} \leq 0.15$
IH 130 Summer fallow	0.04	0.04	$-0.04 \leq \mu_{\bar{d}} \leq 0.13$
IH 130 Stubble	-0.04	0.04	$-0.12 \leq \mu_{\bar{d}} \leq 0.04$
Case 2470 Stubble	0.01	0.07	$-0.13 \leq \mu_{\bar{d}} \leq 0.16$

fuel flow meter level when the observations of fuel level were made. The 95 percent confidence interval estimates of true mean difference in measured and predicted fuel consumption for the remaining tractors were very reasonable.

From a small interval estimate it can be concluded that the difference between measured and predicted fuel consumption was consistent for each tractor. Consistency is more important than accuracy when predicting fuel consumption since accuracy can be increased by including the mean difference ($\bar{d}$) in the prediction equation.

Several factors could have caused the predicted values of fuel consumption to be lower than the measured values for the Massey-Ferguson 235, Allis-Chalmers 170 and, Allis-Chalmers 6040 tractors. The predicted values were not necessarily incorrect since it is possible that the measured values did not accurately represent the actual fuel consumption. Two obvious possibilities that could have caused errors were: first, the test plots were only 30 meters long for these tractors compared to 60 meters long for the Ford 3000 and Case 2470; secondly, these tractors used the single connection flowmeter installation compared to the twin connection installation for the Ford and Case tractors. The International 130, being gasoline powered, did not require a return fuel line and thus, a second connection. The longer test strip would cause errors in operating and reading the flowmeter to have less effect since they would be averaged over a longer time period. Possibly the single installation flowmeter did not accurately measure net fuel flow to the engine during the test run.

It was assumed that engine fuel flow had reached equilibrium before the test began. If this was not the case, there could be a net gain of fuel in the fuel system during the run. The fuel return lines would be only partially full at the beginning of the run and completely full at the end of the run. Since only a small amount of fuel was consumed during each run, a small gain of fuel in the system during the test would greatly affect the measured fuel consumption.

Exact predictions of fuel consumption can only be expected if the tractor's Nebraska Test Report data represents the actual engine performance of the tractor being tested. For half of the tractors tested, the Nebraska Test Report data allowed fuel consumption to be predicted both accurately and consistently; the greatest mean difference in predicted-measured fuel consumption was 0.05 gal/hr, though the predicted-measured error for the remaining tractors had several potential sources, the Nebraska Test Report data must be included in this potential error list. The coefficients α and c used in the fuel consumption equation are determined from the tractor's Nebraska Test Report. The tractors tested at Nebraska are probably in excellent condition and represent the maximum performance that could be expected of a tractor of that make and model. Therefore, it would be expected that predicted fuel consumption would be less than the measured fuel consumption for the tractors tested in this study.

VII. EFFECT OF TRACTIVE PERFORMANCE ON ENERGY REQUIREMENTS

The computer model was used to determine the effect of tractive performance and soil strength on fuel consumption. A set of curves was developed showing coefficient of traction, tire efficiency and fuel consumption as a function of wheel slip for three different soil strengths. Since tractor geometrical and physical parameters affect tractive performance and, thus, fuel consumption, the curves were valid only for the tractor for which they were derived. Nebraska Tractor Test Report 1149 for a Case 870 manual diesel tractor was used to supply the needed tractor information.

Figures 20 through 22 show the tractive performance and fuel requirements at three soil strengths. The cone indices were 50, 100 and 150 psi, respectively, for Figures 20 through 22. In each figure the tractor was operating in third gear at three miles per hour. Increases in drawbar horsepower were obtained by increasing the allowable wheel slip which, in turn, increased drawbar pull. As wheel slip increased, engine speed also must increase to maintain the desired field speed.

The tractive performance parameters plotted were tire efficiency and coefficient of traction. Tire efficiency is the ratio of output power to input power. The output power included not only the drawbar horsepower, but also

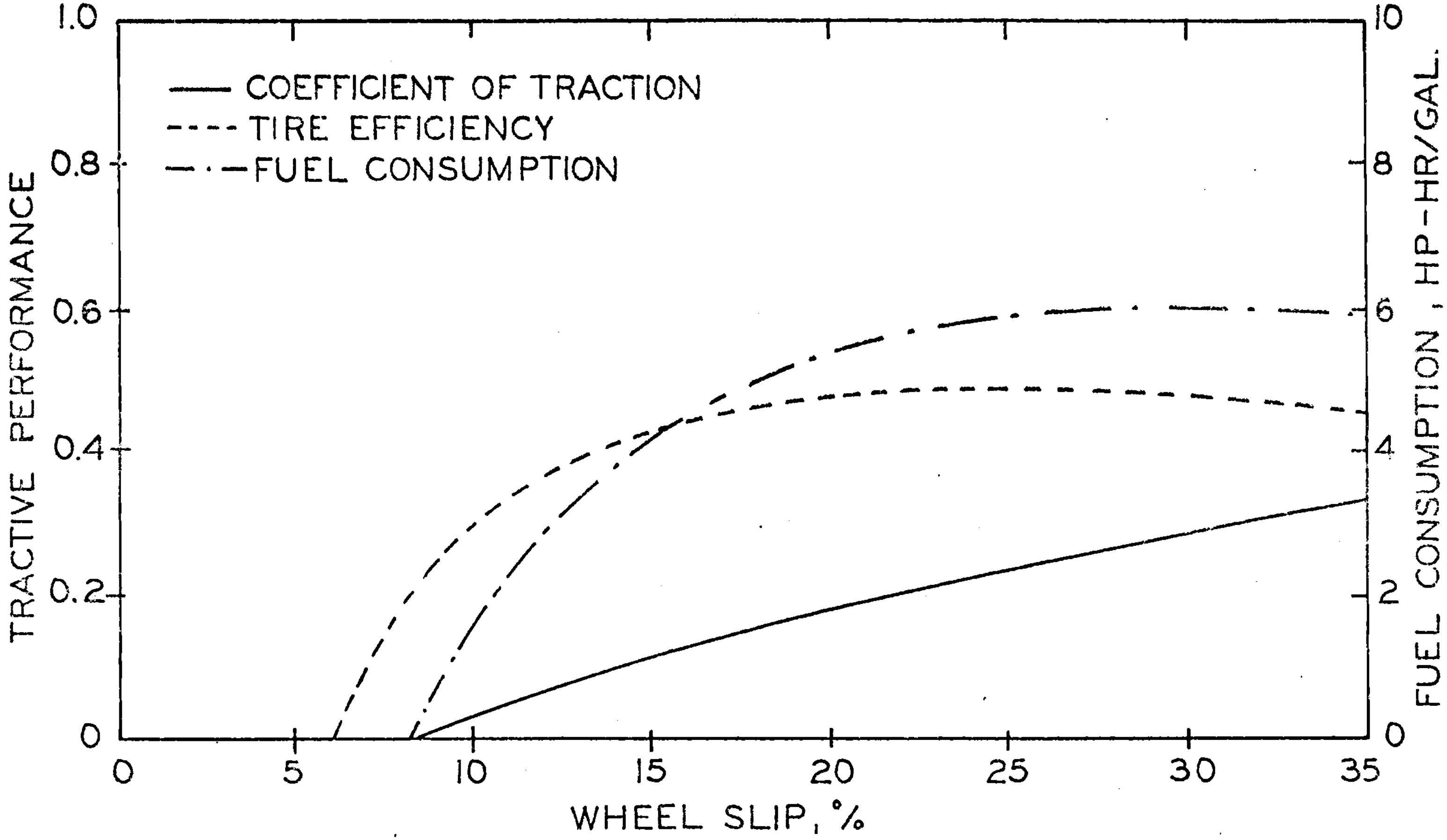


Figure 20. Predicted tractive performance and fuel consumption versus wheel slip of the Case 870 in third gear at three mph on soil with a cone index of 50 psi.

