A drawbar dynamometer for direct reading of horsepower

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A DRAWBAR DYNAMOMETER FOR DIRECT READING OF HORSEPOWER

by

John G. Palmquist

A Thesis Submitted to the Graduate Faculty in Partial Fulfillment of The Requirements for the Degree of MASTER OF SCIENCE

Major Subject: Agricultural Engineering

Signatures have been redacted for privacy

Iowa State College
1949
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I. INTRODUCTION

Instruments which are used to measure power are called dynamometers. The word "dynamometer" is also used for those instruments which measure only the force exerted.

When a dynamometer is used to measure the output from a tractor drawbar the unit of power is a "horsepower" and the unit of draft is the pound.

The origin of the horsepower as a unit of power measurement is explained by the U. S. Bureau of Standards circular No. 34 (24, p. 9). A summary of material presented there is given in the following paragraph.

The term "horsepower" was introduced by Thomas Savery, the inventor of an early type of steam engine. James Watt, who is generally known as the inventor of the modern steam engine, adopted the term "horsepower" as a unit for expressing the power of his steam engines. The present day value of the horsepower was derived from experiments made in the year 1775 under the direction of Watt and his business partner Boulton. They procured some heavy draft horses from the brewery of Barclay and Perkins, London, and these animals were caused to lift a weight from the bottom of a deep well by pulling horizontally on a rope which passed over a pulley.

They found that a draft horse could conveniently raise
a weight of 100 pounds while walking at a rate of two and one-half miles per hour, or 220 feet per minute.

This is equivalent to 22,000 foot-pounds of work per minute. In order that a purchaser of one of his engines might have no ground for complaint, Watt added 50 per cent to this value, as an allowance for friction, and this established 33,000 foot-pounds per minute, or 550 foot-pounds per second, as the unit of power known as the English horsepower.

Several different types of tractor drawbar dynamometers are now in use.

In 1919 the State of Nebraska in the U.S.A. passed a law which became effective in 1921 and which required all tractor companies to submit one of each model tractor to be sold within the state to a performance test. The results of these tests have become widely publicized. Since this important step, interest in the testing of tractors has steadily increased.

The University of Saskatchewan conducts nearly four hundred tractor field days each summer. The tractors are tested while hitched to an implement and performing actual field work.

The interest now is not only in the tractor performance, but also in the efficiency of the implement itself to do a job.

The general approach to the problem of horsepower
measurement at tractor Field Days is one in which separate determinations are made of the speed in miles per hour and the drawbar pull in pounds. These two quantities are then used to compute the average horsepower developed during the test. This computation is usually done by means of a slide rule and after a complete set of data has been taken.

An approach such as this, to the problem of determining the horsepower delivered by the tractor drawbar to the implement, has two faults:

1. There is a considerable time delay between the taking of the data and its final presentation in terms of energy or horsepower.

2. The figures attained for the horsepower are the average of constantly fluctuating values. No provision has been made to determine the changes in horsepower that have occurred during the period of the test.

If a dynamometer could be built which would give instantaneous and continuous readings of horsepower, the above faults would be corrected.

The problem, therefore, consists in the creation of a machine which will take into account the three factors of force, distance, and time; and accurately convert these into an instantaneous reading of horsepower.
II. REVIEW OF LITERATURE

A. Early History

It is not known when man first sought to measure the power required to pull objects, but Wilson (25) in a review of the history of dynamometer development has reported the following outstanding events:

1. As early as 1803 scientific experiments were carried out to compare the rolling resistance of different types of wheeled carriages on roads.

2. In 1841 at Paris, France, General Morin conducted experiments with a dynamometer, which used the extension of a spring to measure the pull on a vehicle. (This type of apparatus was still used in almost all railway dynamosmeters in 1935, but it had not been found suitable for power measurement in agricultural work.)

3. With the advent of the agricultural tractor, work on dynamometers was seriously undertaken.

4. In 1919 at Lincoln, England, dynamometer trials were made of tractors. Two hydraulic machines were used: the Hyatt and the N.P.L.

Also used at this period were the Watson, Haines and Keen, Amsler, and Polikeit dynamometers.
B. Classification

Traction dynamometers are essentially instruments capable of measuring the force required to move an object. The Review of Literature in this thesis will cover traction dynamometers only, unless the machine is a combination of a traction and torsion dynamometer.

Traction dynamometers are divided into two classes, absorption dynamometers and transmission dynamometers. Absorption dynamometers will be defined as those which measure the load which they also supply to the tractor. In this respect they may be simply called loading machines. Transmission dynamometers will be defined as those which measure the load as they transmit it to some other machine.

C. Absorption Dynamometers

An absorption dynamometer was used at Iowa State College in 1923 to determine the draft and work output of horses.

A further development of the original machine led to the "Iowa Dynamometer Car for Horses" which was patented by E. V. Collins and J. B. Davidson in 1926. Some twenty of these machines were built at the college and the right to build others was granted to those educational institutions who asked for it.

It was from this "Dynamometer Car for Horses" that the
University of Saskatchewan developed an absorption dynamometer which was first used to test the strength of teams of horses, and later the horsepower of tractors.

The machine was made from the frame of a large motor truck, the rear wheels of which were used to drive a gear type water pump. This was the means used to develop a heavy traction load.

The machine was pulled by a telescoping tongue which was connected to the machine by a cable. The cable passed over a series of pulleys and finally was anchored to the top of a 2000 pound-concrete weight. This weight was held in vertical guides and was free to be lifted through a height of approximately three feet.

The gear pump had two delivery valves placed in series. The first valve was manually controlled and the other was connected by a linkage to the 2000-pound weight. The water from the pump was delivered to a reservoir which was also the source of supply.

It was possible to change the speed of the pump in relation to ground travel by shifting gears in the truck transmission. This allowed the operator to maintain the same load on the tractor regardless of the gear in which the tractor was pulling.

The machine usually was used to provide a 2000-pound traction load, but a greater load could be obtained by
connecting two more 1000-pound concrete weights to the initial one. These weights were also in guides and acted in a similar manner to the first weight.

When operating the machine, it was first necessary to place the truck transmission in the gear which was suited to the speed of the tractor. The next thing required was to close the manual delivery valve. As this was done, the force required to operate the pump was steadily increased until it took slightly over 2000-pounds pull to move the machine. At this point, the 2000-pound weight was lifted into its mid position about a foot above the floor.

If the rolling resistance of the unit changed and it took more than 2000 pounds to pull the machine, the tractor would do work on the concrete weight and lift it higher than its mid position. As the weight was lifted beyond its mid position, it operated a linkage that began to open the second by-pass valve. This action decreased the load on the pump, reduced the load on the tractor to the neighborhood of 2000 pounds, and stopped the weight from rising further. If the rolling resistance of the machine became less, the above procedure was carried out in the reverse order. First the weight would begin to fall and do work on the machine; and from this the other steps follow.

Davidson, Collins, and McKibben (6) describe an absorption dynamometer which was constructed at Iowa State College.
in 1935. This machine was capable of furnishing a constant drawbar load of any desired magnitude between 1000 and 5000 pounds.

The unit consisted essentially of a track-type tractor chassis, the tracks of which were connected by a suitable transmission to a hydraulic pump. The pump furnished a resistance which was in addition to the force required to overcome the rolling resistance of the apparatus.

The pulling force was applied to a movable piston, which was restrained from movement by the pressure of the fluid in the cylinder. A small pump, equipped with a pressure relief valve, was used to supply constant pressure to the cylinder. The pressure could be set at different values by opening or closing the pressure relief valve. This pressure was a measure of the drawbar load of the dynamometer.

When in operation, the pressure supplied to the cylinder by the pump is equal to the pressure caused by the drawbar load on the piston.

The dynamometer load was maintained constant in the following manner. Should the rolling resistance of the machine increase, the drawbar load increased and the fluid pressure exerted by the piston became greater than that of the pump. The resulting forward movement of the piston operated a linkage which opened the discharge valve of the main pump and decreased the load which the pump was supplying to the machine.
Barger (1) describes an absorption dynamometer which was built in 1937 at Kansas State College. This machine contained a hydraulic cylinder which was used to measure the load, and a gear pump which was used to supply the load. It was, therefore, a combination of a transmission dynamometer and a loading machine.

A Ford V-8 truck chassis and transmission was used to drive the gear type oil pump. The load was controlled by manual operation of the discharge valve of the pump.

This unit was pulled by the hydraulic cylinder and the oil pressure in this cylinder was fed to a direct reading gage in front of the operator.

The operator was able to read from the gage the drawbar pull in pounds, and from it adjust the discharge valve of the pump in order to maintain, or obtain, any desired drawbar pull.

The oil pressure from the pulling cylinder was also fed to a recording mechanism. A standard strip type chart was used, and it was driven by a fifth wheel. Its movement was therefore directly in proportion to distance traveled.

A needle, operated by a solenoid and in unison with a stop watch, was used to record the beginning and the end of a test. The speed of the machine was given by a modified car speedometer.

It was possible to calibrate or test this machine by
three methods:

1. Calibration on a regular Crosby gage tester.
2. Loading the dynamometer cylinder with known weights.
3. Placing the unit in series with a dynamometer of known calibration.

The load indicator mechanism made use of one of six different springs, depending upon the magnitude of the load to be measured. The calibration curves for these springs had to be uniform in slope throughout the range of loading if accurate results were to be obtained.

This absorption unit was able to provide a drawbar pull of 6000 pounds and could be used at speeds up to 20 miles per hour.

D. Transmission Dynamometers

1. The spring type

The spring type are those in which the load is transmitted either by compression or extension of a spring. An example of this is the Iowa Direct-Reading Traction Dynamometer.

Davidson (5) has the following general remarks:

The simplest type of spring dynamometer is a large spring scale in which the force acting is indicated on a dial as the spring is lengthened or compressed.
Because the resistance of a field tillage machine varies widely, the direct reading spring dynamometer is difficult to read; therefore, it is usually equipped with a mechanism for making a graphic record of the pull on a strip of paper or circular chart. Instruments of this kind are called recording dynamometers.

When such an instrument is used, the average force acting over a given distance or in a given length of time, may be obtained by averaging the forces recorded. This may be done by an area measuring device called a planimeter. The area thus obtained is divided by the length in order to determine the mean height. This height is proportional to the average pull in pounds.

To facilitate speedy determination of the average pull, the instruments may be self integrating.

Davidson has the following general description of the Iowa Recording and Integrating Dynamometer.

This machine was made by Central Scientific of Chicago, Illinois, and it is a recording and integrating machine. The draft was recorded on a strip chart, and the average draft over a distance of 50 feet was determined by a wheel and disc integrator. The final result was indicated by the amount of rotation accomplished by a small wheel which ran on the surface of a flat disc.

The flat disc was rotated proportional to ground travel.
The distance that the small wheel was placed from the center of the disc was proportional to drawbar pull.

A heavy load moved the wheel towards the outside rim of the rotating disc and under this condition a greater wheel rotation recorded greater average draft.

The integrator was operated by a string fed wheel and it automatically stopped after the wheel had made the number of revolutions corresponding to 50 feet of travel. The average draft for this distance was read from a dial on the side of the machine.

The string that turned the integrator was carried on a spool mounted on the dynamometer box. When the string left the spool, it passed twice around a rimmed wheel before it fed back to a stake in the ground, or to the hand of a person.

2. The hydraulic type

The hydraulic types are those which depend on hydraulic pressure to transmit the load.

Wilson (25) reports that the hydraulic type of transmission dynamometer consists of a container fitted with a plunger and filled with a suitable fluid. The plunger and container are coupled respectively to the drawbar of the prime mover and that of the implement. The pull is transmitted from the one to the other through the fluid and is
Fig. 1a. Iowa Direct-Reading Traction Dynamometer

Fig. 1b. Iowa Recording and Integrating Dynamometer
therefore converted into hydraulic pressure directly proportional to the magnitude of the load. This pressure is conveyed through a pipe line to any convenient point for measurement and recording.

The actual measurement and recording of drawbar pull is thus reduced to that of measuring and recording pressure. Hydraulic dynamometers differ mainly in the methods employed to measure this pressure.

Wilson (25) describes several makes of hydraulic dynamometers and his comments are outlined in the next few paragraphs.

The hydraulic pressure may be recorded by a Bourdon tube, but in order to be easily read, a dampening device is required in the oil line. A needle valve introduced into the pipe line can be used to control the flow of liquid and so damp out the excessive vibrations of the recording instrument. The zero point for Bourdon tube gages is always uncertain.

The hydraulic pressure may also be recorded by applying it to a small spring loaded plunger. Such a plunger must be provided with a lapped fit and be perfectly free in its cylinder. A piston stop is usually provided to prevent the spring from being overstressed. These springs, as supplied, are wound with initial tension. For work with low loads, it is often necessary, before calibration, to
remove some of this tension so that the dynamometer will commence to record at a lower pull. Different size loads may be measured by changing the strength of the spring.

Different ranges of pull are measured on some dynamometers by making use of multipliers to apply the load to the cylinder. By hitching in different holes, or by coupling the linkage together at different points, the drawbar pull can be greatly increased without altering the range of hydraulic pressure in the dynamometer.

Dynamometer records of drawbar pull may be evaluated by making use of a graduated transparent ruler. The mean displacement for every half inch length of record is estimated and the average for all these is the mean displacement.

The chief advantage of the hydraulic dynamometer is that momentary deflections in the pull are damped out. Since the maxima are decreased, and the minima are increased, no appreciable error is introduced.

Barger and Promersberger (2) describe a cart type hydraulic dynamometer which was designed and built at Kansas State College for the purpose of testing tillage machinery. This unit was self-contained and transmitted the load through a hydraulic cylinder which was inserted in the drawbar that extended through the machine.

