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# Design and construction of a torsion dynamometer

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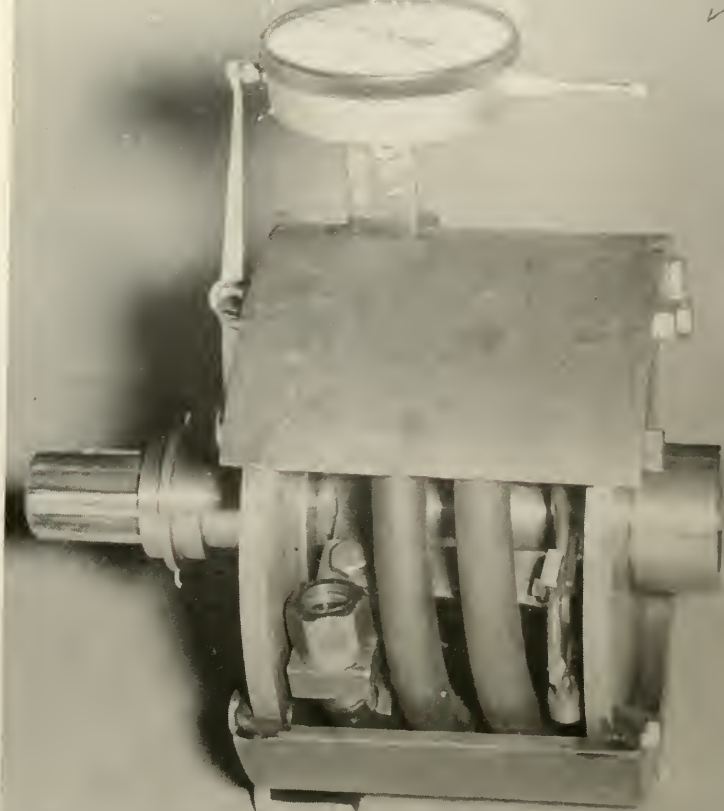
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# Design and Construction
# of a Torsion Dynamometer

## THE AUTHORS

Authors of *Design and Construction of a Torsion Dynamometer* are P. John Zachariah, formerly Assistant in Agricultural Engineering, and Ross Phillips, Assistant Professor of Agricultural Engineering, and Assistant Agricultural Engineer in the Agricultural Experiment Station.

# Design and Construction of a Torsion Dynamometer

## Introduction

P. JOHN ZACHARIAH and ROSS PHILLIPS

MEASURING devices are essential to the development of any mechanical art or science. While dynamometers have been used for more than a century most of these instruments are cumbersome and unsuited for measuring the power requirements of tractor-drawn farm implements which are powered by the power-take-off of the tractor. The dynamometers which have been used for measuring the power requirements of farm machinery are large, mounted on trailers, and expensive. The dynamometer described in this Bulletin is light in weight, cheap, sufficiently accurate, and does not change the normal hitch of farm implements. It has been field tested with a number of machines.

## Design Details and Construction

The torsion dynamometer is based on the principle that an elastic member, when subjected to a bending load within its elastic limit, will be deflected proportional to the load applied. Since stress developed at any cross section along the length of the member and deflection between any two points on the member have a constant relation, loads imposed on the member can be measured by measuring this deflection. This physical principle was combined with other design details to obtain the following features:

1. Adaptability for measurement of torque
   a. transmitted through the power-take-off of any make of tractor
   b. transmitted to machines of both trailing and mounted type without excessively disrupting the normal hitch position
   c. to measure 2 to 24 H.P. at rated power-take-off speed of 540 RPM
2. Sufficiently light in weight to facilitate easy handling.
3. Sturdiness and compactness.
4. Simplicity in construction so that it can be easily operated.
5. To give fairly accurate results while testing implements under normal field conditions.

3

The dynamometer consists of an input shaft and an output shaft on the same axis connected by a helical spring. The input shaft (5 in Fig. 1) is connected to the tractor power-take-off or power source, and the output shaft (6 in Fig. 1) is connected to the machine to be tested. When power is transmitted to a driven machine through the dynamometer, the spring connecting the two shafts deflects in bending. By means of suitable linkages, (the distortion of the spring is proportional to the torque applied) the dial gage reading is a linear function of the torque transmitted. Another instrument, not integral to the dynamometer, indicates the revolutions per minute of the shaft. From the torque and revolutions per minute of the shaft, horsepower can be calculated.