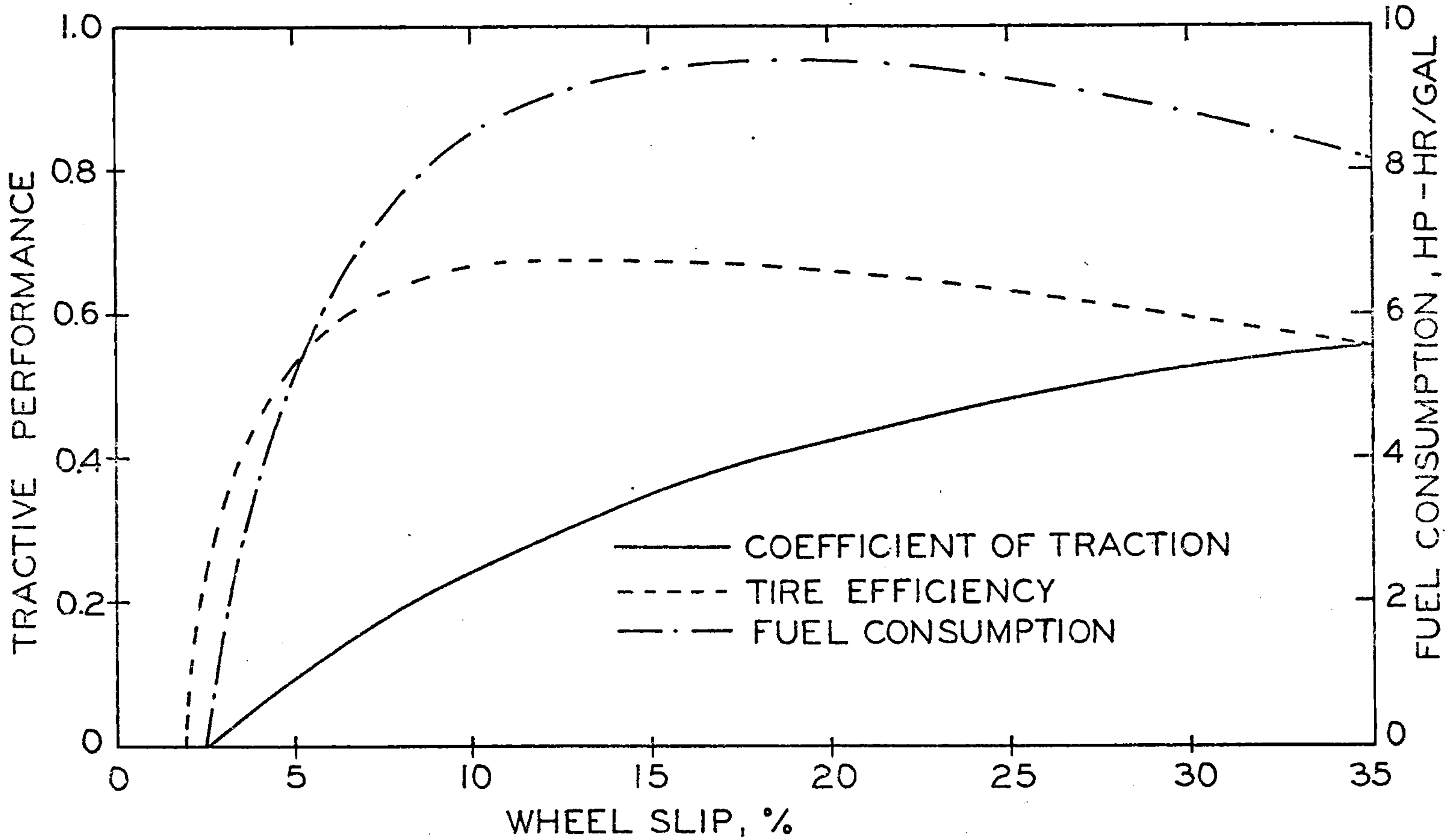


Figure 21. Predicted tractive performance and fuel consumption versus wheel slip of the Case 870 in third gear at three mph on soil with a cone index of 100 psi.

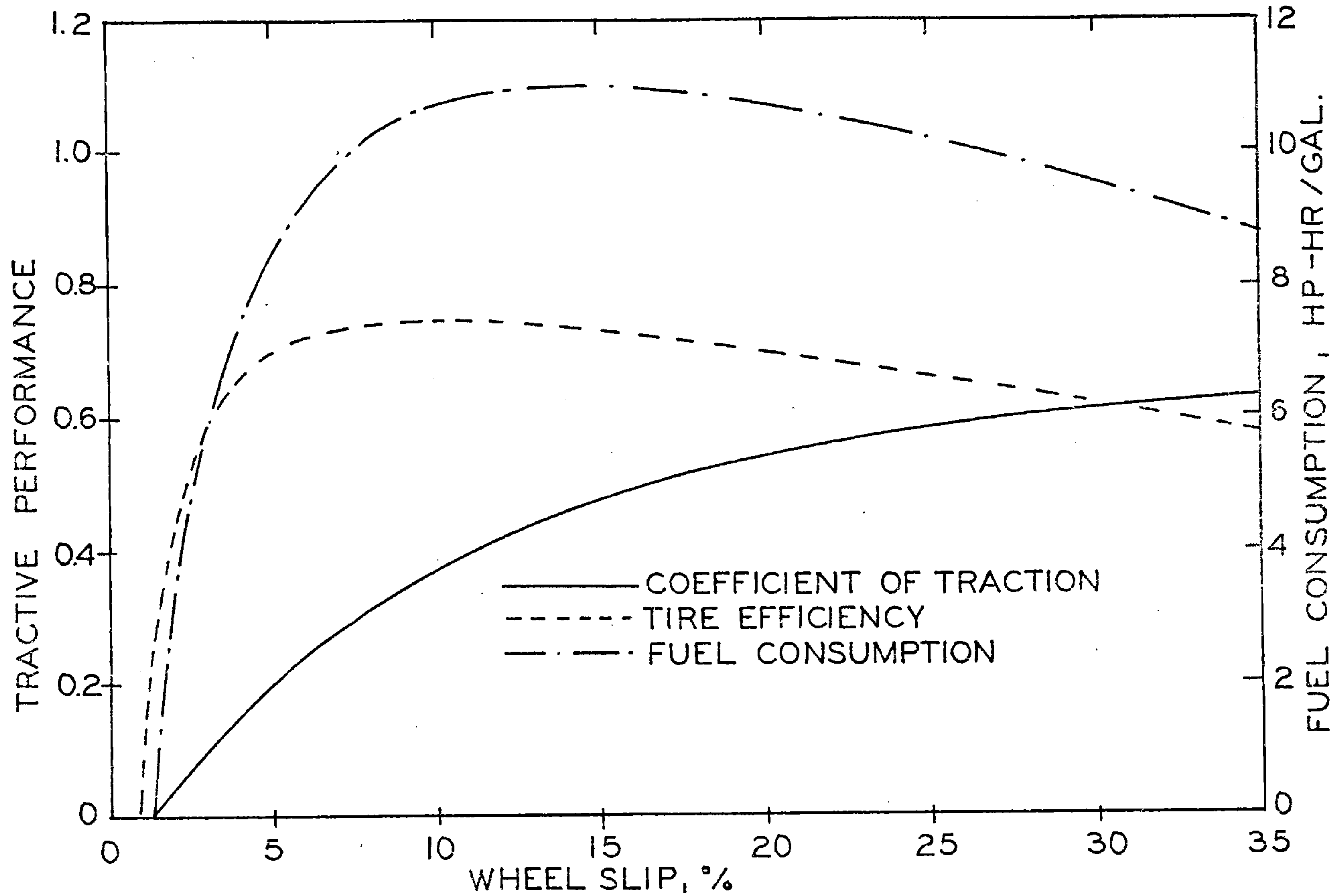


Figure 22. Predicted tractive performance and fuel consumption versus wheel slip of the Case 870 in third gear at three mph on soil with a cone index of 150 psi.

the power required to overcome the front axle rolling resistance. The front axle rolling resistance was included since the tractor tire cannot distinguish between it and drawbar load. Also, the engine must produce enough power for both drawbar and front axle rolling resistance requirements.

The coefficient of traction shown is the ratio of drawbar pull to dynamic rear axle weight. This term is a measure of the tractor's tractive performance. The fuel consumption parameter is drawbar horsepower-hour per gallon of fuel. This parameter determines the rate at which the tractor is doing work per gallon of fuel used. Larger values indicate increased fuel economy.

From Figures 20 through 22 it can be seen that maximum tire efficiency occurs at lower wheel slips than maximum fuel economy. Also, the coefficient of traction is relatively small at the wheel slip associated with maximum tire efficiency. The maximum fuel economy occurs at 27, 19 and 13 percent wheel slip, while maximum tire efficiency occurs at 25, 13 and 9 percent wheel slip for cone indices of 50, 100, and 150 psi, respectively. The slope of the tire efficiency curve between maximum tire efficiency wheel slip and maximum fuel economy wheel slip is relatively flat. The change in tire efficiency is relatively small between the two wheel slips. A substantial increase (approximately

30 percent) occurs in the coefficient of traction between maximum tire efficiency wheel slip and maximum fuel economy wheel slip for the two firmer soils. This means that approximately 30 percent more pull is being developed by increasing the wheel slip to the maximum fuel economy level. Other factors also influence the desired wheel slip. Normally 15 to 20 percent wheel slip is designated as the maximum acceptable wheel slip to reduce tire wear to acceptable levels. Higher wheel slips can also adversely affect soil structure.

Figure 23 shows fuel consumption versus wheel slip for the three soil cone indices. The maximum fuel economy wheel slip decreases as soil strength increases. Also, the slope of the fuel consumption curve between zero and maximum fuel economy wheel slip rapidly increases with increasing soil strength. In this range a small increase in wheel slip produces a large increase in fuel economy.