The unit was both direct reading and recording; it weighed 850 pounds and required a minimum length of time
to be hooked up. The drawbar was adjustable in height and was supported on ball bearing wheels; vertical and side components were carried by the dynamometer frame. The weight of the unit was important when tests were made of large rolling machines on a down grade, since the machines tended to whip under these conditions.

The drawbar resistance of heavy machines could be measured by putting a stronger spring in the draft recording mechanism.

Giles (3) in describing a hydraulic drawbar dynamometer, constructed at the University of Missouri, reports the following details about the recording mechanism.

The recorder used was a 12 inch round chart type, manufactured by the Foxboro Company, Foxboro, Massachusetts. It was protected for overloading and was capable of reading a 3000 pound load at 350 pounds per square inch hydraulic pressure. To cover the complete range of loads expected, three different hydraulic cylinders and two different charts were required. Electro magnetic pens, operated by 1-1/2 volt flashlight batteries, were used to mark the time intervals.

The recorder was placed in a dust-free aluminum case, with a transparent front, and then mounted on two 15 inch lengths of rubber tubing. These were cut from a 30x3 tube and vulcanized at the ends. The air pressure in these
tubes was low; but that within the bottom tube had to be slightly greater. The recorder was tilted back 10° in order to insure pen contact with the graph.

This type of mounting was entirely satisfactory in controlling vibration of the recording instrument. The total weight of the recorder and frame was 22 pounds.

To evaluate the results, a planimeter was not used; instead, readings were taken every ten seconds and averaged.

The oil pressure transmission line leading to the recorder was 1/8 inch, flexible, all metal, tubing made by the Titeflex Hose Company, Newark, New Jersey.

Oleson (19) in 1928 constructed at Iowa State College a dynamometer for measuring the tractive effort in an automobile. The dynamometer was built in a car and was operated by linkage connected to the rear axle housing.

The recording device made use of two sylphon bellows and was capable of measuring both the power output from the wheels during normal driving and the power input to the wheels when the car was coasting freely down a hill. Oleson describes the sylphon bellows as a seamless, jointless, metallic structure; capable of large linear extension, and having power to resist a large amount of hydrostatic pressure. The folds of this unit should possess a high degree of resilience.

The University of Saskatchewan has used a hydraulic
dynamometer for several years in connection with their field days.

This unit consisted of a very short hydraulic cylinder which was connected by a length of pressure hose to a bourdon gage. The piston force was transmitted to the oil through a rubber diaphragm. This arrangement prevented any oil leakage and eliminated the need for a lapped fit between the cylinder and piston. The cylinder was axially loaded at each end and no outside linkages were required to transmit the load.

The upper part of the piston contained a heavy shaft which projected through the wall of the cylinder. This shaft was threaded and fitted with a burr. When the cylinder was placed under excess loads it was the function of this burr to contact the cylinder wall and prevent all the load from passing through the diaphragm into the oil. This arrangement prevented the piston from either damaging the diaphragm or spoiling the bourdon gage.

The Bourdon gage and the cylinder were proportioned so that the reading of the gage was ten to one. A load of 3000 pounds on the cylinder was 300 p.s.i. of hydraulic pressure on the gage. The gage was capable of reading 5000 pounds pull but the cylinder was designed to carry only 3000 pounds.

The cylinder weighed about 50 pounds and was easily
placed between the drawbar of the tractor and the implement. Since the cylinder was short it did not seriously lengthen the tractor drawbar.

The bourdon gage was held by a man standing on the tractor. A needle valve was placed in the hydraulic line beside the gage. This valve was always kept closed when the tractor started. This was done in order to prevent the impact load from damaging the gage. During steady operation, the valve was opened and then slowly closed until the load vibrations were damped out.

It was possible to easily read the gage although the load might have been alternating 500 pounds or more in magnitude.

These dynamometer cylinders have been used by the University of Saskatchewan for several years at nearly four hundred field days each year. They have proved highly satisfactory in their operation.

In 1937, Hendrix (12) at the University of Tennessee developed a hydraulic dynamometer, the power transmitting unit of which consisted of a semi-flexible steel bellows made by welding steel plates together. This unit, completely sealed except for a tube connection to the recording unit, was filled with liquid and no piston or packing was required. A liquid reservoir was used to increase or decrease the fluid in the bellows, so the gage would read zero at no load.
This bellows was capable of withstanding the greatest tractor drawbar load. Its rupture point when placed under a static load was approximately 15,000 pounds. It had a movement of 1/16 of an inch at 2000 pounds load.

A dynamometer of this type is small in size, light in weight, simple and sturdy. Its cost is low and it permits short coupling of the tractor and implement. It is suitable for either direct reading or recording. The calibration curve is a straight line.

3. The electric type

This type of transmission dynamometer depends on electricity to measure the load while actual transmission of it may be one of several different methods.

In 1940, Giles (9) at the University of North Carolina, constructed an electric dynamometer which made use of the "Electric-ultra Micrometer Circuit".

He describes its operation as follows: The drawbar pull was transmitted through two flat springs upon which are mounted two condenser plates. The deflection of the springs with load altered the distance between the two condenser plates. The electric current delivered by the electric ultra micrometer circuit was proportional to the distance between the condenser plates and therefore was a measure of the draft in pounds.
This dynamometer was reasonable in cost, simple to make, and permitted close coupling of the tractor and implement. It proved unsatisfactory for the following reasons:

1. The micrometer circuit required batteries and radio tubes which had to be kept in good condition.

2. Time was required to warm up the circuit before it would operate.

3. The calibration was not reliable since the condenser plates were apt to become bent; and because the dielectric constant of dusty air between the condenser plates was not the same as that for clean air.

4. The meter was delicate and could not be easily read since there was no convenient method of dampening the meter hand in the field.

For these reasons this dynamometer was not as dependable as other types. The builder states that it was a good idea, but it didn't work.

In 1938, Garver and Brooks (7), at the University of California Experiment Station, constructed an electrical dynamometer capable of reading the horsepower directly in amperes.
This dynamometer was made so that the total current flowing in a circuit was directly proportional to both the speed and the load.

The current for this circuit was supplied by a separately excited non-compound generator which delivered a voltage directly proportional to its rotational speed. Since this generator was driven from the front wheel of the tractor, the voltage applied to the circuit was equal to some constant times the speed of the tractor in miles per hour.

The resistance of this circuit was varied by a rheostat which was operated by the spring which transmitted the tractor drawbar pull. This rheostat was so constructed that the resistance decreased directly with the extension of the spring and therefore the current decreased directly as the draft in pounds.

In order to keep the resistance of the circuit independent of the wire temperature, cupron, an alloy of nickel and copper, was used. This alloy had a constant resistance regardless of its temperature.

The current flowing in a circuit is given by Ohms law as equal to the voltage applied across the circuit divided by the total resistance of the circuit.

\[ I = \frac{E}{R} \]
Since it has already been stated for this machine that

\[ E = K_1 \times \text{speed in m.p.h.} \]

and

\[ R = K_2 \times \frac{1}{\text{draft}} \]

it follows that

\[ I = K_3 \times \text{speed} \times \text{draft} \]

\[ = K_4 \times \text{horsepower}. \]

The current measured by the ammeter is equal to some constant \( K_4 \) times the horsepower. When properly calibrated, the ammeter read horsepower directly.

The total work done in horsepower hours could be obtained for a test by the use of an amp-hour meter, as this meter will integrate the variations in current.

This dynamometer proved undesirable as a dynamometer for testing tractors, since an excessive amount of time is required to mount the generator on the front wheel of the tractor.

4. Special types

Wilson (25) describes the Polak dynamometer. This unit consisted of a copper cylinder 10 centimeters in diameter, fitted with a piston having a 10 centimeter stroke. The cylinder was filled with water and had a small hole at its closed end. The pull to be measured was applied to the liquid in the cylinder and pressure was set up proportional to the pull. This pressure forced the water through
the small orifice and out of the cylinder. The amount of water expelled in a given time was a measure of the average pull exerted during that time.

This type of instrument is of little value in testing tractors, since it gives only average values and these not directly.

June Roberts (22) explains the use of a nomographic chart for the Iowa dynamometer. On this chart the values of draft in pounds, speed in miles per hour and horsepower developed were placed on three separate lines, which were spaced about two inches apart.

When the draft in pounds and the speed in miles per hour for any test are known, it was possible by using a straight edge to read the horsepower directly from the chart.

Schwab (23) designed and built a dynamometer cart in 1947 at Iowa State College which was capable of measuring a 20,000 pound draft load with an instrument which was capable of recording only 2000 pounds pull.

Heth (13) points out that when strain gages are used as tractor drawbar dynamometers, it is difficult to eliminate the effects of drawbar weight and side draft from the results.

McCall (17) constructed a transmission dynamometer at Ohio State University. This machine was a cart type,
hydraulic dynamometer; it weighed 1000 pounds and was capable of measuring 8000 pounds of draft. It was equipped with a Gulley recording unit for both power take-off and drawbar loads. A direct reading draft gage was used to indicate the drawbar pull.

The power take-off part of the dynamometer consisted of a set of three sprockets which were mounted on a frame. These three sprockets were connected to one another by a single link chain. A further sprocket was added as a tightener.

The sprocket carrying frame was pivoted about the driven sprocket; whenever a load was transmitted through this mechanism a torque was placed on the frame directly proportional to the load. This torque was measured and from it and the r.p.m., the horsepower was calculated.

E. Summary

In any dynamometer design, a detailed study should be made in order to fix clearly in mind the conditions that are likely to be encountered. It is then possible to build up the counter measures that must be taken in order for the unit to function satisfactorily.

The review of literature reveals the following general requirements for a successful drawbar dynamometer:
1. A reasonable cost so that production of the unit is feasible.

2. It must be reliable and give accurate data upon which constructive thinking may be built.

3. It must be adaptable to, and function accurately, under all farm conditions.

4. Its construction must permit easy and rapid attachment to any combination of tractor and implement.

5. A sturdy and strong construction is necessary so the unit will stand up under punishment.

6. Highly complicated devices are of no use unless they are rugged and sturdy and will work satisfactorily under all conditions which they may encounter.

7. In the construction of any measuring unit extreme care should not be put on one point, to make it very accurate, if another factor cannot be made equally as accurate.

8. The accuracy of a machine should not exceed the purpose for which it is designed.

9. A mechanism may be accurate but if it is awkward to use, or if it does not permit
continuous operation, it will be undesirable as a dynamometer.

10. Instantaneous direct readings, recordings or integrating devices, are desirable for both draft and horsepower measurement.

11. As far as possible, the dynamometer should not alter the hitching arrangement since this will change the draft characteristics of the machine.

12. A flexible hitch will make it impossible to test a machine which must be rigidly supported. Examples of this are: a cultivator or a rubber-tired vehicle which is traveling on a down grade. The former cannot be properly adjusted, and the latter will tend to whip.

13. As far as it is convenient, the unit should be compact and easy to take care of.

14. Every machine that is made should be checked carefully to eliminate any possible chance of injury to those who are operating it.

The review of literature also reveals that to date there is no inexpensive way of measuring drawbar power. No machine has been constructed which will operate entirely satisfactorily to give direct instantaneous readings of
both drawbar pull and horsepower developed.

There is, therefore, a need for the design and construction of a tractor drawbar dynamometer, which will fulfill all the main requirements for a machine of this type, and which will operate in a satisfactory manner to give direct and recorded readings for both horsepower and draft.

This problem is considered to be important from the standpoint of pure science, and also for the general instruction and welfare of the agricultural community.
III. INVESTIGATION

A. Initial Investigation

1. Objectives

The main object of the investigation was to develop a good means of measuring the instantaneous horsepower that is delivered by the drawbar of a tractor.

2. Assumptions

It was assumed that an entirely new approach to the problem would be the one most likely to succeed. No special effort was made to improve upon the ideas of others, as it was generally concluded by them that their ideas were not the answer to the problem.

A formula for the horsepower delivered by the drawbar of a tractor is:

\[
H. P. = \frac{SD}{375} \tag{1.1}
\]

where \( H. P. \) is horsepower, \( S \) is the speed in miles per hour, and \( D \) is the draft in pounds.

Since horsepower is a function of the speed of the tractor multiplied by the drawbar pull, the initial step
was assumed to be the discovery of a formula which contained, within itself, multiplication of at least two of its variables. This formula must also be one in which accurate control of two of the constituents was considered to be, not only possible, but readily adaptable to the speed and draft of a tractor. Finally, if the formula was to be capable of multiplying an infinite number of combinations of two variables, it must have these additional features: either there must be only two significant independent variables in the formula, or the remaining significant independent variables must always interact so as to produce a figure which is constant, or nearly constant, in value.

If this is so, the dependent variable will accurately indicate the multiplication of the two independent variables with which we are concerned. When such a formula has been discovered, the next step will be to find a method whereby the two multiplying factors may be varied: the one directly as the tractor speed, and the other directly as the drawbar pull. This relationship is necessary because only constants may be introduced into the formula. All other factors, except the variables representing speed, draft, and horsepower, must be constants or interact so as to produce a constant.
3. Method of procedure

In the initial attempt, the formula which was chosen as a possible means of multiplication was that which applies to an alternating current circuit containing inductance and resistance in series. The formula for such a circuit is given by Gray (10, p. 302) as

\[ E = IZ \]  

(1.2)

in which \( E \) is the voltage, \( I \) is the current in amperes, and \( Z \) is the impedance of the circuit in ohms.