Internal splines were cut in the outer end of the input shaft (Fig. 2), the splines conforming with American Society of Agricultural Engineers
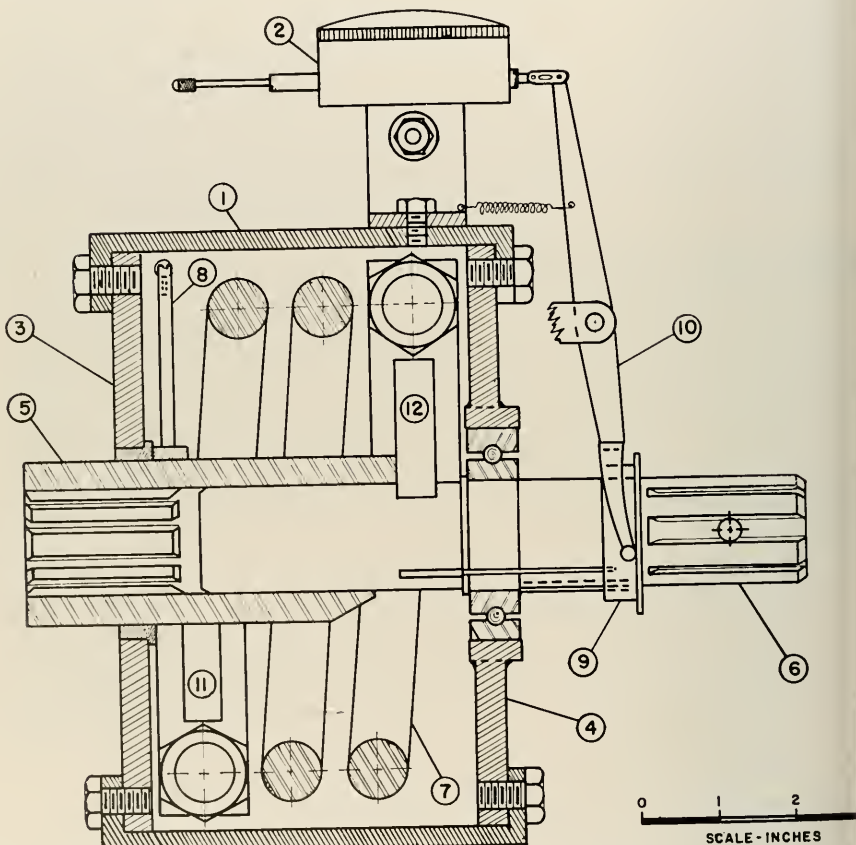


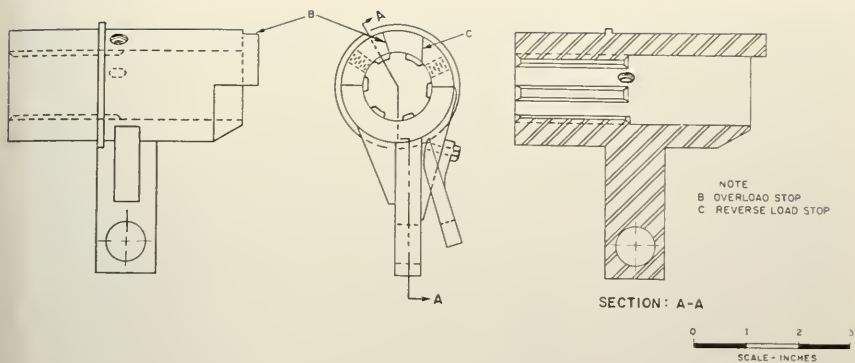FIGURE 1. Sectional view of the torsion dynamometer developed at We: Virginia University.

4

NOTE
B  OVERLOAD STOP
C  REVERSE LOAD STOP

SECTION : A-A

FIGURE 2. The torsion dynamometer input shaft with arm.



SCALE - INCHES

SECTION A-A

FRONT VIEW

END VIEW
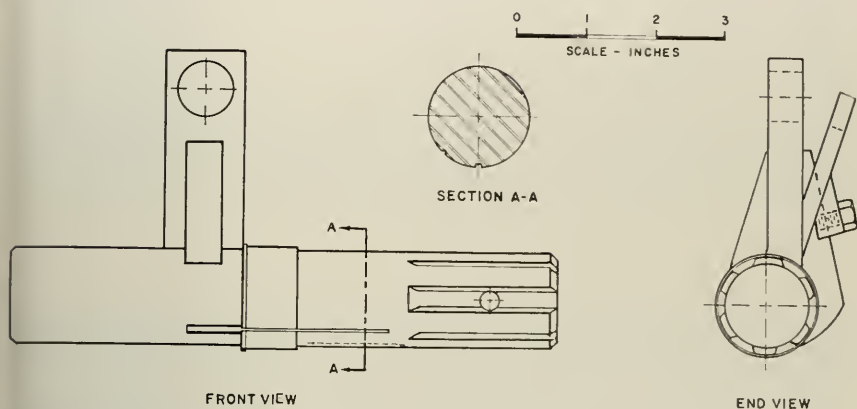
FIGURE 3. The torsion dynamometer output shaft with arm.

(ASAE) Standards for 1⅜-inch power-take-off shafts. The other end of the shaft was bored to fit a portion of the output shaft so that the two shafts would remain in alignment. The other end of the output shaft was made with an external 1-⅜ ASAE spline.

The helical spring (Fig. 4) which formed the primary element of the dynamometer, was made of ¾-inch diameter V.D. Tool Steel. The spring consisted of 2½ turns with a pitch of 1⅛ inches and a mean coil diameter of 6 inches. Nuts were used to connect the spring ends to the two shafts through their shaft arms.