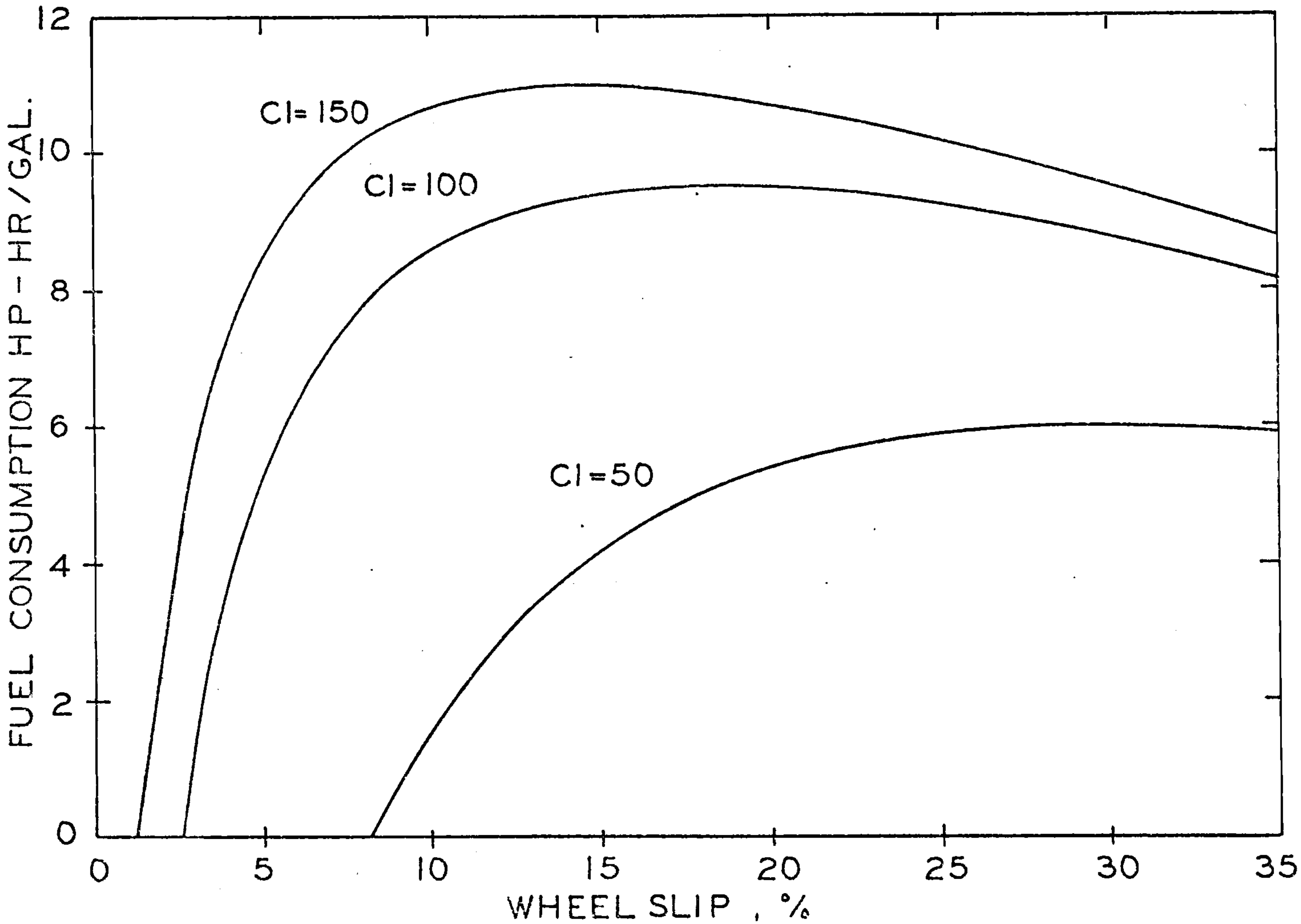


Figure 23. Fuel consumption versus wheel slip at three soil cone indices for the Case 870 in third gear at three mph.

VIII. CONCLUSIONS

A computer model was developed to predict tractive performance and energy requirements agricultural tractors. The model was shown to predict performance reasonably well for agricultural tractors. Acceptable agreement was obtained when comparing the measured tractive performance from the summer fallow test sites with the predicted performance for these sites (Figures 10, 12, 14, 16 and 17). The test results obtained on the pasture and stubble plots could not be duplicated by the modeling equations when using the measured cone index values (Figures 11, 13, 15, 18 and 19). The measured tractive performance could be approximated by substituting lower values of cone index into the modeling equations, since initially the model had over-estimated tractive performance.

Considerable data scatter was present in the measured tractive performance (Figures 10 through 19). This was expected since soil conditions were not homogeneous throughout the test plots. Also the experimental method was not designed to produce extremely accurate results. Its purpose was to determine the relative magnitude of tractive performance and trends that might develop.

The ability of the model to predict measured fuel consumption was dependent on the method used in measuring fuel consumption. For tractors that required the twin connection flow meter installation, the model was able to predict the measured values of fuel consumption both accurately and consistently (Table 13). The predicted and measured values of fuel consumption were exceptionally close for the Ford 3000, International Harvester 130 and Case 2470 tractors. The remaining tractors, which required the single connection flow meter installation, were not as accurately modeled. Although not accurately modeled, the difference between measured and predicted fuel consumption was consistent for each tractor.

Without further tests it cannot be confidently determined whether the error in predicting the measured fuel consumption was caused by the experimental method, the source of data, or the modeling technique. The tractors using the single connection flow meter installation also were tested on the shorter (30 meters) test strip because of space limitations. It is possible that the data for the tractors using the twin connection flow meter installation are actually in error. If this was the case, the measured values of fuel consumption, quite possibly, should be higher. This would put them into agreement with the results of the other tractors.

A major factor affecting the accuracy of the fuel consumption prediction was the mechanical condition of the test tractor. The coefficients α and c used in the fuel consumption equation are determined from the tractor's Nebraska Test Report. The tractors tested at Nebraska are probably in excellent condition and represent the maximum performance that could be expected of a tractor of that make and model. Therefore, it would be expected that predicted fuel consumption would be less than the measured fuel consumption for the tractors tested in this study.

The accuracy of the fuel consumption predictions were also affected by tractive performance predictions. The predicted drive wheel horsepower was converted to equivalent PTO horsepower by dividing by tire efficiency, to obtain axle horsepower, and then by 0.96 to obtain PTO horsepower. Drawbar horsepower was measured during the field tests, but the equipment necessary to measure tire efficiency was not available. The error in the predicted tire efficiency was, therefore, unknown and its subsequent effect on fuel consumption was also unknown.

The tire efficiency predictions seemed reasonable. If one is willing to accept this efficiency, along with the measured fuel consumption of the twin connection flow meter installation for diesel tractors and the single connection for gasoline tractors, the model is in reasonably good agreement with the measured results.

Maximum fuel economy was shown to occur at higher wheel slips than does maximum tractive efficiency in Figures 20 through 22. The wheel slips at which both maximum fuel economy and maximum tire efficiency occur decrease as soil strength increases (cone index). At low wheel slips, both the tire efficiency and fuel economy rapidly increase with increasing wheel slip. The change in tire efficiency between the maximum tire efficiency wheel slip and the maximum fuel economy wheel slip is relatively small. Between these same two wheel slips, the coefficient of traction can increase by 30 percent.

The fuel consumption predictions can be greatly influenced by the tractor data inputed to the model. If the tractor being modeled is available, the static loaded radius of the drive wheels should be measured and inputed directly to the model. If this cannot be done, the model uses the static loaded radius reported in ASAE Recommendation R220.3. The two values of static loaded radius may vary, depending on the tire pressure and load of the test tractor. It is important that an accurate measurement of the static loaded radius be used, since it determines the engine speed required to produce the desired travel speed.

The model reasonably predicts tractor performance if the following assumptions are acceptable:

1. The tractor is equipped with almost new tires (75 percent tread or more).
2. The ratio of unloaded tire section width to unloaded overall tire diameter is approximately 0.30.

3. The ratio of tire deflection to tire section height is approximately 0.20.
4. Both sides of a driving axle have equal wheel slips.
5. Tractor weight is equally distributed between the left and right sides (symmetry).
6. Soil conditions are homogeneous and the cone penetrometer accurately describes soil strength.
7. The tractor is operating on level ground.
8. The load is applied at the drawbar and is horizontal and parallel to the drawbar.
9. The fuel consumption coefficients, α and c, determined from the Nebraska Test Report accurately describe the test tractor's engine performance.

If these assumptions are correct, the predicted results should be indicative of the tractors' actual performance.

IX. SUGGESTED FUTURE RESEARCH

Possibilities for future research are vast. Several questions have arisen from this study. The cone penetrometer reading that should be inserted into the model is not always clear. This is especially a problem on freshly tilled, highly compactible soils. This model made no attempt to determine the effect of dual tires operating together. They were modeled as two separate wheels each carrying one half of the load (Case 2470). Also, the effect on the rear wheel cone index when the front wheels traffic the same soil as the rear wheels, is unknown. The front wheel trafficking would be especially interesting for large four wheel drive tractors operating on tilled soils.