Solving equation (1.2) for \( I \), we obtain

\[ I = \frac{E}{Z} \]  

(1.3)

The impedance for a circuit containing inductance and resistance in series is expressed by Gray (10, p. 311) in the following formula:

\[ Z = \sqrt{R^2 + X^2} \]  

(1.4)

where \( Z \) is the impedance in ohms, \( R \) is the resistance of the circuit in ohms, and \( X \) is the inductive reactance in ohms.

The inductive reactance \( X \) is the name given to a group of factors which occur so frequently in alternating current that they have been given this special name.

These factors are given by Gray (10, p. 305) in the formula,

\[ X = 2\pi f L \]  

(1.5)
where X is the inductive reactance in ohms, f is the frequency in cycles per second and L is the inductance in henrys.

The inductance L is as much a constant of a circuit as is its electrical resistance.

If we choose a circuit in which the resistance is very small, the value of R in equation (1.4) may be neglected and this equation becomes

\[ z = \sqrt{X^2} = X \]

Substituting this in equation (1.5) we can write

\[ z = 2\pi f L \tag{1.6} \]

Putting this value of Z in equation (1.3) yields

\[ I = \frac{E}{2\pi f L} \tag{1.7} \]

which is the approximate equation for an alternating current circuit containing in series an inductance and a very low resistance.

If there is some way to deliver alternating current at a constant voltage regardless of frequency, the equation (1.7) may be written

\[ I = \frac{K_1}{f L} \tag{1.8} \]

where \( K_1 \) is a constant equal to \( \frac{E}{2\pi f} \).

The attempted application of this formula to the horsepower measurement of a tractor was as follows:
It was decided to construct an A.C. generator and to run it at a speed directly proportional to the speed of the tractor. This arrangement would give a frequency which was directly proportional to the speed in miles per hour of the tractor.

The only compensation necessary under this arrangement was some means to keep the voltage constant, regardless of generator speed.

The second aspect of this approach was the construction of an A.C. circuit which had very high inductance but low resistance. Some means would have to be devised whereby the inductance \( L \) could be varied directly as the drawbar pull of a tractor.

If the above conditions are fulfilled the following equations can be written.

\[
L = K_2 D \quad (1.9)
\]

where \( K_2 \) is a constant,

\[
f = K_3 S \quad (1.10)
\]

where \( K_3 \) is a constant.

Substituting equations \((1.9)\) and \((1.10)\) in equation \((1.8)\), we obtain

\[
I = \frac{K_1}{K_3 K_2 D}
\]

If we let \( \frac{K_1}{K_3 K_2} \) be equal to \( K_4 \) we have

\[
I = \frac{K_4}{S D}
\]
Inverting the equation we get

\[ \frac{1}{I} = \frac{SD}{K_4} \]  

(1.11)

If we compare this equation with equation (1.1), the equation for horsepower, \( H.P. = \frac{SD}{375} \), it is seen that the horsepower will be measured as the inverse of the current in amperes.

If we assume \( K_4 \) must be multiplied by \( K_5 \) in order to make \( K_4K_5 \) equal to 375, the equations may be written

\[ H.P. = \frac{SD}{375} = \frac{SD}{K_4K_5} = \frac{1}{IK_5} \]

The equation for horsepower becomes

\[ H.P. = \frac{1}{K_5I} \]  

(1.12)

4. Apparatus

In an alternating current circuit, the losses which are due to hysteresis and eddy currents will be proportional to the frequency.

In order to reduce these losses to a minimum, a generator which would be able to produce a fairly high voltage at low r.p.m. was desirable.

A 12 volt, Dodge model G.I. starter generator was secured and this was rewound into an A.C. generator with
the help of Dr. Pinney of the Physics Department at Iowa State College.

An inductive circuit containing low resistance was constructed. The inductance was varied by changing the length of two air gaps in the magnetic circuit.

The generator was connected to this circuit and five tests were run. The voltage of the generator was kept constant at six volts over a range of speed from 400 to 2000 r.p.m. for all these tests.

In the first three tests readings of amperage versus r.p.m. were taken for fixed air gaps of 1/3, 1/2, and 1 inch.

In the last two tests, readings of amperage versus a changing air gap, from 0 to 4 inches, were taken for a fixed generator speed of 1000 and 1600 r.p.m.

5. Results

The curves of this data were plotted. An empirical formula was set up for the current flowing at 708 r.p.m. and a 1/3-inch air gap. This formula was now used to calculate the expected current at 2000 r.p.m. The discrepancy between the actual and the calculated current revealed an error of 19.5 per cent.

Since the allowable error for a tractor dynamometer should not exceed 5 per cent, the existing equipment was
inadequate. This error was due primarily to hysteresis and eddy current loss; and further calculations were made to see if the method would function if a high grade magnetic alloy Alnico were used. These calculations revealed that the errors due to losses would be less than 5 per cent for only a narrow range of frequency. For this reason, another model was not made.

The discovery was made that the generator constructed was acting as a four pole machine instead of a two pole as planned. This discovery was made when a test of the generator was made on an oscilloscope. The possibility of cutting the losses in half was now present, but such a change would still have left the error due to losses at almost 10 per cent. Dr. Pinney of the Physics department at Iowa State College tested the generator. He said that the wave form was perfect. It seemed apparent that there was nothing further which could be done to the original unit to improve it and so it was abandoned as a working solution to the problem.

6. Conclusions

The use of an alternating current circuit containing inductance is not a feasible means of measuring tractor drawbar horsepower, unless the losses due to hysteresis and eddy currents can be reduced to a point where the
desired accuracy can now be obtained.

A search for high-performance magnetic alloys did not reveal any that were much superior to Alnico.

The conclusion was made that as far as present information revealed, there was no apparent solution to the problem of horsepower measurement by the alternating current, inductive circuit method.

B. Final Investigation

1. Method of procedure

a. The theory of the pivoted beam. The final approach was concerned with the transmitting of a force in a mechanical manner. The force transmitted was directly proportional to speed and the manner in which it was transmitted was changed by the draft. The mechanical linkage was constructed so the manner in which force was transmitted was directly proportional to draft; and the force transmitted was directly proportional to the manner of transmittance.

The mechanical linkage in essence consisted of a pivoted beam to which forces were applied. This beam was kept in equilibrium under the action of these forces. In such a case the clockwise moment must be equal to the counter-clockwise moment.
It was thought that it might be possible to apply the clockwise moment from the initial values of speed and draft and to read their multiplication as the counterclockwise moment was measured.

The first drawing in Figure 3a will be used to explain the theory whereby a pivoted beam can be used to multiply. In the figure the beam is represented by a horizontal line which is pivoted at the point "A". The clockwise moment is applied by the force "F₁" which is at a distance "X₁" from the pivot point. The counterclockwise moment is applied by a force "F₀" at a distance "X₀" from the pivot point.

The following formulas will now be developed.

Since for a pivoted beam in equilibrium the clockwise moment is equal to the counterclockwise moment, the following equation may be written:

\[ F X₁ = F₀ X₀ \]  \hspace{1cm} (2.1)

Solving the equation for \( F \), we get

\[ F = \frac{F₀ X₀}{X₁} \]  \hspace{1cm} (2.2)

Now the formula for drawbar horsepower by equation (1.1) is,

\[ H.P. = \frac{S D}{375} \]

The similarity of this equation to equation (2.2) is
The names of the different parts of the instantaneous multiplier, represented by numbers in Figure 3a, are given as follows:

1. The draft-pressure cylinder
2. The spring-loaded piston
3. The pivoted beam
4. The pivot support
5. The pivot bearings
6. The pivot shaft
7. The self-aligning coupling
8. The speed-pressure cylinder
9. The loading piston
10. The rolling bearing
11. The restraining bearing
12. The horsepower cylinder
13. The restraining piston
14. The recording gage
15. The sliding plate
16. The "I" beam
easily seen. Referring to the second part of Figure 3a it is seen that the pivoted beam will be both hydraulically loaded and restrained. If we let the piston which supplies the force $F_0$ have an area $A_0$, and if the pressure supplied to this cylinder be denoted by $P_0$, the following equation is true:

$$F_0 = P_0 A_0 \quad (2.3)$$

where $F_0$ is the force in pounds, $P_0$ the pressure in p.s.i., and $A_0$ the area is square inches. If we let the piston which takes up the force $F$ (the restraining piston) have an area $A_1$, and if the pressure above this piston is denoted by $P_1$, the equation for $F$ becomes

$$F = PA_1 \quad (2.4)$$

where $F$ is the force in pounds $P$ is the pressure in p.s.i., and $A_1$ is the area in square inches.

Substituting equations (2.3) and (2.4) in equation (2.2) we get

$$PA_1 = \frac{F_0 A_0 X_0}{X_1}$$

Rearranging the terms we obtain

$$P = \frac{A_0 F_0 X_0}{A_1 X_1} \quad (2.5)$$

Hydraulic pressure in this thesis will be measured in pounds per square inch and will be written as p.s.i.
If the distance $x_1$, from the center of the restraining piston to the pivot point, is fixed, the value of $x_1$ becomes a constant.

Since the values of $A_0$ and $A$ are also constants, the value of $\frac{A_1x_1}{A_0}$ may be denoted by a constant $K_6$. The equation may be written

$$K_6 = \frac{A_1x_1}{A_0} \tag{2.6}$$

Using equation (2.6), equation (2.5) becomes

$$P = \frac{P_0x_0}{K_6} \tag{2.7}$$

If the pressure $P_0$ in the loading piston is made directly proportional to the tractor speed $S$, and if the distance $x_0$ is made directly proportional to the tractor draft $D$, the following equations may be written:

$$P_0 = K_7S \tag{2.8}$$
$$x_0 = K_8D \tag{2.9}$$

where $K_7$ and $K_8$ are constants.

Substituting equations (2.8) and (2.9) in equation (2.7) yields

$$P = \frac{K_7SK_8D}{K_6}$$

Rearranging the terms the equation becomes

$$P = \frac{K_7SK_8SD}{K_6} \tag{2.10}$$
The horsepower formula as given in equation (1.1) is

\[ H.P. = \frac{SD}{375} \]

From the similarity of the formulas it is seen that we will want to measure the horsepower in terms of \( P \), the pressure in pounds per square inch in the Horsepower Cylinder.

At this point it is necessary to establish what quantity of horsepower we wish to measure. The maximum drawbar horsepower is not expected to exceed forty.

The maximum drawbar pull for this problem was taken as 3000 pounds and the maximum speed as six miles per hour.

Let \( K_g \) be equal to the amount of horsepower represented by one pound per square inch (p.s.i.) of hydraulic pressure in the restraining piston. This may be expressed as

\[ H.P. = K_g P \quad (2.11) \]

From equations (1.1), (2.10) and (2.11) we can write

\[ H.P. = \frac{SD}{375} = K_g P = \frac{K_7 K_g K_9 SD}{K_6} \quad (2.12) \]

From this we can make the equation,

\[ \frac{SD}{375} = \frac{K_7 K_g K_9 SD}{K_6} \quad (2.13) \]

Solving this equation we have,

\[ \frac{K_7 K_g K_9}{K_6} = \frac{1}{375} \quad (2.14) \]
where $K_7$ is given by equation (2.8), $K_8$ by (2.9), $K_9$ by (2.11), and $K_9$ by equation (2.6).

If we substitute the value of $K_9$ into equation (2.6), we obtain

$$\frac{A_0 K_7 K_8 K_9}{A_1 X_1} = \frac{1}{375} \quad (2.15)$$

We will call the term $\frac{A_0 K_7 K_8}{A_1 X_1}$ the "Machine Constant", and $K_9$ we will call the "Gage Constant". It is by fixing these two constants that we will bring about a simple and direct reading of direct horsepower.

If $K_9$ in equation (2.11) is equal to unity, the form of equation (2.15) will be

$$\frac{A_0 K_7 K_8}{A_1 X_1} = \frac{1}{375}$$

and the horsepower will be read as one horsepower for every p.s.i. of gage pressure.

If the value of $\frac{A_0 K_7 K_8}{A_1 X_1}$ is not equal to $\frac{1}{375}$, it may be possible to make $\frac{A_0 K_7 K_9 K_9}{A_1 X_1}$ equal to $\frac{1}{375}$ by arbitrarily choosing $K_9$ so as to produce this result.

It appears highly probable that some combination of values for $K_7$, $K_8$, $K_9$, $A_0$, $A_1$, and $X_1$ can be found which will produce a machine which will read horsepower with each horsepower represented by some convenient even number of p.s.i. pressure. All readings of horsepower with this pivoted beam method will be direct and there will be no
compaction of readings at any part of the dial. If each horsepower is represented by an even number of p.s.i., pressure, no calibration of the dial will be needed; but a standard dial face graduated in p.s.i. will be used.

b. Distance proportional to draft. The first condition required is that the distance \( X_0 \) be directly proportional to the draft of the tractor, equation (2.9). This condition can be fulfilled in the following manner. The drawbar pull if transmitted through a hydraulic cylinder will give a hydraulic pressure which is directly proportional. If this pressure is fed to a spring loaded piston, a distance will then be obtained which is directly proportional to drawbar pull.

c. Force proportional to speed. The second condition required by equation (2.9) is that a hydraulic pressure must be obtained which is directly proportional to the speed of the tractor in miles per hour.

It is known that an r.p.m. can be obtained which is directly proportional to the rate of ground travel, and that this r.p.m. is transmitted into force in the speedometer of a car. Since it appeared that this problem would require a much greater force than that found in speedometers, the help of an electrical engineer was sought.

Dr. Boast, in the electrical engineering department
of Iowa State College, suggested that the best means of solving this problem would be by the use of a separately excited D.C. generator. He indicated that as long as the load on the generator was kept constant, the torque on the generator housing would usually be directly proportional to the r.p.m.