Another area of interest is determining the correct or appropriate cone index to use for pasture or stubble plots. The ground cover and surface roughness play a role in determining tractive performance.

It would be helpful if more field tests and PTO load tests could be conducted so that further validation of the fuel consumption portion of the model could be accomplished. If PTO performance was tested, the validity of using the Nebraska Test Reports to determine the coefficients α and c could be obtained.

Another possible area of study would be to use the computer model to develop a series of graphs showing the effects of tire sizes and number, wheelbase length, drawbar height, drive wheel arrangement (two- or four-wheel drive), and soil strength on tractive performance and fuel consumption of a standard tractor.

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APPENDIX

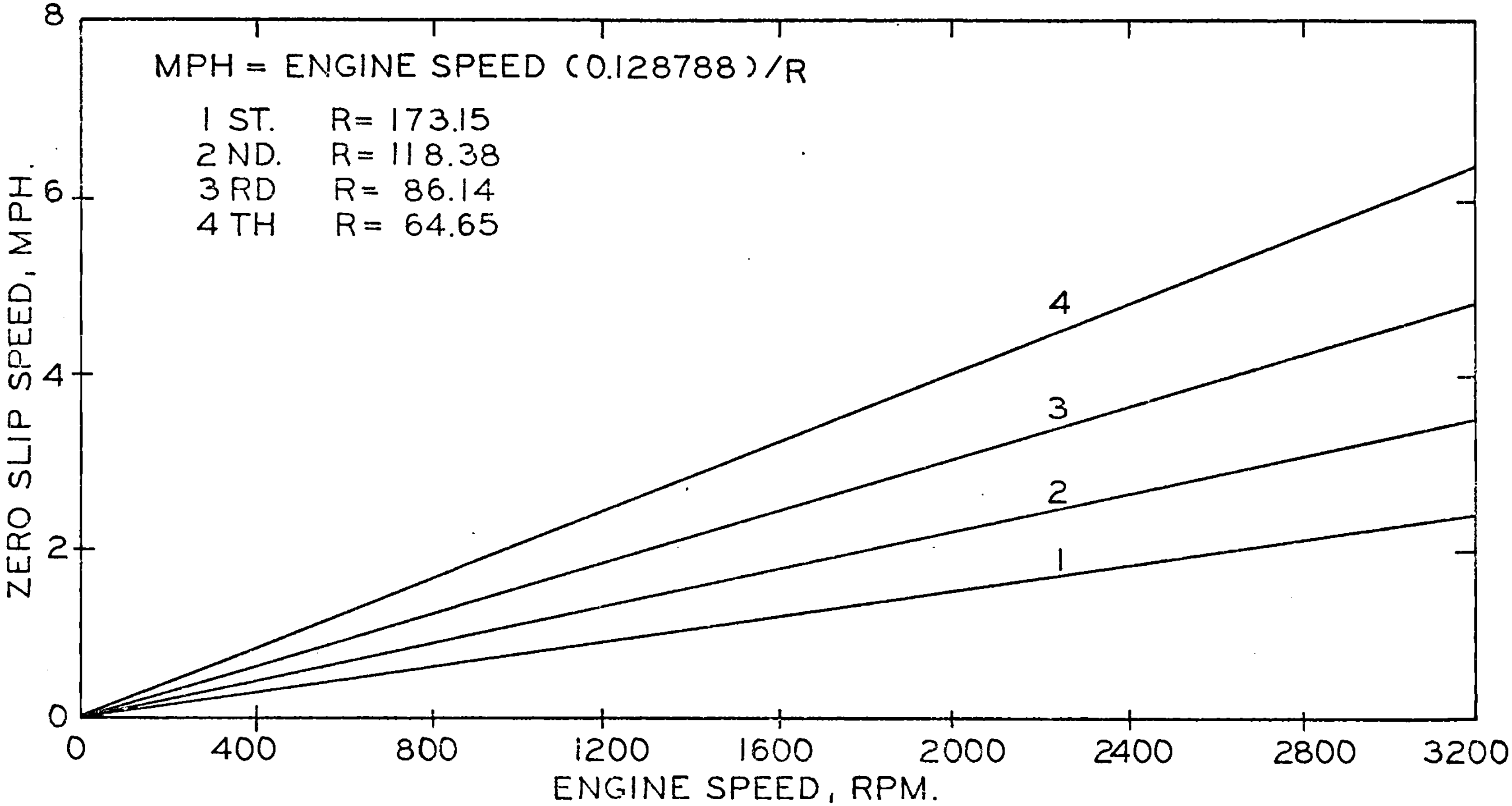

Figure A.1. Zero slip tractor speed versus engine speed and overall gear reduction (R) for each gear tested of the Massey-Ferguson 235 tractor.

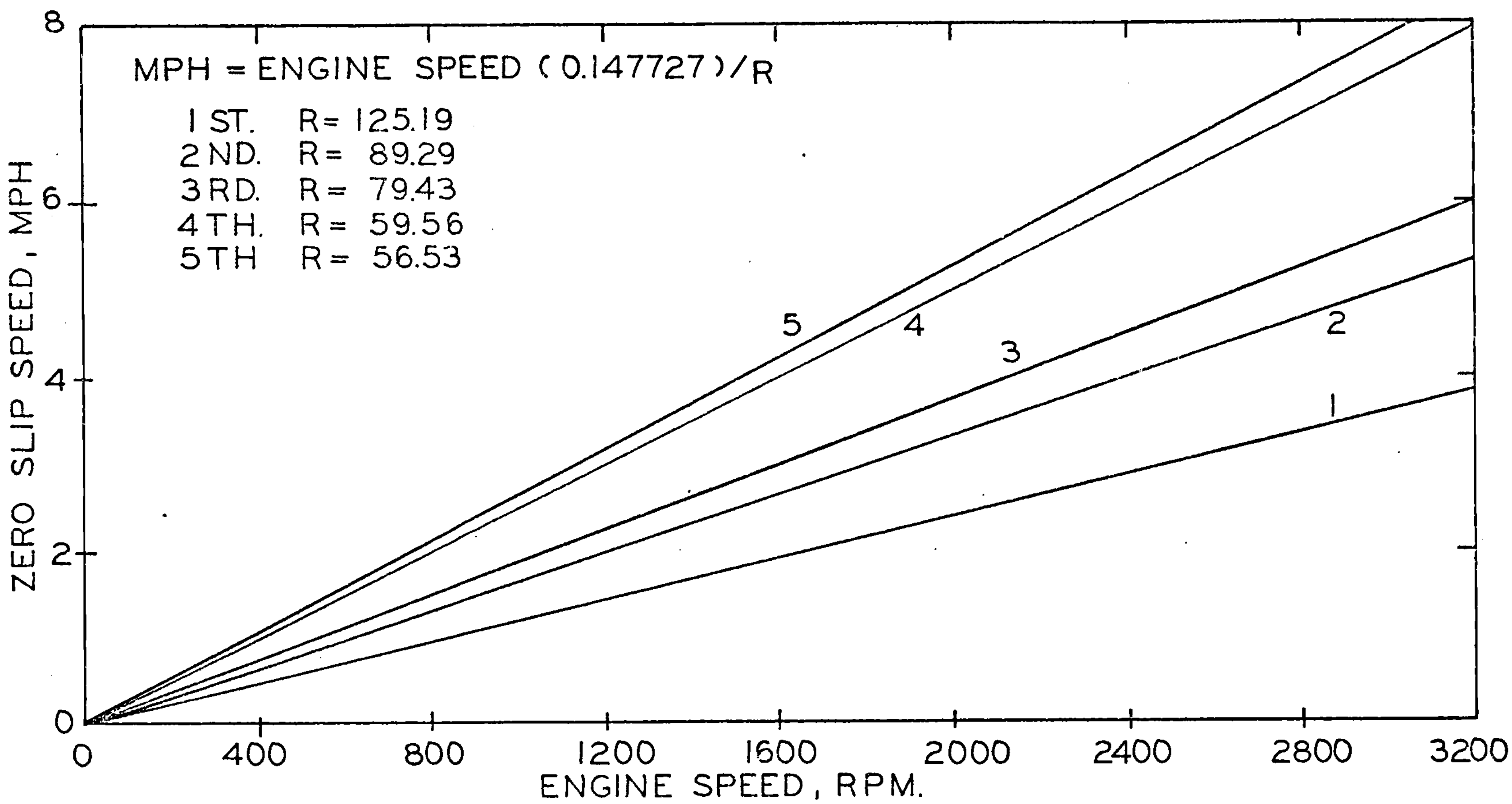

Figure A.2. Zero slip tractor speed versus engine speed and overall gear reduction (R) for each gear tested of the Allis-Chalmers 170 tractor.

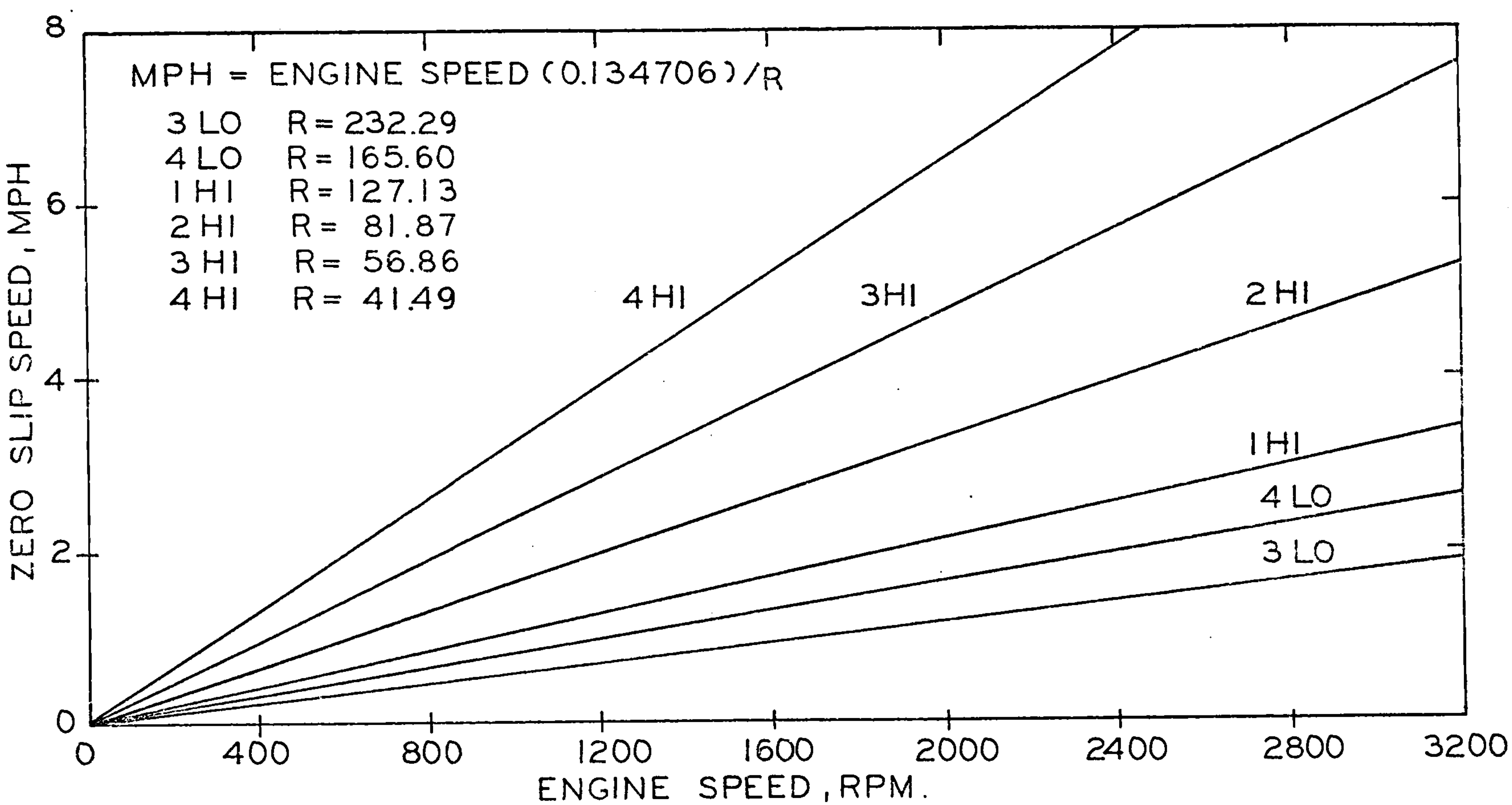

Figure A.3. Zero slip tractor speed versus engine speed and overall gear reduction (R) for each gear tested of the Allis-Chalmers 6040 tractor.

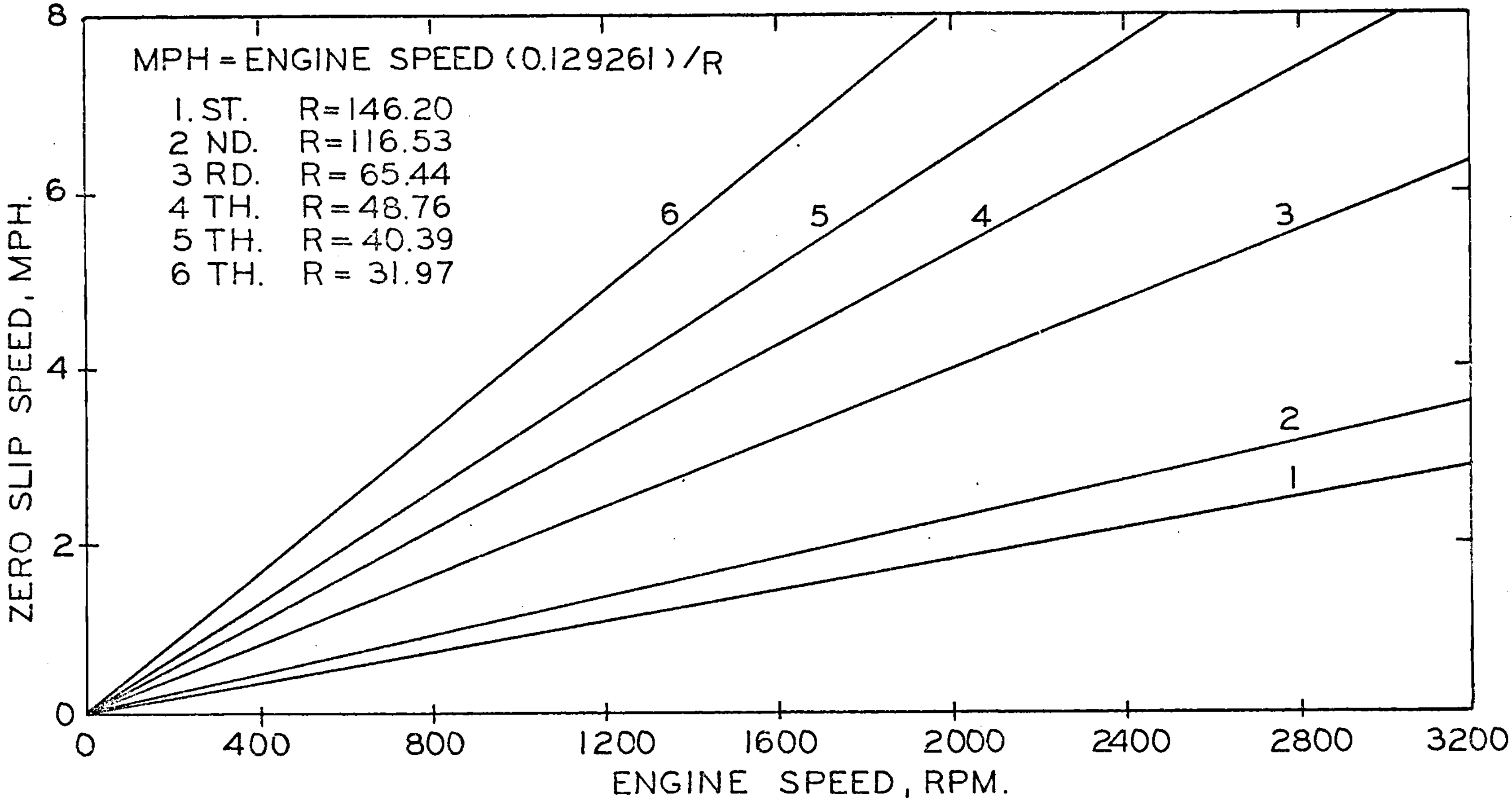

Figure A.4. Zero slip tractor speed versus engine speed and overall gear reduction (R) for each gear tested of the Ford 3000 tractor.