A case where this would not be true would be one in which the armature resistance of the generator was considerable. In this case, the I.R. drop in the armature circuit could not be neglected; the terminal voltage would not be directly proportional to the r.p.m. of the generator, and this would spoil the direct relationship between the housing torque and the r.p.m. Usually the armature resistance is so small it can be neglected. The only other factor that might cause serious trouble would be that arising out of armature reaction. He indicated that a 32 volt motor or generator would probably be the unit best suited to the problem. It was pointed out that armature reaction could be kept at a minimum if the armature currents used were kept as small as possible. Dr. Boast also indicated that the housing torques would decrease in value, as low armature currents were used; but a balance could probably be struck which would be a solution to both problems.

Armature reaction can be reduced by the use of a
generator which contains interpools, but a solution to the problem was attempted with a generator which did not contain these.

A review of the theory connected with separately-excited, direct-current generators was made and a brief presentation of this will now be undertaken.

Marks (16, p. 1997) gives the following formulae for an electrical generator: the induced e.m.f. *E* is given as,

\[ E = K_{10} \phi N \]  

(3.1)

where \( K_{10} \) is a constant; \( \phi \) the flux entering the armature from one north pole; \( N \) the speed in r.p.m.

The terminal voltage \( V \) is given by the equation,

\[ V = E - I_a R_a \]  

(3.2)

where \( I_a \) is the armature current and \( R_a \) is the armature resistance including the brush and contact resistance.

The electrical torque developed by the armature is given by Marks (16, p. 2001) as,

\[ T = K_{11} \phi I_a \]  

(3.3)

where \( K_{11} \) is a constant.

The electrical torque placed upon the generator housing will be approximately equal to that required by the armature, since it is the electrical interference of this housing that is responsible for the electrical load on the armature.

*E.m.f. means electromotive force*
The mechanical forces required to overcome bearing friction, or to accelerate the armature, are not transferred over to the housing.

The mechanical friction drag of the brushes and some wind resistance will add to the total torque placed on the generator housing. These mechanical forces will be considered at a later point in the problem. All references to the torque are for the present considered to be that which is electrically applied.

The formula for the housing torque may be written the same as that given for armature torque, since the two are approximately equal.

Since it is the torque on the housing that is important to this problem, future references to torque will be taken to mean the electrical torque on the generator housing.

Since the housing torque is approximately equal to the electrical armature torque, it may be expressed by the same formula; namely,

\[ T = K_{11} \phi I_a \]

The current flowing in the generator armature is the same as the current in the total circuit composed of the armature plus an outside resistance. This can be expressed as,

\[ I_a = I_e \]  \hspace{1cm} (3.4)
where \( I_a \) is the current in the generator armature, and \( I_e \) is the current in the external resistance.

Ohms law for the current flowing in a resistance is

\[
I = \frac{E}{R}
\]

(3.5)

where \( E \) is the voltage, \( R \) the resistance in ohms. The current flowing in the external resistance is, therefore,

\[
I_e = \frac{V}{R_e}
\]

(3.6)

where \( R \) is the resistance in the external circuit.

Substituting equations (3.2) and (3.4) in equation (3.6) we have

\[
I_a = \frac{E - I_a R_a}{R_e}
\]

(3.7)

Substituting equation (3.1) in (3.7) yields

\[
I_a = \frac{K_{10} \phi N - I_a R_a}{R_e} = \frac{K_{10} \phi N - I_a R_a}{R_e} = \frac{K_{10} \phi N - I_a R_e}{R_e}
\]

Since the armature resistance, \( R_a \), is very small and \( R_e \) is large in comparison, the value of the term \( \frac{I_a R_e}{R_e} \) can be neglected and the equation for \( I_a \) may be written

\[
I_a = \frac{K_{10} \phi N}{R_e}
\]

(3.8)

Substituting equation (3.8) in (3.3), we obtain

\[
T = \frac{K_{11} \phi K_{10} \phi N}{R_e}
\]

which becomes

\[
T = \frac{K_{10} K_{11} \phi N}{R_e}
\]

(3.9)
If φ and $R_e$ remain constant the value of \( \frac{K_{10} K_{11} \phi^2}{R_e} \) can be represented by a constant $K_{12}$ and the formula is reduced to

\[
T = K_{12} \phi
\]

(3.10)

From this formula it is seen that the housing torque is directly proportional to the r.p.m. of the generator.

In order to keep the external resistance, $R_e$, at a constant value, it is necessary to use a type of wire which has a constant temperature coefficient. The resistance of such a coil of wire will remain appreciably the same regardless of the current flowing in the wire.

For this purpose, constantan wire was thought to be suitable. In order to keep the flux $\phi$ constant it was necessary to use a separately excited generator. When such a generator is used, the only factor which may seriously alter the flux is armature reaction. This is caused by the current which is flowing in the armature. Whenever a current flows in a wire, it sets up a field about the wire itself. The armature currents set up a field which lies across the main flow of flux through the armature.

This secondary field is opposed to the main flux at one side of the pole and assists the main flow at the other side.

The effect of the secondary flux is to deflect the
main flux so that instead of going directly across the armature it is concentrated at one side of each pole. The flux then proceeds across the armature at an angle to meet the flux concentration entering the opposite pole. When the flux is made to flow in this manner, the amount of flow is reduced. It is this reduction in flux that is the result of armature reaction, and the amount of reduction will be proportional to the armature current.

It is possible to construct a generator in which the armature reaction is compensated for, but this is not true of the generator used in this problem. The only way left, in which to avoid excess interference by armature reaction, is to keep the armature currents below a certain value.

This was done by inserting a fairly high resistance in the external circuit. The actual determination of this resistance was done by experiment. If the torque on the generator fails to keep a constant relationship with speed, while the generator is still within the r.p.m. required by a tractor traveling at six miles per hour, either the external resistance must be increased or another generator which is not affected by armature reaction must be secured.

If these measures are successful, equation (3.9) will be true and the torque will be directly proportional to the r.p.m. of the generator.
Once this condition has been established it will only be necessary to convert this torque into hydraulic pressure. This problem can be solved by transferring the pull from the generator housing to a hydraulic cylinder and piston. This will give us hydraulic pressure directly proportional to r.p.m. and the conditions of equation (2.8) will be fulfilled. Then no more difficulties of great importance will appear to be left in the way of direct horsepower measurement.

2. Apparatus

   a. The generator. The generator was constructed from a 32 volt D.C. motor.

      The following information was listed on the nameplate:

      D.C. Motor, Model No. 29873, Type SD, Frame 1346,
      Form D, Volts 32, Amps. 9, H.P. 1/4, Speed 1725,
      comp. wound, Temp rise 40° open, Time rating —,
      No. 20920.

      Since the motor was compound wound, it was first necessary to remove the compound winding. There is usually a six volt supply of D.C. current on a modern tractor, so the decision was made to re-wind the field coils in a way which would permit their excitation by a six volt source instead of 32.

      This was done by splitting the field coil wire into
a number of equal paths and a parallel circuit was formed in place of a series circuit.

Since the value of 6 volts is approximately 13.7 per cent of 32 volts, the new resistance of one of the equal length paths should be approximately 13.7 per cent of the resistance of the field coil wire if the same amount of field current is to be kept flowing in the wire.

The amount of heat produced in a conductor by an electric current is given by Hausmann (11, p. 370) as

\[ W = RI^2t \]  \hspace{1cm} (4.1)

where \( W \) is the energy in joules ( liberated in the form of heat), \( R \) is the resistance in ohms, \( I \) is amperes, and \( t \) is the time in seconds.

From this equation it is seen that the heat liberated varies as the square of the current. It was for this reason that the decision was made to keep the field current close to its former value and prevent a cooling problem.

From the standpoint of this research it is desirable to have a generator which has a high torque, and since the torque is proportional to the square of the flux (equation 3.9), an attempt was made to increase the flux set up by the field coils.

---

One joule is equal to 0.239 calories.
The flux produced by a coil of wire is given by Gray (10, p.68) and may be written as

\[
\phi = \frac{4\pi nI \mu A}{10lc}
\]  

(4.2)

where \( \phi \) is given in maxwells, or lines, \( n \) is the number of turns of wire, \( I \) the current, \( \mu \) the permeability of the metal, \( A \) the cross-sectional area in square centimeters, and \( lc \) the length of the magnetic path in centimeters.

The only factors in the above formula which are affected by the field coils are the amp-turns "nI".

Since we have already decided not to increase the field current, the only way left to increase the flux is by increasing the number of turns of wire in the field coils.

The amount by which we can increase the length, or weight, of the field coil wire in terms of the original length of wire, can be computed as follows: Let the initial resistance of the field coil be "x" ohms and to this we will add "y" ohms to produce a total series resistance of "x + y" ohms.

It has been stated that the resistance of each path in the new parallel circuit should be 18.7 per cent of the initial field coil resistance "x". We will have to split the total length of wire into an even number of coils,
such as 2, 4, 6, etc., so an equal number may be placed on each of the two poles of the generator. If we split the total length into six equal parts, the best approximation is made towards producing a coil which has a resistance of 18.7 per cent of the initial resistance. This will be true because the additional resistance "y" will be small in terms of the initial resistance "x".

Having established six as the number of parts into which we wish to split the total length of wire, we can express the resistance of each part as \( \frac{x + y}{6} \). Since this value is expected to be 18.7 per cent of "x", the initial resistance, we may write the equation,

\[
\frac{x + y}{6} = \frac{18.7x}{100}
\]

Solving this equation we obtain

\[ y = 0.12x \]

and we conclude that the field current will remain the same if the initial coil is increased in length by 12 per cent of its original value.

The 12 per cent figure was not used, however, except as a general guide. One hundred and fifty feet (approximately) of additional magnet wire was bought and added to the initial field coil wire.

The resistance of this total length of wire was measured on a L.C. R. impedance bridge, type 650-A, made
by the General Radio Company, Cambridge, Mass., U.S.A.

This resistance was approximately 26.4 ohms and the measurement was done in the electronics laboratory of the physics building at Iowa State College.

In order to obtain the six equal divisions, the entire length of the coil was first stretched out in a single length. The wire was then doubled back to form two equal parts. These two wires were paced and marked into three separate and equal lengths. The two end sections were now swung into the center section and six equal lengths were formed.

These lengths of wire were rolled onto a large V-belt pulley, from which it was hoped that they could be unwound and divided into two parts. They became entangled and it was necessary to completely unwind the wires before separation could be secured. This next time they were wound on two different pulleys.

From each pulley, the three wires were unwound onto a square, wooden form which was turning in a lathe. When the first coil was complete, it was removed from the form, and then the second coil was wound in the same manner as the first.

These coils were wound with black tape and then carefully fitted into the generator. After they had been properly shaped to fit the curvature of the generator,
they were removed. The tape was taken off and a coating of glyptal applied. The coils were not taped again and were returned to the generator after the glyptal paint had dried. This was done in order that maximum cooling of the coils would be obtained.

The cast iron legs were sawn from the generator housing in order to eliminate some of the effects of gravity from the position of the generator housing which was now to be pivoted on two bearings. The generator was placed in a lathe and the housing was shaped so two ball bearings could be slipped onto each end of the frame.

The generator, equipped with these bearings, was mounted in a wooden frame and the generator armature was driven by a "V" pulley which was belted to a variable speed electric motor.

b. The external resistance. The external resistance was made from No. 20 constantan wire. The cross-sectional area of No. 20 wire is given by Marks (16, p. 1987) as 1020 circular mils. The resistivity of constantan is quoted by Hausmann (11, p. 374) as 295 ohms per circular mil per foot.

The generator had been rated as a 9 ampere motor. From this the assumption was made that an external resistance would be sufficiently large if it reduced the current to 2 amperes. Such a resistance must be equal to
16 ohms if it is to be used on a 32 volt source. This was calculated from Ohms law.

The resistance of a wire is given by Hausmann (11, p. 372) as

\[ R = \rho \frac{l}{A} \]  (5.1)

where \( \rho \) is the resistivity of the metal, \( l \) the length, and \( A \) the area.

When \( \rho \) is expressed in ohm-circular mil per foot and \( A \) is given in circular mils, the length \( l \) must be expressed in feet.

Rearranging equation (5.1), we have

\[ l = \frac{RA}{\rho} \]  (5.2)

We have already stated the type of wire we wish to use, including its resistivity and cross-sectional area. We have further said that we wish to obtain a resistance of 16 ohms. Substituting these values into equation (5.2) we obtain

\[ l = \frac{16 \times 1020}{295} = 55.3 \text{ feet}. \]

For the initial setup, only 31 feet of constantan wire was secured.

A resistance unit was constructed from this wire in the following manner. The cotton covering was first removed and then the wire was made into a small spring.
This job was done on a lathe into which a 1/2-inch round wooden stake had been placed. The lathe was set in a thread-cutting position and at a low number of threads per inch.

One end of the wire was fastened to the stake at a point close to the lathe chuck. The machine was thrown into reverse operation and the slow travel of the carriage was used to evenly wind the wire on the stake.

This spring of resistance wire was removed from the stake and hooked across a 220 volt A.C. source of electricity. The wire became white hot. After a half-minute the wire was unplugged from the A.C. circuit and allowed to cool. This operation coated the surface of the wire with an oxide and made it dull in color.

A wooden spool, approximately six inches long by four inches in diameter was constructed by placing small, circular wooden sticks at definite intervals around the circumference of a circle. These sticks were supported at the ends by holes which had been drilled in two blocks of wood.