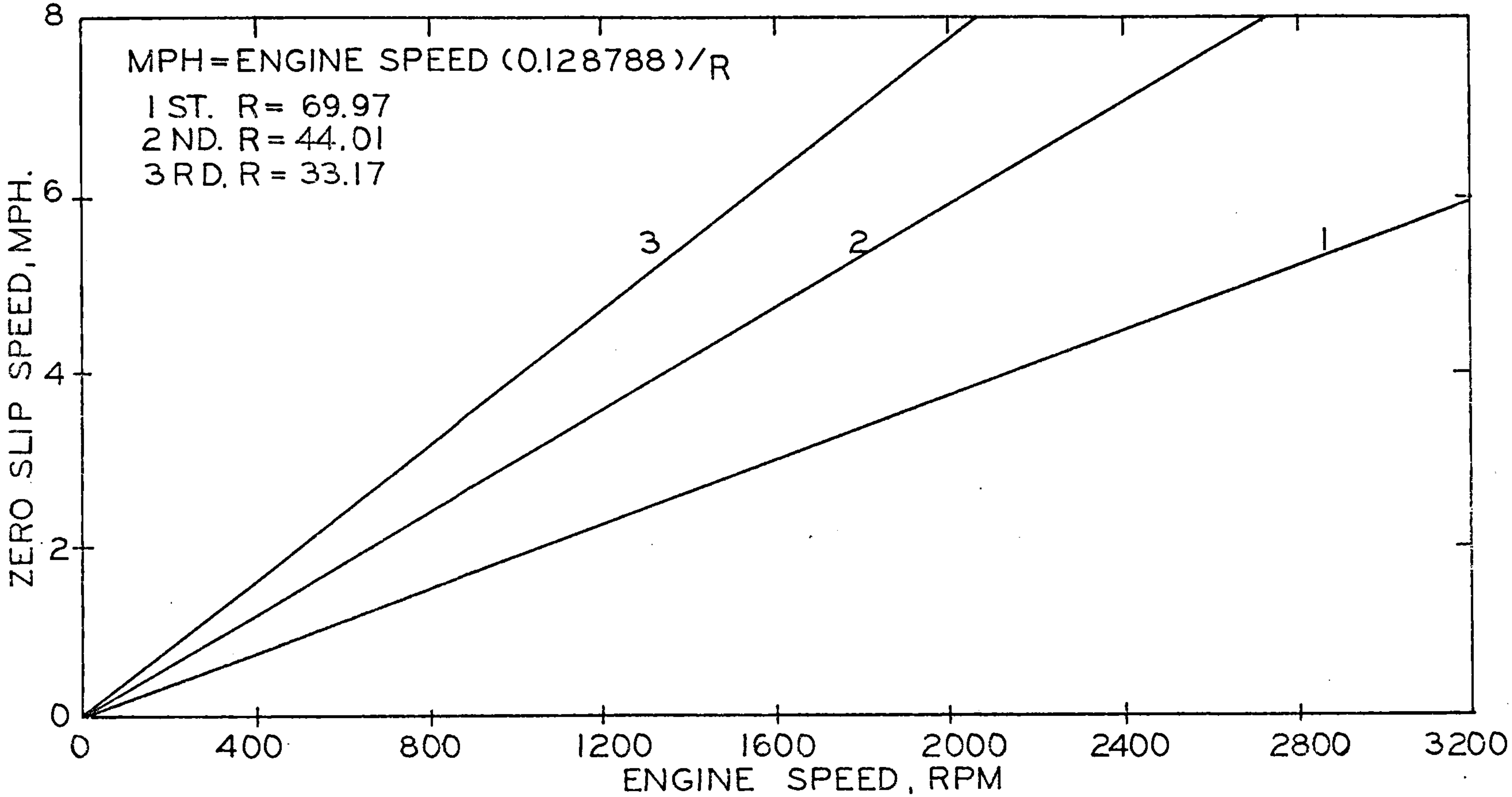

Figure A.5. Zero slip tractor speed versus engine speed and overall gear reduction (R) for each gear tested of the International Harvester 130 tractor.

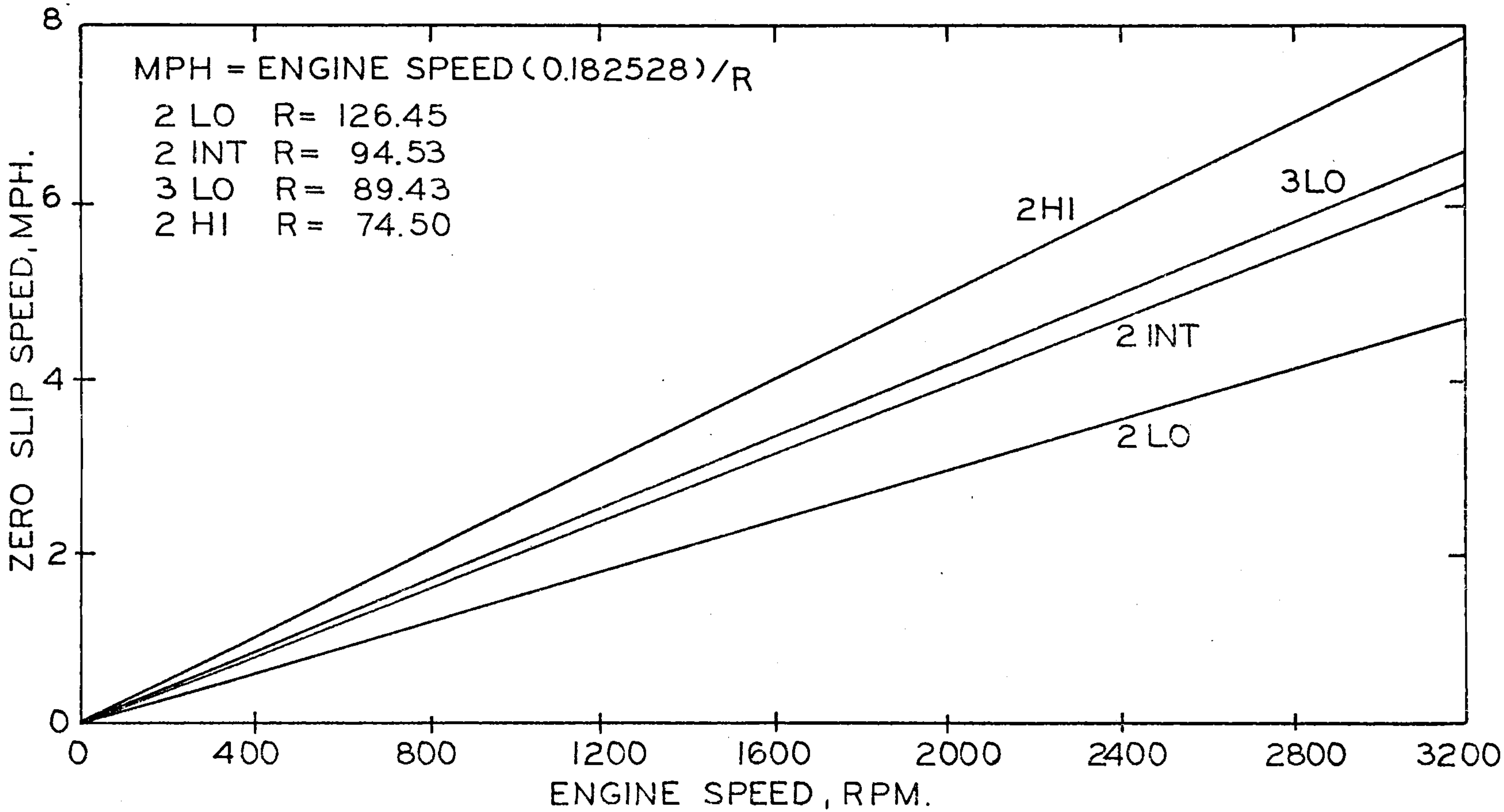

Figure A.6. Zero slip tractor speed versus engine speed and overall gear reduction (R) for each gear tested of the Case 2470 tractor.

Table A.1. GENERAL INFORMATION FOR THE TRACTORS FIELD TESTED.

	Massey-Ferguson 235	Allis-Chalmers 170	Allis-Chalmers 6040	Ford 3000	International Harvester 130	Case 2470
Drive Type	2-wheel	2-wheel	2-wheel	2-wheel	2-wheel	4-wheel
Drive Wheels	Singles	Singles	Singles	Singles	Singles	Duals
Tire Size						
Rear	13.6-28	18.4-28	14.9/13-28	16.9-24	11-24	18.4-34
Front	6.00-16	7.50-16	6.00-16	7.50-16	5.00-15	18.4-34
Tire Pressure (psi)						
Rear	12	11	12	12	10	12.5
Front	24	32	26	27	25	12.5
Drive Wheel Static Loaded Radius (in)	21.65	24.83	22.64	21.68	21.65	30.68
Static Wt (lb)						
Rear	2495	5335	4035	3690	2150	11910
Front	1620	2610	1960	1690	1045	14915
Drawbar Ht (in)	15	17.5	16	13.5	13	17.5
Wheel Base (in)	74.5	95.5	76	75.8	71	102
Engine Displacement (in^3)	153	235.9	152.7	175	123	504
Rated Speed (rpm)	2250	1800	2250	2000	1400	2200
Fuel Type	Diesel	Diesel	Diesel	Diesel	Gas	Diesel

Table A.2. MASSEY-FERGUSON 235 TEST RUNS AT FARM SERVICES SUMMER FALLOW PLOT.

Run No.	Gear	Engine Speed RPM	Drawbar Pull lb	Speed mph	Wheel Slip %	Drawbar Horsepower	Fuel Consumption gal/hr	Fuel Consumption hp hr/gal
1	2	2250	900	2.08	14.86	5.00	2.06	2.43
2	2	2250	1100	2.07	15.64	6.06	1.69	3.59
3	2	2000	1000	1.85	14.79	4.94	1.89	2.61
4	2	2000	800	1.93	11.39	4.11	1.71	2.41
5	2	1800	800	1.70	13.04	3.63	1.62	2.24
6	2	1800	800	1.76	10.28	3.75	1.61	2.32
7	3	2250	1100	2.62	22.09	7.69	2.14	3.59
8	3	2250	1200	2.66	20.84	8.52	2.17	3.92
9	3	2000	900	2.55	14.66	6.12	1.82	3.36
10	3	2000	1100	2.49	16.90	7.29	1.44	5.07
11	3	1800	500	2.47	8.33	3.29	2.01	1.63
12	3	1800	1200	2.22	17.43	7.11	1.74	4.09
13	4	2250	1600	3.20	28.70	13.64	2.50	5.45
14	4	2250	1800	3.24	27.67	15.56	2.43	6.41
15	4	2000	1200	3.17	20.56	10.13	1.94	5.23
16	4	2000	1200	3.17	20.56	10.13	1.83	5.53
17	4	1800	900	3.14	12.54	7.53	1.71	4.41
18	4	1800	900	3.23	10.03	7.74	1.98	3.92

Soil Cone Index, psi

Depth	0 ↔ 4 in	0 ↔ 6 in	0 ↔ 9 in
Before Tests	50	65	95
After Tests	71	80	101

Table A.3. MASSEY-FERGUSON 235 TEST RUNS AT FARM SERVICES PASTURE PLOT.

Run No.	Gear	Engine Speed RPM	Drawbar Pull lb	Speed mph	Wheel Slip %	Drawbar Horsepower	Fuel Consumption $\frac{gal}{hr}$	Fuel Consumption $\frac{hp\ hr}{gal}$
1	1	2250	1000	1.51	10.07	4.01	1.95	2.06
2	1	2000	1100	1.30	12.41	3.82	1.69	2.27
3	2	2250	1800	1.85	24.46	8.87	2.20	4.03
4	2	2250	1650	1.84	24.67	8.11	2.26	3.59
5	2	2000	1200	1.91	12.41	6.10	1.95	3.14
6	2	1800	800	1.89	3.74	4.02	1.15	3.48
7	2	1600	800	1.64	5.96	3.49	1.11	3.14
8	3	2250	1700	2.66	20.84	12.07	2.08	5.79
9	3	2000	1300	2.60	13.02	9.02	2.12	4.25
10	3	1800	1300	2.24	16.88	7.76	1.90	4.08
11	4	2250	1800	3.29	26.60	15.80	2.35	6.72
12	4	2000	1700	3.20	19.78	14.49	2.17	6.66
13	4	1800	1300	3.20	10.87	11.08	1.74	6.37

Soil Cone Index, psi

Depth	0 ↔ 4 in	0 ↔ 6 in	0 ↔ 9 in
Before Tests	455	464	
After Tests	457	461	

Table A.4. ALLIS-CHALMERS 170 TEST RUNS AT FARM SERVICES SUMMER FALLOW PLOT.

Run No.	Gear	Engine Speed RPM	Drawbar Pull lb	Speed mph	Wheel Slip %	Drawbar Horsepower	Fuel Consumption	
							$\frac{\text{gal}}{\text{hr}}$	$\frac{\text{hp hr}}{\text{gal}}$
1	2	1900	2400	2.91	7.60	18.59	3.66	5.09
2	2	1800	2300	2.74	8.03	16.80	4.10	4.10
3	2	1800	2300	2.74	8.03	16.80	2.05	8.20
4	2	1600	2350	2.44	7.83	15.29	6.64	2.30
5	2	1600	1900	2.50	5.41	12.69	3.19	4.14
6	2	1400	2200	2.20	5.02	12.91	4.19	3.08
7	3	1925	2500	3.18	11.18	21.20	3.46	6.14
8	3	1925	2400	3.21	10.31	20.55	5.46	3.76
9	3	1800	2200	3.02	9.70	17.73	3.29	5.39
10	3	1800	2300	2.96	11.70	18.13	4.83	3.76
11	3	1600	2200	2.70	9.43	15.81	3.12	5.07
12	3	1600	2400	2.62	11.92	16.78	4.64	3.62
13	4	1900	2800	4.22	10.43	31.51	5.74	5.49
14	4	1875	2800	4.19	9.82	31.32	3.14	9.98
15	4	1800	2600	4.07	8.91	28.20	4.15	6.79
16	4	1800	2650	4.04	9.44	28.57	5.50	5.19
17	4	1600	2500	3.67	7.60	24.45	3.74	6.53
18	4	1600	2500	3.63	8.61	24.18	4.69	5.16
19	5	1875	2600	4.39	10.51	30.41	4.92	6.18
20	5	1875	2800	4.33	11.63	32.33	3.39	9.54

Table A.4. (Continued)

Run No.	Gear	Engine Speed RPM	Drawbar Pull lb	Speed mph	Wheel Slip %	Drawbar Horsepower	Fuel Consumption gal/hr	Fuel Consumption hp hr/gal
21	5	1800	2550	4.33	7.95	29.44	6.19	4.76
22	5	1800	2700	4.19	10.84	30.20	3.71	8.14
23	5	1600	2500	3.67	13.00	24.45	4.99	4.90
24	5	1600	2500	3.81	8.81	25.42	3.63	6.70

Soil Cone Index, psi

Depth	0 ↔ 4 in	0 ↔ 6 in	0 ↔ 9 in
Before Tests	66	90	130
After Tests	74	120	148

Table A.5. ALLIS-CHALMERS 170 TEST RUNS AT FARM SERVICES PASTURE PLOT.

Run No.	Gear	Engine Speed RPM	Drawbar Pull lb	Speed mph	Wheel Slip %	Drawbar Horsepower	Fuel Consumption gal/hr	Fuel Consumption hp hr/gal
1	1	1950	2600	2.05	11.13	14.19	3.90	3.64
2	1	1950	2800	2.07	10.26	15.42	3.65	4.22
3	1	1800	2400	1.95	8.43	12.45	3.71	3.36
4	1	1800	2600	1.94	8.66	13.45	3.43	3.92
5	1	1600	2300	1.81	3.92	11.12	3.58	3.11
6	2	1950	2800	2.79	13.68	20.79	3.69	5.63
7	2	1950	2500	2.94	8.78	19.62	4.31	4.56
8	2	1800	2400	2.76	7.25	17.68	3.95	4.48
9	2	1600	2000	2.45	7.49	13.06	3.50	3.73
10	3	1950	1800	3.39	6.55	16.27	3.92	4.15
11	3	1800	1700	3.12	6.77	14.15	3.82	3.70
12	3	1600	2100	2.72	8.70	15.22	3.70	4.12
13	4	1800	1300	4.27	4.27	14.82	4.07	3.64
14	4	1600	1400	3.77	5.00	14.08	3.59	3.92
15	4	1600	1700	3.77	5.00	17.09	3.85	4.44
16	5	1600	1500	3.81	8.81	15.25	3.89	3.92
17	5	1400	1450	3.46	5.45	13.38	3.30	4.06

Soil Cone Index, psi

Depth	0 ↔ 4 in	0 ↔ 6 in	0 ↔ 9 in
Before Tests	279	356	393
After Tests	300	350	387

Table A.6. ALLIS-CHALMERS 6040 TEST RUNS AT FARM SERVICES SUMMER FALLOW PLOT.

Run No.	Gear	Engine Speed RPM	Drawbar Pull lb	Speed mph	Wheel Slip %	Drawbar Horsepower	Fuel Consumption	
							$\frac{gal}{hr}$	$\frac{hp\ hr}{gal}$
1	3 Lo	2200	1800	1.14	10.88	5.46	3.83	1.43
2	4 Lo	2200	2700	1.38	23.17	9.90	4.12	2.41
3	1 Hi	2200	2100	2.00	14.08	11.22	4.09	2.74
4	1 Hi	2200	2000	2.01	13.82	10.72	4.24	2.53
5	1 Hi	2000	2300	1.78	16.01	10.92	3.95	2.77
6	1 Hi	2000	2400	1.72	18.79	11.01	4.14	2.66
7	1 Hi	1800	2300	1.61	15.85	9.85	3.50	2.82
8	1 Hi	1800	1700	1.74	8.61	7.90	3.91	2.02
9	2 Hi	2200	1900	3.12	13.78	15.88	4.46	3.55
10	2 Hi	2200	2275	3.05	15.74	18.51	4.36	4.25
11	2 Hi	2000	2300	2.73	17.10	16.73	4.36	3.84
12	2 Hi	2000	2300	2.62	20.35	16.08	4.55	3.54
13	2 Hi	1800	1950	2.58	12.85	13.42	4.22	3.18
14	2 Hi	1800	2300	2.26	23.69	13.86	4.03	3.44
15	3 Hi	2200	2100	4.33	16.92	24.25	4.71	5.14
16	3 Hi	2200	2400	4.25	18.51	27.18	5.20	5.23
17	3 Hi	2000	2400	3.95	16.68	25.26	5.10	4.95
18	3 Hi	2000	2400	3.84	19.06	24.54	4.96	4.95
19	3 Hi	1800	2100	3.57	16.28	19.99	4.13	4.84
20	3 Hi	1800	2300	3.50	18.04	21.44	4.52	4.74
21	4 Hi	1825	2100	4.86	17.93	27.23	4.96	5.49
22	4 Hi	1825	2100	4.90	17.34	27.43	5.00	5.49

Table A.6. (Continued)

Soil Cone Index, psi			
Depth	0 ↔ 4 in	0 ↔ 6 in	0 ↔ 9 in
Before Tests	72	97	140
After Tests	85	137	155

Table A.7 ALLIS-CHALMERS 6040 TEST RUNS AT FARM SERVICES PASTURE PLOT.