The resistance spring was now wound about the circumference of the spool to form the complete resistance unit.

Any portion of the total length of wire could be used for the load on the generator.
c. The instantaneous multiplier. Further reference to the term "instantaneous multiplier" in this thesis will be given as I.M. and it will be understood that a multiplier of the pivoted beam type is implied.

(1) Initial unit. A rough model of an I.M. was constructed in the spring of 1948 in which the beam consisted of a steel bar, and the load was applied to it by a "V" belt pulley. The hydraulic piston, which supplied the load to the pulley, was carried on a small wooden car. The car was equipped with small "V" belt pulley wheels and ran on a track which consisted of two small steel rods.

The generator housing was restrained by a linkage which connected to a diesel injection piston about 0.6 inches in diameter. A length of rubber tubing was used to connect the diesel injection cylinder to the cylinder which was on the wooden car. The outside diameter of the tubing was a half-inch and the inside diameter was one-eighth of an inch.

The cylinder on the car was one taken from an airplane and it had an area approximately three times that of the diesel cylinder.

The cylinder which was used to restrain the beam was a war surplus unit meant to operate the bomb bay doors on a B-29 airplane. The outside diameter of this cylinder was approximately 2-3/4 inches.
The piston acted on the beam at a point which was only two inches from the pivot point.

The hydraulic gage used to record the pressure in the restraining piston was made for an oxygen tank in acetylene welding. It was made by Smith Corporation and was capable of measuring 30 p.s.i.

This rough unit was tested and did not prove satisfactory; but valuable information was gained.

In general, the parts were not constructed well enough, and the forces of friction and gravity caused serious interference.

The whole system was sluggish, and some attempts were made to free it up.

The hydraulic cylinders used to load the beam were placed in equilibrium by the addition of lead weights to the light airplane cylinder. Brake fluid was used in these cylinders instead of No. 10 oil. This was not satisfactory as excessive leakage occurred at the airplane cylinder.

The movement of the beam was considered excessive; this was largely due to the close coupling between the pivot point and the restraining piston.

The system was not abandoned because no errors were found which were incapable of being remedied.

This unit was scrapped and a new attempt was made to
Fig. 3b. Instantaneous Multiplier
build an I.M. which would contain few or no interfering faults.

(2) Final unit. This new unit was constructed when more time was available and care was taken to make it well. In order to overcome the problem of cylinder leakage, four used diesel injection pistons were secured from the Diesel generating plant at Story City.

A new beam was constructed by welding together two small channel irons. This beam was smoothed off on the top and bottom by a surface grinder. The final outside dimensions were 22-7/8" x 1" x 1-1/2".

The beam was mounted so the depth was greater than the width.

A hole was drilled through the side of the beam 14 inches from one end, and a steel bushing was pressed into these holes. A short steel shaft was placed through the bushing and made secure by two allan set screws. This steel shaft was ground a ten-thousandth of an inch oversize in a centerless grinder, and two small ball bearings were pressed onto the ends. Two brass plates, three inches square, were drilled in the center and into these the two pivot bearings were inserted with a light press fit. The brass plates were drilled and tapped at the outside corners and were fastened, each to a strip of 3/16-inch flat iron by means of eight small screws. These
screws passed through holes drilled in the flat irons and were anchored in the tapped holes of the brass plates. The holes through the flat irons were made considerably larger than the screws in order to permit a leveling of the beam crossways.

The flat iron strips were drilled with a single hole at the base, and by means of this were fastened to the central section of an "I" beam. Two brass cylinders acted as spacers for the flat irons and were placed on both sides of the I beam; they had been drilled through the center and a bolt was now passed through all five members. In this way the flat iron strips were securely attached to the I beam.

The I beam was 3 inches high by 2-5/16 inches wide. It had been secured to act as a base for the I.M., and also to provide a flat surface upon which a sliding unit could rest.

The total length of the I beam secured was 7 feet, but only 22-7/8 inches were used for the I.M.

One flange of the I beam was chosen for the top, and the upper surface of this was cut smooth in a milling machine. Since this surface still offered considerable sliding friction, the I beam was fastened on a grinder, and the surface was ground smooth. This grinding action was best done with a very quick stroke and a light cut.
It was also necessary to trim the stone with a diamond several times in order to keep it cutting freely.

A brass plate 5x3-1/2x1/2 inches (approximately) was secured as the bearing for the smooth upper surface of the I beam. In order to keep this plate integral with the I beam, except for sliding motion, two brass bars were bolted to the edges of the plate. These bars were cut in a milling machine so they contained a groove, which was slightly larger than the edge of the upper flange on the I beam. This flange was formed by turning the I beam on its side and bolting it to the bed of a milling machine. The underside of each flange was cut away to approximately 1/2 inch depth. It was this flange and groove arrangement which anchored the sliding plate to the machine and guided its movement.

One of the Diesel injection pistons was used to slide the plate. It was desirable that this movement be at least five inches. If possible, a ten-inch movement would be best, since this would increase the accuracy of the machine as far as drawbar loads were concerned.

The Diesel injection piston would only permit a throw of four inches (approximately), but its permissible throw was increased to over eight inches in the following manner.

The top of the piston was turned down in a lathe until it was smaller than the plunger itself. The piston
was made of exceedingly hard steel and it was necessary to use a cutting tool of sintered tungsten carbide in order to turn it. The cutting tool remained continuously at a red heat during the operation. It is possible with a cutting tool of this metal to cut steel so hard that the shavings burn up as soon as they are cut.

With the piston cut away, it was now possible to sink the piston way down in its bore and still maintain an inch or so of precision contact.

It was in this way that the throw of the piston was doubled.

When the piston was in the receded position, it protruded several inches out of the base of the cylinder. It became necessary to increase the length of the cylinder in order to accommodate the piston.

The cylinder was made of high grade cast iron and was therefore quite soft.

The outside diameter of the cylinder was 1-5/8 inches (approximately) and was threaded with a 1-1/4 inch die on a pipe threading machine (no oil was used, since the lubrication is not needed for cast iron or brass).

A 1-1/4 inch pipe coupling, a 5-inch length of pipe, and a pipe cap were added to the cylinder to produce an overall cylinder length of 13 inches.

When the piston was in the receded position, it was
also necessary that it be lengthened by an amount sufficient to ensure a projection of at least two inches beyond the end of the cylinder.

The end of the steel cylinder already had a small circular projection, which was a half-inch in diameter and 1-1/4 inches in length. This projection was used to secure a circular steel bar to the top of the piston. The end of the bar was drilled slightly larger than the projection and the union of the two was made secure by an allan set screw, which was entirely embedded in the bar.

The steel bar was turned to 0.975 inch in diameter and so was smaller than the piston itself, which measured one inch in diameter.

The other end of the bar was drilled and tapped. A length of threaded rod was used to secure this end to an aluminum housing, which contained a self-aligning bearing. This bearing was attached to the sliding brass plate by a 3/16-inch bolt which passed through both members.

The hole which was drilled in the brass plate was also tapped and its base was cut away in order that the head of the bolt would not contact the surface of the I beam.

A short length of a light spring was placed over
the bolt and between the aluminum housing and the plate. This arrangement provided a means to carry the weight of the piston and yet permit it to float in the cylinder. A burr was placed on the top of the bolt, but it was not tightened.

In order to obtain a travel of the plate that was proportional to the hydraulic pressure in the cylinder, it was necessary to spring load the piston.

Two door springs, 13-1/4 inches long and a half-inch in diameter, were used.

A brass plate 4-1/2x1-1/4x3/8-inches was placed at the end of the cylinder.

The plate was drilled in the center and placed over the short length of pipe that was threaded into the end of the cylinder plug. Two holes, three inches on centers and a half-inch in diameter, were drilled 3/4 of an inch from the two ends of the plate. Two more 1/4-inch holes were drilled vertically downward through the plate and they struck the half-inch holes at their centers. The circular loops at the ends of the springs were each placed in the half-inch holes and were secured here by 1/4-inch bolts, which passed through the loops and the vertical holes of the plate.

At the other end of each spring, there was a metal plug and into it an eye bolt was threaded. These eye
bolts were slipped over a 5/16x4-inch steel pin which passed at right angles through the axis of the piston. The steel pin was contained in a hole which had been drilled through the steel bar at a point close to the outer end. The pin was secured by a 1/4-inch allan set screw.

A 1/16-inch hole was drilled through the pin at each end and small cotter keys were inserted into them.

The cylinder was clamped to the surface of the I beam by two 2-3/8x7/16x3/8-inch brass bars. These bars were spaced approximately 3-1/2 inches apart along the top of the cylinder and were bolted to the upper flange of the I beam by four 3/16-inch bolts.

The hydraulic pressure which is supplied to the spring loaded piston must be directly proportional to the drawbar pull of the tractor, but for the purpose of this experiment the following arrangement was made.

The B-29 bomb bay cylinder was anchored to an iron rail in the cement floor and a load was applied to the piston by a lever which was made from an I beam five feet long. The lever had a mechanical advantage of eight, and it was operated by hand. The weight of the I beam was 5.55 pounds per foot and this weight was responsible for part of the load.

A test was made with one of the door springs and it was found that 15 pounds was required to stretch the springs
through eight inches. The total force which the springs
would exert on the Diesel piston when it was fully ex-
tended was, therefore, equal to 32 pounds (approximately).

The exact area of the B-29 cylinder was not known,
since it was never taken apart; but an estimate was made
by using the outside diameter of the cylinder. This esti-
mate was 4.5 square inches.

The area of the Diesel piston was calculated from the
formula for the area of a circle,

\[ A = \frac{\pi d^2}{4} \]

where \( A \) is the area, \( d \) the diameter.

The diameter of the Diesel piston was one inch and
its area was 0.786 square inch.

The ratio of the areas of the two pistons was, there-
fore, approximately 5.73:1.

Since the maximum load on the small piston was esti-
mated to be 32 pounds, the maximum force required on the
B-29 piston was expected to be 184 pounds. Since the
lever which applied the load had a mechanical advantage
of eight, the force which a man would be required to exert
on the end of the beam was only 23 pounds, provided the
weight of the lever was neglected.

The B-29 cylinder was connected to the Diesel cyli-
der by a 6-1/2 foot length of pressure hose. The outside
diameter of this hose was approximately 3/4 of an inch. The hose was meant for a machine called Blackhawk Porto Power, and in this machine it was used to conduct oil from a piston pump to a loading cylinder. Each end of the hose was protected by a short length of heavy, coil spring.

A needle valve was put in the line at the end of the hose, and a hydraulic gage was used between the needle valve and the Diesel cylinder to measure the oil pressure in the line.

It is necessary to use a needle valve in an actual field condition to provide a means of throttling the fluctuations in drawbar load and to protect the mechanism from violent starting loads. The needle valve should be kept entirely closed until the tractor has started pulling the load and is traveling at normal speed.

The two cylinders and the connecting line were filled with No. 20 oil. Care was taken to get rid of all the air in the system, as this would spoil its operation. In order to replenish any oil leakage, and also to provide a means of bleeding the line, a zerk fitting was placed on the B-39 cylinder. Oil is very easily pumped into the system by first putting it in a zerk grease gun.

The threaded connections except where copper gaskets were used, were coated with shellac. This coating was allowed to dry for several minutes before the parts were put together again.
This procedure was fairly successful, but a very slow leakage still persisted. It was later discovered that a material called "Gasket Goo" made by the Pep Mfg. Co., Inc., New York, was excellent for sealing connections of this kind. A tube of it was secured and used for some of the connections. Its performance was very successful. No oil leakage was noticed where it was used.

The Diesel cylinder, used to load the pivoted beam, was identical to the one already used for placement of the sliding plate, but it was shortened by a couple of inches. The cast iron cylinder was easily cut off in a lathe, but it was necessary to cut off the steel piston by a Norton grinding wheel. The piston was cooled during this operation by a steady stream of water. This job was done by Professor H. W. Maynor of the mechanical engineering department of Iowa State College.

The top of the cylinder would also have been shortened if it had not been made of such hard material. The author did not wish to do any welding close to the smooth part of the piston, as there was a possibility of deforming the piston and spoiling its free movement in the cylinder. Some electric welding was done on the extreme top of this piston, in order to form a bracket upon which a ball bearing was mounted.

The cylinder was threaded and sealed with a 1-1/4
inch pipe plug. It was then set on top of the brass plate and held secure by two short brass bars which rested on the upper rim of the plug.

These bars were bolted to the brass plate by four 1/4-inch bolts. The plate was drilled with four holes and these were tapped. Care was taken to cut the bolts so they would not reach the base of the holes and have contact with the surface of the I beam.

The bars which held the cylinder were placed parallel to the ends of the plate. The cylinder was easily adjusted laterally by loosening the bolts which held the bars, and then sliding the cylinder on the plate until it reached the desired position.

Longitudinal adjustment of the speed-pressure cylinder was possible by changing the point at which the draft-pressure cylinder was clamped to the I beam. It was not possible to clamp the draft-pressure cylinder very tightly to the I beam, as the forces were concentrated at two points on the upper part of the cylinder and this caused the piston to stick. This mounting was not considered good, but it was the simplest to make and served its purpose well, as long as sufficient care was taken.

The ball bearing at the top of the loading piston was intended to roll on the ground surface which was on the bottom of the pivoted beam.
The names of the numbered parts given in Figure 4b are as follows:

1. The Draft Cylinder
2. The Draft-Pressure Hose
3. The Draft Gage
4. The Spring Retaining Plate
5. The Draft-Pressure Cylinder
6. The Draft Springs
7. The "I" Beam
8. The Plate Sliding Piston
9. The Spring Retaining Bar
10. The Self-Aligning Coupling
11. The Speed-Pressure Cylinder
12. The Loading Piston
13. The Rolling Bearing
14. The Sliding Plate
15. The Speed Piston
16. The Speed Cylinder
17. The Speed-Pressure Tube
18. The Speed-Piston Frame
19. The Speed-Force Line
20. The Step-up Disc
21. The Disc Bearings
22. The Disc Shaft
23. The Initial Speed-Force Line
24. The Generator-Housing Bearings
25. The Generator
26. The Generator Rotor Shaft
It was not known if the bearing would remain parallel to the direction of movement, since the piston which carried it was free to turn. In operation, it was found that the bearing did not exhibit any tendency to turn.

The hydraulic pressure required by the loading cylinder was fed to it through a rubber tube which had an O.D. of 1/2-inch and an I.D. of 1/8-inch. The end of the tube was pushed over a tapered, ribbed, metal connector, and this connector was threaded into the cylinder wall. The tube was four feet long and was connected at the other end to the cylinder which was used to take the force of the generator housing. This cylinder shall be called the speed cylinder. This part of the hydraulic system was filled with sewing machine oil. It was also equipped with a zerk fitting. In order for the speed cylinder to take its load from the pulley on the generator housing, it was necessary to build a frame onto the piston. The frame permitted the piston to be loaded on its central axis at a point directly opposite the cylinder cap.

The frame was rectangular in shape; the sides were two 7/16-inch steel rods which were threaded at both ends for a distance slightly greater than one inch. The ends of the frame were composed of two 4x1-1/3x3/8-inch brass.

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# O.D. means Outside Diameter
## I.D. means Inside Diameter
plates. Two holes, 7/16 of an inch in diameter, were drilled in each plate. These holes were spaced 3-5/16 inches on centers, and through these the threaded sections of the rods were placed.

Each threaded end of the steel rods had two burrs on it and it was between these that the brass plates were clamped.

One of the brass plates was drilled in the center with a hole which was slightly smaller than the 35/32 of an inch diameter projection at the top of the piston. The plate was then attached to the piston by pressing the projection through the hole in the plate.

The entire unit was now mounted by drilling and tapping two shallow holes on either side of the cylinder wall. The bolts which were screwed into these holes were held in two vertical slots. These slots were contained in the center of two brackets which had been formed from a 24 inch length of flat iron. The flat iron was 3-1/2 inches wide and 1/4-inch thick. The brackets were made in the following way.

The strip of flat iron was bolted to the bed of a milling machine and the two 3/8x6-1/2-inch grooves were cut. The flat iron was removed, cut in two, and four 1/4-inch holes drilled in each piece. These holes were needed in order to screw the bracket to a wooden base.
The irons were then heated in a forge and bent through 90 degrees. The base formed in each case was 2-7/8 inches long, and the vertical height 9-1/4 inches.

The mounting bolts contained a burr, and the bracket was squeezed between this and the head of the bolt. The bolts were smaller than the slots. Three lock washers were used in order to make the attachment flexible and prevent the bolts from throwing high stresses into the cylinder wall.

The slots were provided because it was thought that it might be necessary to bring the speed piston and the loading piston into a definite relationship as far as gravity is concerned. It was thought that the system of two pistons could be potentially loaded in one direction and by this arrangement annul the initial housing torque of the generator before an electrical load was applied.

The restraining piston was mounted directly above the end of the pivoted beam. In order to do this, two short lengths of two inch diameter brass bars were clamped on either side of the I beam, by two 5/16-inch bolts. These bars were each drilled with a vertical hole, and this hole was tapped all the way through. Two 7/16x24-inch steel rods were threaded at both ends and mounted in these tapped holes.

The horsepower cylinder had a bracket in which two
holes had already been drilled. The spacing of the rods was made to coincide with these.

The cylinder was mounted on these rods by locking each side of the bracket between two burrs.

The bracket holes were larger than the rods and some lateral and longitudinal adjustment of the cylinder was allowed.

A ball bearing had been mounted on the head of the restraining piston. This was done in order to keep vertical all the forces which acted on the beam and also to allow for some beam movement.

A hydraulic gage was mounted on the top of the horse-power cylinder cap. The first gage used was one which had a range from 0 to 5 p.s.i.

The author anticipated that the machine might be sluggish under test and so it was decided to add an auxillary multiplying mechanism to the torque delivered by the generator.

A circular aluminum disc was obtained which was meant to be the chuck for a wooden lathe. This disc contained a 3/4-inch threaded hole in the center. A 3/4-inch steel shaft 13-3/4 inches long was turned to 9/16 of an inch diameter at the ends and fitted with two ball bearings.

The shaft was then threaded from one end, a distance of 6-3/4 inches. This length of threading was sufficient
to allow the disc to be placed opposite a wire which was connected to the outside of the generator housing. It was necessary to use the outside of the generator housing as a pulley in order to prevent it from swinging too far when placed under load.

The disc was locked to the shaft by a 3/4 N.F. # burr.

The two bearings containing the shaft were mounted in two blocks of wood and these wooden pieces were attached to the wooden frame upon which the generator was pivoted.

The disc was 6 inches in diameter and its shaft was 3/4 of an inch.

The multiplication between a force applied at the rim of the disc and one taken off at the circumference of the shaft was, therefore, eight times.

The original curves obtained for the generator were those which resulted from the use of a 2-1/2 inch diameter pulley which had been cut on the generator housing. The outside of the generator housing, now used as a pulley, was 6.35 inches in diameter.

The total new multiplication, above that of the old, was calculated to be 5.46 times.

If there had been an easy way to secure a pulley,

#All threading that was done on this machine was national fine thread.
smaller than 1/2 inch, to the generator housing, the con-
struction of the auxiliary multiplier would not have been necessary.

The electrical wires connected to the generator were supported on a wire post and they were kept as loose as possible. No trouble was encountered with them, but if the rotation of the generator under maximum load had been much greater, it would have been necessary to attach a system of slip rings to the generator.

The auxiliary multiplier was hooked up as follows:

A fine copper wire, 1/64 of an inch in diameter, was attached to the outer circumference of the generator housing. The other end wire was wound around the surface of the disc and secured to the circumference by a small hole which was drilled through the rim.

A small hole had been drilled through the 3/4-inch shaft and this hole was used to secure the ends of a violin string. The center of the string was looped around a small steel bar, 1-1/4 x 5/16 inches, which was mounted on the rectangular frame of the speed piston. This steel bar was fastened to the brass plate by two 3/16-inch bolts.

The initial test was made using a 0 to 5 p.s.i. gage on the restraining cylinder. It was found that in the 0 to 5 p.s.i. range the machine was very sluggish and slow to restore itself.
An attempt was made to increase the level of energy at which the machine was operating. This was done by decreasing the electrical resistance of the generator load.

It was found that the 1/4 H.P. electrical motor was no longer capable of turning the generator at the desired speed, and no appreciable increase in torque was obtained.

The electric motor was removed from the set-up and replaced by a 2-1/2 H.P. single cylinder gasoline engine. This engine had ample power to operate the generator, and the vibration which it produced had a loosening effect on the system.

On this final trial, the entire unit functioned quite well. The gage seemed to follow very closely all changes in the speed of the generator and all serious sluggishness seemed to have disappeared from the I.M.

The effect of the position of the loading piston was also very quickly recorded by the gage.

Some leakage of oil was noticed at the restraining piston. It was decided that a heavier oil should be used here in order to keep the beam level.

The action of the sliding plate and loading cylinder was very satisfactory at the beginning, but some sticking seemed to develop. A thorough check into the cause of this was not made. Several things could have been improved upon and these will appear in the discussion part.
of the thesis. The author was satisfied that the machine was capable of working. Three different tests were now made on this machine.

1. Gage pressure versus generator r.p.m. (loading piston at a fixed distance from the pivot point).

2. Gage pressure versus loading piston position (generator speed constant).

3. The action of the spring loaded piston (movement plotted against hydraulic pressure).

The tachometer used to determine the generator r.p.m. was of the clock mechanism type.

The exact position of the loading piston was determined by measuring the distance between the front of the brass plate and the end of the I beam.

The springs used to load the plate-moving piston were not considered to be strong enough from the beginning and they were used only to demonstrate the principle. However, the data taken of their action will still be given.

A final check which was made of the electrical resistance of the armature had a series of values between 1.2 and 7 ohms, depending on the position of the commutator with respect to the brushes.
IV. RESULTS

A. The Initial Generator Test

The initial test of the generator was made by connecting a wire to a 2-1/2 inch diameter circle on the generator frame. This wire was held by a spring balance that had been mounted on a wooden frame. The balance had a 9-inch diameter dial face and was capable of measuring a total of 30 pounds. Ten pounds was measured for every complete revolution of the dial pointer.

The field current was supplied by a six volt battery.

The resistance of the generator load was changed for each test. The values of force obtained were plotted against the r.p.m. of the generator. The tables of results are listed according to the resistance used in the external circuit. Values for $E_t$, the terminal voltage of the generator, and $I_a$, the armature current, are also recorded, but no attempt will be made to plot them on a graph.

The generator was also tested to see what sort of a curve it could produce when the load consisted of tungsten filament light bulbs.

Six volt, 15 watt house bulbs were used and the
calculations were made for a circuit which would not be expensive and which would still give a fair test.

The current flowing in a six volt bulb can be expressed by ohms laws as $\frac{6}{R}$. Since watts are equal to the voltage times the amperage, we may write

$$\text{Watts} = \text{Volts} \times \text{Amps} \quad \text{(6.1)}$$

If we substitute in this equation the values of 15, 6 and $\frac{6}{R}$, for watts, volts and amps we may solve for "R", the resistance of one six-volt bulb. Solving the equation, we obtain $R = 2.4$ ohms.

From this information and the laws governing parallel circuits, a circuit was devised which would have a total resistance of 4.8 ohms. This circuit consisted of two banks of four bulbs.

**TABLE I**

Generator Test

$R = 1.8$ ohms (approx.)

<table>
<thead>
<tr>
<th>Run</th>
<th>R.P.M.</th>
<th>Force</th>
<th>Et</th>
<th>Ia</th>
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<td></td>
<td></td>
<td>lbs.</td>
<td>volts</td>
<td>amps</td>
</tr>
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<td>8.0</td>
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<td>--</td>
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<td>320</td>
<td>2.60</td>
<td>2.8</td>
<td>1.4</td>
</tr>
<tr>
<td>10</td>
<td>145</td>
<td>2.10</td>
<td>1.5</td>
<td>0.5</td>
</tr>
</tbody>
</table>
### TABLE II

**Generator Test**

\( R = 3 \text{ ohms (approx.)} \)

<table>
<thead>
<tr>
<th>Run</th>
<th>R.P.M.</th>
<th>Force</th>
<th>( E_t )</th>
<th>( I_a )</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1670</td>
<td>5.8</td>
<td>19.5</td>
<td>6.8</td>
</tr>
<tr>
<td>2</td>
<td>1420</td>
<td>5.0</td>
<td>16.8</td>
<td>5.6</td>
</tr>
<tr>
<td>3</td>
<td>1120</td>
<td>4.0</td>
<td>13.2</td>
<td>4.5</td>
</tr>
<tr>
<td>4</td>
<td>890</td>
<td>3.1</td>
<td>10.2</td>
<td>3.5</td>
</tr>
<tr>
<td>5</td>
<td>595</td>
<td>2.0</td>
<td>6.8</td>
<td>2.1</td>
</tr>
<tr>
<td>6</td>
<td>425</td>
<td>1.5</td>
<td>4.8</td>
<td>1.5</td>
</tr>
<tr>
<td>7</td>
<td>260</td>
<td>1.1</td>
<td>2.5</td>
<td>0.7</td>
</tr>
</tbody>
</table>

### TABLE III

**Generator Test**

\( R = 7.8 \text{ ohms (approx.)} \)

<table>
<thead>
<tr>
<th>Run</th>
<th>R.P.M.</th>
<th>Force</th>
<th>( E_t )</th>
<th>( I_a )</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>2325</td>
<td>4.65</td>
<td>31.0</td>
<td>4.5</td>
</tr>
<tr>
<td>2</td>
<td>1940</td>
<td>3.90</td>
<td>25.3</td>
<td>3.5</td>
</tr>
<tr>
<td>3</td>
<td>1635</td>
<td>3.40</td>
<td>21.8</td>
<td>2.9</td>
</tr>
<tr>
<td>4</td>
<td>1305</td>
<td>3.00</td>
<td>17.5</td>
<td>2.2</td>
</tr>
<tr>
<td>5</td>
<td>1075</td>
<td>2.60</td>
<td>14.1</td>
<td>1.8</td>
</tr>
<tr>
<td>6</td>
<td>735</td>
<td>2.00</td>
<td>9.0</td>
<td>1.1</td>
</tr>
<tr>
<td>7</td>
<td>430</td>
<td>1.60</td>
<td>5.6</td>
<td>0.5</td>
</tr>
<tr>
<td>8</td>
<td>295</td>
<td>1.30</td>
<td>3.0</td>
<td>0.1</td>
</tr>
</tbody>
</table>
TABLE IV

Generator Test

R = 14 ohms (approx.)