Run No.	Gear	Engine Speed RPM	Drawbar Pull lb	Speed mph	Wheel Slip %	Drawbar Horsepower	Fuel Consumption gal/hr	Fuel Consumption hp hr/gal
1	3 Lo	2200	1500	1.19	6.41	4.78	3.33	1.43
2	4 Lo	2200	1950	1.54	14.00	8.00	4.03	1.99
3	1 Hi	2200	1800	2.10	10.04	10.07	4.28	2.35
4	1 Hi	2200	1900	2.02	13.30	10.24	4.13	2.48
5	1 Hi	2000	1500	2.00	5.77	7.99	3.94	2.03
6	1 Hi	1800	2000	1.62	15.01	8.65	3.75	2.31
7	2 Hi	2200	2250	2.86	21.10	17.13	4.28	4.01
8	2 Hi	2000	2000	2.74	16.77	14.61	4.29	3.41
9	2 Hi	1800	2000	2.54	14.17	13.56	4.15	3.27
10	3 Hi	2200	2000	4.30	17.46	22.94	4.68	4.90
11	3 Hi	2000	2000	4.19	11.51	22.37	4.28	5.23
12	3 Hi	1800	2000	3.57	16.28	19.04	4.37	4.36
13	4 Hi	1850	2500	4.86	19.04	32.42	5.29	6.12

Soil Cone Index, psi

Depth	0 ↔ 4 in	0 ↔ 6 in	0 ↔ 9 in
Before Tests	370	444	459
After Tests	329	392	

Table A.8. FORD 3000 TEST RUNS AT HYSLOP FARM SUMMER FALLOW PLOT.

Run No.	Gear	Engine Speed RPM	Drawbar Pull lb	Speed mph	Wheel Slip %	Drawbar Horsepower	Fuel Consumption	
							$\frac{gal}{hr}$	$\frac{hp\ hr}{gal}$
1	1	2200	1300	1.67	13.94	5.80	1.20	4.85
2	2	2200	1300	2.16	11.57	7.48	1.10	6.79
3	2	2200	900	2.20	9.85	5.28	1.05	5.04
4	3	2150	1500	3.42	19.36	13.70	1.40	9.80
5	3	2000	1400	3.39	14.21	12.63	1.38	9.15
6	3	1500	1200	2.70	9.04	8.62	1.01	8.55
7	3	1000	700	1.87	5.38	3.49	0.45	7.84
8	4	2100	1700	4.66	16.29	21.13	2.38	8.88
9	4	2000	1600	4.49	15.33	19.15	1.68	11.40
10	4	1500	1500	3.51	11.63	14.05	1.55	9.05
11	4	1000	1000	2.41	9.28	6.41	0.70	9.22
12	5	2100	1850	5.52	17.82	27.25	2.35	11.60
13	5	2000	1800	5.31	17.12	25.46	2.17	11.76
14	5	1500	1600	4.12	14.24	17.57	1.33	13.20
15	5	1000	1300	2.91	9.23	10.07	0.74	13.59
16	6	1600	1700	5.33	17.67	24.15	2.45	9.87
17	6	1500	1600	4.97	18.04	21.21	1.86	11.40
18	6	1000	1400	3.36	17.02	12.53	1.14	10.97

Soil Cone Index, psi

Depth	0 ↔ 4 in	0 ↔ 6 in	0 ↔ 9 in
Before Tests	60	71	112
After Tests	80	87	118

Table A.9. INTERNATIONAL HARVESTER 130 TEST RUNS AT HYSLOP FARM SUMMER FALLOW PLOT.

Run No.	Gear	Engine Speed RPM	Drawbar Pull lb	Speed mph	Wheel Slip %	Drawbar Horsepower	Fuel Consumption $\frac{gal}{hr}$	Fuel Consumption $\frac{hp\ hr}{gal}$
1	1	1200	800	2.00	9.59	4.26	1.09	3.92
2	1	1200	1400	1.76	20.27	6.58	1.32	4.99
3	1	1200	1400	1.71	22.49	6.39	1.05	6.10
4	1	1400	1400	2.05	20.60	7.64	1.39	5.49
5	1	1400	1600	1.92	25.61	8.18	1.44	5.70
6	1	1400	1300	2.02	21.57	7.01	1.51	4.63
7	1	1400	1300	2.01	22.04	6.97	1.50	4.63
8	1	1550	1350	2.19	23.13	7.90	1.64	4.81
9	1	1550	1400	2.00	29.79	7.48	1.64	4.57
10	2	1200	1000	3.04	13.52	8.10	1.55	5.23
11	2	1200	1000	3.06	12.75	8.17	1.25	6.53
12	2	1400	1100	3.44	16.01	10.10	1.41	7.19
13	2	1400	1000	3.50	14.69	9.32	1.67	5.60
14	2	1500	1200	3.59	18.24	11.48	1.95	5.88
15	2	1500	1150	3.73	15.07	11.43	1.53	7.51
16	2	1550	1200	3.61	20.41	11.55	1.79	6.45
17	2	1550	1225	3.48	23.28	11.36	1.73	6.59

Soil Cone Index, psi

Depth	0 ↔ 4 in	0 ↔ 6 in	0 ↔ 9 in
Before Tests	63	82	
After Tests	78	87	90

Table A.10. INTERNATIONAL HARVESTER 130 TEST RUNS AT HYSLOP FARM STUBBLE PLOT.

Run No.	Gear	Engine Speed RPM	Drawbar Pull lb	Speed mph	Wheel Slip %	Drawbar Horsepower	Fuel Consumption gal/hr	Fuel Consumption hp hr/gal
1	1	1050	800	1.86	3.81	3.97	0.63	6.27
2	1	1050	800	1.85	4.33	3.94	0.76	5.23
3	1	1200	900	2.12	4.15	5.08	0.94	5.43
4	1	1200	900	2.10	4.74	5.05	1.07	4.70
5	1	1400	900	2.37	7.99	5.69	1.13	5.04
6	1	1400	900	2.41	6.32	5.80	1.31	4.41
7	1	1400	1000	2.42	5.97	6.46	1.15	5.60
8	1	1400	1300	2.39	7.33	8.28	1.30	6.37
9	1	1400	1300	2.40	6.98	8.31	1.14	7.28
10	1	1550	1300	2.53	11.25	8.78	1.38	6.37
11	1	1550	1300	2.53	11.25	8.78	1.21	7.28
12	2	1200	1200	3.26	7.22	10.43	1.55	6.72
13	2	1200	1200	3.26	7.22	10.43	1.33	7.84
14	2	1400	1350	3.73	9.00	13.42	1.52	8.82
15	2	1400	1300	3.77	7.98	13.07	1.54	8.49
16	2	1550	1300	4.14	8.68	14.36	1.83	7.84
17	2	1550	1300	4.22	6.94	14.63	1.72	8.49

Soil Cone Index, psi

Depth	0 ↔ 4 in	0 ↔ 6 in	0 ↔ 9 in
Before Tests	250	301	
After Tests	261	312	

Table A.11. CASE 2470 TEST RUNS AT MACNAB COMPANY RANCH WHEAT STUBBLE PLOT.

Run No.	Gear	Engine Speed RPM	Drawbar Pull lb	Speed mph	Wheel Slip %	Drawbar Horsepower	Fuel Consumption gal/hr	Fuel Consumption hp hr/gal
1	2 Lo	2200	6700	3.06	3.74	54.62	6.14	8.90
2	2 Lo	2000	6700	2.80	3.15	49.96	5.61	8.90
3	2 Lo	1800	6000	2.53	2.55	40.52	4.22	9.60
4	2 Int	2175	3200	4.14	1.37	35.35	5.92	5.97
5	2 Int	2150	6500	3.97	4.35	68.83	7.43	9.26
6	2 Int	2000	10000	3.58	7.32	95.44	9.01	10.59
7	2 Int	1800	7600	3.31	4.65	67.16	6.26	10.73
8	3 Lo	2150	8200	4.19	4.23	91.71	8.70	10.54
9	3 Lo	2000	8000	3.89	4.70	82.99	7.68	10.81
10	3 Lo	1800	7700	3.57	2.84	73.30	6.80	10.78
11	3 Lo	1800	7500	3.47	5.60	69.36	6.37	10.89
12	2 Hi	1800	8500	4.23	3.99	95.57	7.78	12.34
13	2 Hi	1600	7500	3.75	4.36	74.98	6.25	12.00

Soil Cone Index, psi

Depth	0 ↔ 4 in	0 ↔ 6 in	0 ↔ 9 in
Before Tests	246	391	
After Tests	357	485	