<table>
<thead>
<tr>
<th>Run</th>
<th>R.P.M.</th>
<th>Force</th>
<th>$E_t$</th>
<th>$I_a$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>lbs.</td>
<td>volts</td>
<td>amps</td>
</tr>
<tr>
<td>1</td>
<td>2610</td>
<td>2.85</td>
<td>36.0</td>
<td>2.8</td>
</tr>
<tr>
<td>2</td>
<td>2145</td>
<td>2.40</td>
<td>30.0</td>
<td>2.2</td>
</tr>
<tr>
<td>3</td>
<td>1760</td>
<td>2.00</td>
<td>24.8</td>
<td>1.8</td>
</tr>
<tr>
<td>4</td>
<td>1365</td>
<td>1.50</td>
<td>19.2</td>
<td>1.2</td>
</tr>
<tr>
<td>5</td>
<td>960</td>
<td>1.15</td>
<td>13.0</td>
<td>0.8</td>
</tr>
<tr>
<td>6</td>
<td>820</td>
<td>0.90</td>
<td>10.6</td>
<td>0.5</td>
</tr>
<tr>
<td>7</td>
<td>630</td>
<td>0.76</td>
<td>8.1</td>
<td>0.4</td>
</tr>
<tr>
<td>8</td>
<td>375</td>
<td>0.50</td>
<td>4.0</td>
<td>0.15</td>
</tr>
</tbody>
</table>

TABLE V

Generator Test

R = 4.8 ohms* (approx.)

<table>
<thead>
<tr>
<th>Run</th>
<th>R.P.M.</th>
<th>Force</th>
<th>lbs.</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>2395</td>
<td></td>
<td>6.62</td>
</tr>
<tr>
<td>2</td>
<td>2085</td>
<td></td>
<td>6.21</td>
</tr>
<tr>
<td>3</td>
<td>1900</td>
<td></td>
<td>5.94</td>
</tr>
<tr>
<td>4</td>
<td>1715</td>
<td></td>
<td>5.72</td>
</tr>
<tr>
<td>5</td>
<td>1545</td>
<td></td>
<td>5.47</td>
</tr>
<tr>
<td>6</td>
<td>1275</td>
<td></td>
<td>5.03</td>
</tr>
<tr>
<td>7</td>
<td>1070</td>
<td></td>
<td>4.72</td>
</tr>
<tr>
<td>8</td>
<td>660</td>
<td></td>
<td>4.03</td>
</tr>
<tr>
<td>9</td>
<td>450</td>
<td></td>
<td>3.53</td>
</tr>
<tr>
<td>10</td>
<td>235</td>
<td></td>
<td>3.00</td>
</tr>
</tbody>
</table>

*The generator resistance load was composed of two parallel circuits; each parallel circuit contained four 15 watt, 6 volt house bulbs, hooked in series.
Fig. 5  GENERATOR TEST
Constantan-Wire Load
Fig. 6 GENERATOR TEST
6-Volt House-Bulb Load
A test was made of the force exerted by the generator housing when the field current to the generator was completely cut off.

At 340 r.p.m., the load was 0.69 pounds. It did not vary much from this value regardless of the r.p.m. of the generator.

B. Instantaneous Multiplier

When this unit was tested, the distance from the pivot point of the beam to the central axis of the restraining roller bearing was estimated to be 12-7/8 inches. The value of $X_1$ is, therefore, equal to 12.37.

The distance from the vertical line passing through the pivot point to the end of the I beam was estimated as 13.34 inches.

The measurements used to determine the position of the loading piston were those taken of the distance between the end of the sliding brass plate and the end of the I beam. The position of the loading piston in terms of its distance from the pivot point was calculated, and it is the distance recorded.

The fixed distance of the loading piston, which was used to test the speed performance of the unit, was estimated to be 12.85 inches. It was necessary to remove the
gasoline engine and revert to the electric motor to obtain the lower speeds. The values obtained in this test are put in a separate table and will also be plotted on a separate graph.

For all I.M. tests the external resistance, used in conjunction with the generator, was equal to 7.8 ohms (approx.).

**TABLE VI**

Instantaneous Multiplier Test

Gage pressure versus generator speed
Position of the loading piston constant at \( X_0 = 10.6 \) inches (approx.)

<table>
<thead>
<tr>
<th>Run</th>
<th>R.P.M.</th>
<th>Gage reading</th>
<th>( I_a^{#} )</th>
<th>( I_f^{#} )</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>p.s.i.</td>
<td>amps</td>
<td>amps</td>
</tr>
<tr>
<td>1</td>
<td>4100</td>
<td>27.0</td>
<td>5.0</td>
<td>7.0</td>
</tr>
<tr>
<td>2</td>
<td>3325</td>
<td>22.4</td>
<td>4.4</td>
<td>7.0</td>
</tr>
<tr>
<td>3</td>
<td>2820</td>
<td>19.2</td>
<td>4.4</td>
<td>7.0</td>
</tr>
<tr>
<td>4</td>
<td>2530</td>
<td>17.0</td>
<td>4.0</td>
<td>7.0</td>
</tr>
<tr>
<td>5</td>
<td>2295</td>
<td>15.6</td>
<td>3.6</td>
<td>7.0</td>
</tr>
<tr>
<td>6</td>
<td>1850</td>
<td>12.6</td>
<td>2.6</td>
<td>7.0</td>
</tr>
<tr>
<td>7</td>
<td>1630</td>
<td>10.6</td>
<td>2.6</td>
<td>7.0</td>
</tr>
<tr>
<td>8</td>
<td>1320</td>
<td>9.0</td>
<td>2.0</td>
<td>7.0</td>
</tr>
<tr>
<td>9</td>
<td>1115</td>
<td>7.4</td>
<td>1.8</td>
<td>7.0</td>
</tr>
</tbody>
</table>

\( I_f^{\#} \) is the field current, \( I_a^{\#} \) is the armature current.
Fig. 7  I. M. Test

\( X_0 = 10.61 \text{ Inches} \)
TABLE VII

Instantaneous Multiplier Test

Gage pressure versus generator speed
Position of the loading piston constant
at $X_0 = 10.6$ inches (approx.)

<table>
<thead>
<tr>
<th>Run</th>
<th>R.P.M.</th>
<th>Gage reading</th>
<th>$I_a$</th>
<th>$I_f$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>p.s.i.</td>
<td>amps</td>
<td>amps</td>
</tr>
<tr>
<td>1</td>
<td>2260</td>
<td>13.8</td>
<td>3.6</td>
<td>6.0</td>
</tr>
<tr>
<td>2</td>
<td>1690</td>
<td>10.9</td>
<td>2.9</td>
<td>6.0</td>
</tr>
<tr>
<td>3</td>
<td>1220</td>
<td>7.4</td>
<td>2.0</td>
<td>6.0</td>
</tr>
<tr>
<td>4</td>
<td>770</td>
<td>4.5</td>
<td>1.0</td>
<td>6.0</td>
</tr>
<tr>
<td>5</td>
<td>455</td>
<td>2.2</td>
<td>0.8</td>
<td>6.0</td>
</tr>
<tr>
<td>6</td>
<td>365</td>
<td>1.2</td>
<td>0.6</td>
<td>6.0</td>
</tr>
</tbody>
</table>

TABLE VIII

Instantaneous Multiplier Test

Gage pressure versus position of loading piston
Generator speed constant at 3185 r.p.m.

<table>
<thead>
<tr>
<th>Run</th>
<th>$X_o$</th>
<th>Gage reading</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>inches</td>
<td>p.s.i.</td>
</tr>
<tr>
<td>1</td>
<td>10.18</td>
<td>20.2</td>
</tr>
<tr>
<td>2</td>
<td>9.93</td>
<td>20.0</td>
</tr>
<tr>
<td>3</td>
<td>9.63</td>
<td>19.0</td>
</tr>
<tr>
<td>4</td>
<td>9.43</td>
<td>18.6</td>
</tr>
<tr>
<td>5</td>
<td>9.18</td>
<td>18.5</td>
</tr>
<tr>
<td>6</td>
<td>8.93</td>
<td>18.2</td>
</tr>
<tr>
<td>7</td>
<td>8.68</td>
<td>18.1</td>
</tr>
<tr>
<td>8</td>
<td>8.43</td>
<td>19.5</td>
</tr>
<tr>
<td>9</td>
<td>8.18</td>
<td>16.5</td>
</tr>
<tr>
<td>10</td>
<td>7.93</td>
<td>16.0</td>
</tr>
</tbody>
</table>

(Continued on next page)
TABLE VIII (Cont'd)

<table>
<thead>
<tr>
<th>Run</th>
<th>$X_o$\textsuperscript{**}</th>
<th>Gage reading</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>inches</td>
<td>p.s.i.</td>
</tr>
<tr>
<td>11</td>
<td>7.68</td>
<td>15.5</td>
</tr>
<tr>
<td>12</td>
<td>7.43</td>
<td>14.5</td>
</tr>
<tr>
<td>13</td>
<td>7.18</td>
<td>14.0</td>
</tr>
<tr>
<td>14</td>
<td>6.93</td>
<td>13.5</td>
</tr>
<tr>
<td>15</td>
<td>6.68</td>
<td>13.6</td>
</tr>
<tr>
<td>16</td>
<td>6.43</td>
<td>13.2</td>
</tr>
<tr>
<td>17</td>
<td>6.18</td>
<td>12.6</td>
</tr>
<tr>
<td>18</td>
<td>5.93</td>
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<tr>
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<td>20</td>
<td>5.43</td>
<td>11.2</td>
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<tr>
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<td>4.93</td>
<td>10.4</td>
</tr>
<tr>
<td>23</td>
<td>4.68</td>
<td>9.8</td>
</tr>
<tr>
<td>24</td>
<td>4.43</td>
<td>9.3</td>
</tr>
<tr>
<td>25</td>
<td>4.18</td>
<td>8.6</td>
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<tr>
<td>26</td>
<td>3.93</td>
<td>8.1</td>
</tr>
<tr>
<td>27</td>
<td>3.68</td>
<td>7.6</td>
</tr>
<tr>
<td>28</td>
<td>3.43</td>
<td>7.4</td>
</tr>
<tr>
<td>29</td>
<td>3.18</td>
<td>6.8</td>
</tr>
<tr>
<td>30</td>
<td>2.93</td>
<td>6.3</td>
</tr>
<tr>
<td>31</td>
<td>2.68</td>
<td>5.8</td>
</tr>
<tr>
<td>32</td>
<td>2.43</td>
<td>5.3</td>
</tr>
<tr>
<td>33</td>
<td>2.18</td>
<td>4.7</td>
</tr>
<tr>
<td>34</td>
<td>1.68</td>
<td>2.6</td>
</tr>
<tr>
<td>35</td>
<td>1.43</td>
<td>2.2</td>
</tr>
</tbody>
</table>

\textsuperscript{*}The field current $I_f$ was 7.0 amps. $I_a$ varied between 4.5 and 4.6 amps.

\textsuperscript{**}$X_o$ is the distance the loading piston is from the pivot point of the beam.
Fig. 8  I.M. TEST

R = 7.8 ohms

$X_0 = 10.6$ Inches Approx.
Fig. 9  I.M. TEST
R.P.M. = 3185 Approx.
The springs which were used gave a placement of the loading cylinder and its assembly were considered too weak. They had been used chiefly to demonstrate the method of placement.

A test was still made of them, however, and the results will be presented. There seemed to be some sticking of the sliding brass plate. The exact cause of this sticking was not determined. During the test, the polished surface of the I beam was flooded with #20 oil. The generator speed during the test was approximately 3000 r.p.m. and the load produced by this speed transmitted through the loading piston to the beam.

The position of the sliding piston was checked against hydraulic pressure; first for increasing hydraulic pressure, and second for decreasing hydraulic pressure. The generator speed was approximately 3000 r.p.m., and the pivoted beam was inclined upwards one degree at the free end.
V. DISCUSSION

A. The Feasibility of the Machine

In order for this principle of horsepower measurement to work, it has already been stated that the loading force must be directly proportional to generator speed, and the position of the loading piston must be directly proportional to draft. In other words, if we plot force against generator speed we must obtain a straight line, and if we plot loading piston position against draft we must obtain a straight line.

In Figure 5 it is clearly indicated that a straight line of force versus speed can be maintained even at low r.p.m. if the generator load of constantan wire is 7.8 ohms in resistance. Figure 7 indicates that the straight line characteristics of the curve can be maintained at generator speeds up to 3600 r.p.m.

Figure 9 is a graphic demonstration of the ability of the pivoted beam to multiply a fixed speed by a range of draft and maintain a constant multiplication factor.

The ability to obtain a straight line from plotting "draft hydraulic-pressure" against "Loading Piston" placement is possible if suitable units are used, but exact
placement was not obtained in this machine for two major reasons:

1. The springs used had too low a spring modulus.
2. The load was not a vibrating load, but was constantly applied, first in the one direction and then in the other.

In actual field conditions the drawbar load of a tractor is likely to constantly fluctuate as much as 500 pounds, and in order to read the draft with reasonable accuracy this fluctuation will likely be damped down to 100 pounds of gage fluctuation. This damping-down is accomplished by a needle valve placed in the draft-pressure hose.

A test of the sliding assembly of this machine was made. The forces transmitted through the pivoted beam during the test to the horsepower gage were those which resulted from a generator speed of 3005 r.p.m. The angle of inclination of the pivoted beam was upwards one degree at its free end. The position of the sliding assembly was plotted, first against increasing hydraulic pressure, and second against decreasing hydraulic pressure. The average distance between the two curves was equivalent to 16 p.s.i.

If friction drag is taken to be the same in both directions, we conclude that the curve for no friction would lie half-way between these two curves, provided we
neglect the hysteresis effect in the springs. The amount of friction drag is therefore indicated as equivalent to 8 p.s.i. gage pressure. Since the diameter of the piston is one inch, the estimated friction drag is equal to 6.3 pounds.

If this machine was to be used, it is estimated that 5 per cent accuracy in placement could be obtained if the spring modulus of each spring was 6 pounds per inch.

The final machine should be made so all the loads are carried on ball bearings and the friction drag is small. Such a design will permit the forces which produce placement to remain small. If the forces are small, the size of members needed to withstand these forces will be also small, and a lighter instrument can be made. It would be desirable if the final design was light enough to permit the operator to easily carry it in his hands.

The existing machine appears to be feasible provided the strength of the load placement springs is increased.

B. The Theory of Operation

Originally it was planned to construct an I.M. that would follow directly a calculated arrangement of the formula (2.15),

\[
\frac{A_0 K_7 K_9 K_g}{A_1 X_1} = \frac{1}{375}
\]
but it was found that it was impossible to obtain all of the parts in the sizes asked for by the formula.

The decision was made to go ahead and construct an I.M. from whatever parts were available. The formula was kept in mind and so far as possible, the parts used were kept close to its requirements.

Finally, the construction was altered again in order to produce a machine which would have the maximum chance of free movement.

The main objective became the construction of a machine that would operate, and the exact way in which it operated became secondary in importance.

Now that the machine constants are set, a few calculations will be made to determine the specific type of machine that has been built.

The machine was purposely built very flexible. Whenever possible, features were added that permitted an easy change of the machine constants. By constructing the machine in this manner it was thought that the machine constants could be juggled and final achievement of a total constant of \( \frac{1}{375} \) would be much simpler.

The present values of the constants will be given and applied to the formulas.

If we let the generator r.p.m. be signified by the letter "U", and if we let this r.p.m. be equal to the
tractor speed, in miles per hour, multiplied by a constant, \( M \), we may write

\[ U = MS \quad (6.1) \]

Solving the equation for \( S \), we have

\[ S = \frac{U}{M} \quad (6.2) \]

Equation (2.8) states

\[ P_0 = K_7 S \]

Substituting equation (6.2) in (2.8), we obtain

\[ P_0 = \frac{K_7 U}{M} \quad (6.3) \]

Rearranging the terms, we may write

\[ K_7 = \frac{P_0 M}{U} \quad (6.4) \]

The term \( \frac{P_0}{U} \) is a constant the value of which was calculated in the following manner.

In the I.M. test given in Table VI, the value of \( X_0 \) is constant at 10.6 inches. The value of the constant \( X_1 \) for all the tests has been given as 12.87 inches. Substituting these values in equation (2.5), we have

\[ P = \frac{10.6 A_0 P_0}{12.87 A_1} \]

which becomes

\[ P = \frac{0.825 A_0 P_0}{A_1} \quad (6.5) \]

The diameter of the restraining piston was 0.876 inches,
and that of the loading piston was one inch. From these figures the area of $A_1$ was calculated to be 0.603 square inches, and the area of $A_0$, 0.786 square inches. Substituting these values in equation (6.5) and simplifying, we obtain

$$ P = 1.07 P_0 $$

(6.6)

Transposing, we obtain

$$ P_0 = \frac{P}{1.07} $$

(6.7)

Dividing both sides of the equation by $U$, we may write

$$ \frac{P_0}{U} = \frac{P}{1.07U} $$

(6.8)

Using formula (6.8) and the experimental data of Table VI the values of $\frac{P_0}{U}$ were calculated for all values of $U$.

The average value of $\frac{P_0}{U}$ was $6.26 \times 10^{-3}$. Therefore, we may write

$$ \frac{P_0}{U} = 6.26 \times 10^{-3} $$

(6.9)

Substituting equation (6.9) in (6.4), yields

$$ K_7 = 6.26 \times 10^{-3} $$

(6.10)

From equation (2.9) we may write

$$ K_8 = \frac{X_0}{D} $$

(6.11)

The maximum throw which can be obtained from the plate-sliding piston on this machine is 10 inches. If we let the maximum draft to be measured be 2940 pounds, we
may substitute these last two values into equation (6.11) and obtain
\[ K_8 = \frac{10}{2940} \]
Solving,
\[ K_8 = 3.4 \times 10^{-3} \]
(6.12)
The values of \( A_1 \) and \( A_0 \) are already given as 0.603 and 0.786 square inches.
The value of \( X_1 \) has also been given as 12.87 inches.
Substituting the values of \( A_0 \), \( K_7 \), \( K_8 \), \( A_1 \), and \( X_1 \) into equation (2.15), we have
\[ \frac{0.786 \times 6.26 \times 10^{-3} \times 3.4 \times 10^{-3} \times K_9}{0.603 \times 12.87} = \frac{1}{375} \]
Simplifying the equation, we obtain
\[ 2.15 \times 10^{-6} M \times K_9 = \frac{1}{375} \]
(6.13)
Therefore,
\[ MK_9 = \frac{1}{2.15 \times 10^{-6} \times 375} \]
which becomes
\[ MK_9 = 1240 \]
(6.14)
Rearranging the terms, the expression becomes
\[ K_9 = \frac{1240}{M} \]
(6.15)
This is the equation for the gage constant \( K_9 \).
From equation (6.1) we may write
\[ M = \frac{V}{S} \]
(6.16)
Substituting equation (6.16) in (6.15), we have

\[ K_9 = \frac{12403}{U} \quad (6.17) \]

If we let the maximum tractor speed which we wish to test with this machine be six miles per hour, equation (6.17) becomes

\[ K_9 = \frac{7450}{U_1} \quad (6.18) \]

where \( U_1 \) is the generator r.p.m., which is equivalent to the maximum tractor speed of six m.p.h. The maximum generator speed reached under test was 4100 r.p.m.

The straight characteristic of the curve plotted in Figure 7 is not affected until the r.p.m. is above 3600, but it was noticed during the test of the generator that considerable vibration began to occur when the generator speed became greater than 3000 r.p.m. It was decided, therefore, that the maximum generator speed should not exceed this amount, but its value should be as large as other factors will permit.

The remaining dictating factor is the decision that the gage constant \( K_9 \) shall be a whole number.

If the gage constant is equal to 2, the value of \( U_1 \) in equation (6.18) is 3720 r.p.m. This is beyond the set limit of 3000 and will not be allowed.

The final solution to equation (6.18) is to allow the
gage constant to be equal to 3,

\[ K_9 = 3 \] (6.19)

Therefore, one p.s.i. of pressure on the gage indicates 3 horsepower.

The maximum generator speed, \( U_1 \), can be expressed from equation (6.18) as

\[ U_1 = \frac{7450}{K_9} \] (6.20)

Substituting in the value of 3 for \( K_9 \), we solve equation (6.20) for \( U_1 \),

\[ U_1 = 2483 \] (6.21)

The maximum generator speed is now fixed at 2483 r.p.m.

for a tractor speed of six miles per hour.

From equation (6.15) we can write

\[ M = \frac{1240}{K_9} \] (6.22)

Substituting 3, the value of the gage constant into equation (6.22) and solving for \( M \), we have

\[ M = 413 \text{ (approx.)} \] (6.23)

Substituting the value of \( M \) in equation (6.1), we may write

\[ U = 4133 \] (6.24)

The generator speed should be equal to 413 times the speed of the tractor in m.p.h.

From equations (6.11) and (6.12) we have
\[ \frac{X_0}{D} = 3.4 \times 10^{-3} \]

which may be written

\[ X_0 = 3.4 D \times 10^{-3} \quad (6.25) \]

We have said that the maximum values of tractor speed and draft which we intend to measure with this I.M. are six miles per hour and 2940 pounds.

The maximum horsepower that we intend to measure, calculated from equation (1.1), will be equal to

\[ \frac{2940 \times 6}{375} \]

which is 47 horsepower.

The size of gage required to meet the maximum conditions must be capable of measuring a pressure of 47 divided by 3, or 15.6 p.s.i. The present gage on the I.M. reads up to 30 p.s.i.

Now that the exact working of the machine has been established a few further observations will be made on the experimental data.

Since the curves for 1.8 and 3 ohms in Figure 5 are concave upwards for low generator speeds, and since the curve for a tungsten filament in Figure 6 is concave downwards for the same low speeds, it may be possible to make a combination of constantan and tungsten which will produce a straight line curve for the housing torque at low speeds.
If this cannot be done, equation (6.34) and the experimental curves indicate that the I.M. will not be accurate for a tractor speed below one mile per hour. Since the experimental data was read from a small oxygen-tank, bourdon gage, which was only 2-1/4 inches in diameter, the accuracy of reading may not have been good. For this reason the upper portion of the curve in Figure 8 is discounted. No similar falling away of the curve at this point is noted in Figure 7.

In Figure 9 the variation of the data from a perfectly straight line may be partly ascribed to the following:

1. The operator may not have read the horse-power dial correctly.
2. The generator speed may not have remained constant at 3135 r.p.m. First because the speed of the gasoline motor may have varied, and second because slippage of the V belt drive may have occurred.
3. Some alteration in the exact point of beam loading may have occurred due to slight imperfections in the mechanism. That is, the calculated and the actual distance of \( X_0 \) may have been different.

For these reasons no special attempt was made to estimate the accuracy of the pivoted beam method of multiplication.
The maximum error indicated by the point which seemed farthest off the straight line curve was measured. This error was -0.6 p.s.i. for a curve value of 14.2 p.s.i. The percentage of error is equal to \( \frac{0.6}{14.2} \times 100 \). The maximum error indicated by the curve for the I.M. is, therefore, equal to 4.23 per cent. The average error indicated by the curve is estimated to be equal to 2 per cent.

The curve for a resistance of 7.8 ohms in Figure 5 was not used to determine the variation of the electrical generator from a strictly straight line characteristic; but the values of \( \frac{P_0}{U} \), as calculated from Table VI, were used to determine a percentage of error.

The maximum variation from the average of \( 6.26 \times 10^{-3} \) was \( 6.08 \times 10^{-3} \). The error is \( 0.18 \times 10^{-3} \), and the percentage error is equal to

\[
\frac{0.18 \times 10^{-3}}{6.26 \times 10^{-3}} \times 100 = 2.88\%
\]

The maximum error introduced by the generator between 1115 and 4100 r.p.m. is equal to 2.88 per cent.

A further estimate of the generator's deviation from straight line torque on its housing was made by the use of Figure 8. The last point on the curve exhibits an error of 5.8 per cent. It is hardly fair to quote high percentage of error which occurs for only one point, especially since the curve in Figure 7 indicates for
approximately the same generator r.p.m. an error of only 2 per cent.

A summary of the work indicates that the maximum I.M. and generator errors are equal to 4.2, and 2.8 per cent approximately; but the average error for each of these units is estimated to be 2 per cent. The total average error is, therefore, expected to be approximately 4 per cent.
VI. CONCLUSIONS

1. A satisfactory approach has been reached in the problem of direct and instantaneous horsepower measurement.

2. The laboratory model constructed indicates that it is possible to accurately multiply a force and a distance by hydraulically applying forces to a member which is in equilibrium about a pivot point.
VII. SUMMARY

The use of alternating current to measure horsepower was not practical because of the losses which were introduced by hysteresis and eddy currents in the model constructed and in the calculations made for a model containing laminated Alnico.

It was possible to obtain a force directly proportional to speed by the use of a separately excited D.C. generator connected to a constant resistance load. This force was the torque that was exerted on the generator frame. The maximum error was found to be 2.83 per cent, but the average error was estimated to be 2 per cent.

A maximum error of 4.33 per cent was obtained when a force and a distance were multiplied by the machine in this thesis. The average error in this operation was expected to be 2 per cent. The result of the multiplication was indicated directly on a dial.

It was estimated that two springs with a modulus of at least six pounds per inch would have to be used on this laboratory model if the error of load placement was to be less than 5 per cent.

One solution to the laboratory model was a setup where the generator was run at a speed 413 times the speed
of the tractor in miles per hour, the loading piston movement was one inch for every 294 pounds of draft, and every p.s.i. of pressure on the horsepower gage was equivalent to three horsepower.

The above solution was one of several which could have been used for the laboratory model. The final design of this unit is expected to be quite different in construction. It is hoped that a unit may be produced which will be very accurate and practical.

Since it is possible to obtain a force which is a measure of the torque transmitted to useful work, it is possible to adapt the device of this thesis to the direct and instantaneous measurement of torsion horsepower also.

With the use of an infinitely variable speed increaser, it may be possible to quickly set the device to test any car, truck, train or tractor.
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IX. VITA

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XI. APPENDIX
Symbols Used

\( A = \) area in sq. cm.
\( A_0 = \) area of loading piston in sq. in.
\( A_1 = \) area of restraining piston in sq. in.
\( C.M. = \) circular mils
\( D = \) draft in pounds
\( E = \) voltage
\( f = \) frequency
\( F = \) restraining force
\( F_0 = \) loading force
\( H.P. = \) horsepower
\( I = \) current in amperes
\( I_a = \) armature current
\( I_e = \) external resistance current
\( K_1, K_2, K_3, \ldots \) etc. = constants
\( l = \) length in feet
\( l_c = \) length in centimeters
\( L = \) inductance
\( M = \) a constant such that \( M S = U \)
\( m.p.h. = \) miles per hour
\( n = \) the number of turns of wire
\( N = r.p.m. \)
\( P = \) pressure on the restraining piston
\( P_0 = \) pressure on the loading piston
p.s.i. = pressure in pounds per sq. in.

R = resistance in ohms

R_a = armature resistance

R_e = external resistance

r.p.m. = revolutions per minute

S = speed in miles per hour

T = torque

U = the generator r.p.m.

U_1 = maximum generator r.p.m.

V = terminal voltage of generator

X = inductive reactance

X_o = distance of loading force to pivot point

X_1 = distance from pivot point to restraining piston

Z = impedance

ϕ = flux

μ = permeability

ρ = resistivity