When there is no load the shaft arms rest 180 degrees apart. The spring winds tighter when torque is applied. The spring ends are rigidly connected to the shafts through arms (11 and 12 in Fig. 1). Deflection in the spring under transmission of torque changes the relative position of the two shaft arms. This relative motion is transmitted to a dial gage.
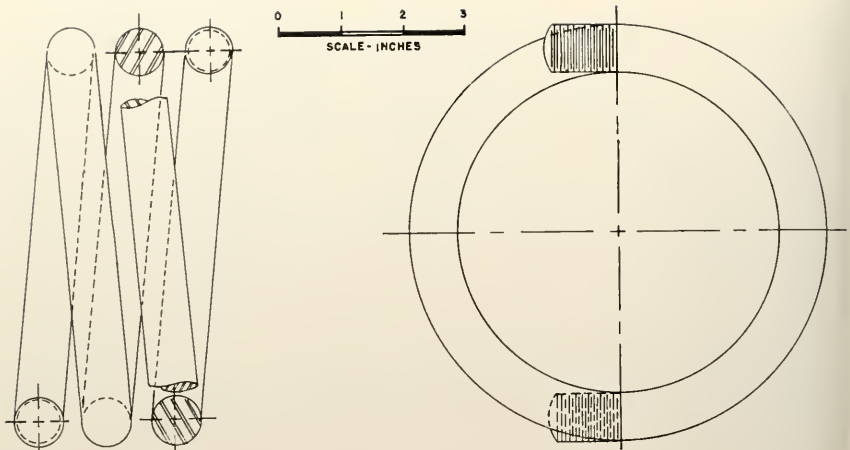
5

**FIGURE 4. Torsion spring of the dynamometer.**

The deflection indicator arm (8 in Fig. 1) and a sliding collar (9 in Fig. 1) are connected by means of a 0.012-inch diameter music wire passing through a copper tube fixed to the output shaft. A lever transmits the movements of the sliding collar to the dial gage. The collar is free to slide axially on the shaft, but a key prevents it from rotating on the shaft.

Change in angle or clockwise rotation of the input shaft with respect to the output shaft causes arm 8 to move away from the end of the copper tubing (Fig. 6). Under no-load condition, as shown in Figure 5, arm 8 remains close to the copper tubing. The wire passing through it re mains taut due to the extension spring connected to the lever arm 10 An increase in torque causes arm 8 to move away from the copper tubing and a decrease in torque reduces the gap between the arm and copper tubing. Thus, with the arm and wire, the angular change between the spring ends is translated into a proportional axial movement of the sliding collar 9, Figure 1. The gage readings, when calibrated in known torque values, indicate the torque transmitted through the system.

Torsional overloads are often imposed during the operation of a driven implement. This may be due to a sudden release of the clutch clogging of the implement, or variation in field conditions. Factor affecting the amount of overload are: the amount of kinetic energ stored in the rotating parts of the tractor; the moment of inertia of th rotating parts of the implement; the amount of resilience in the driv between the rotating parts of the tractor and that of the driven imple ment; and the amount of torque transmitted to the implement.

Protection of the spring and indicator from torsional overloads an reverse loading is provided with overload and reverse load stops. Whe

The copper tubing connected to the out-put shaft

The deflector arm connected to the in-put shaft
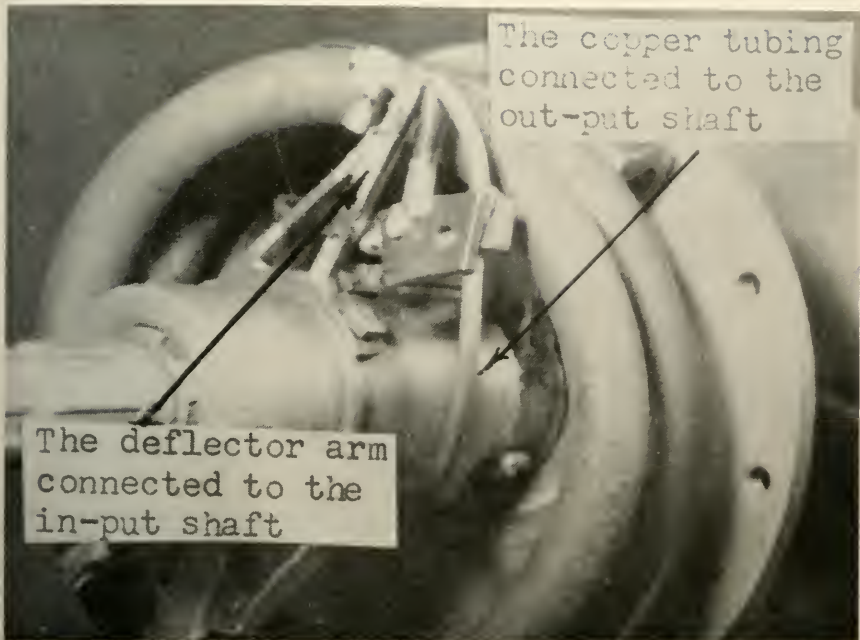
FIGURE 5. Relative positions of the deflection indicator arm and end of the copper tubing when no torque is applied. (There is little or no clearance between end of the copper tube and the deflection indicator arm.)



Under .25 full deflection

Under full deflection
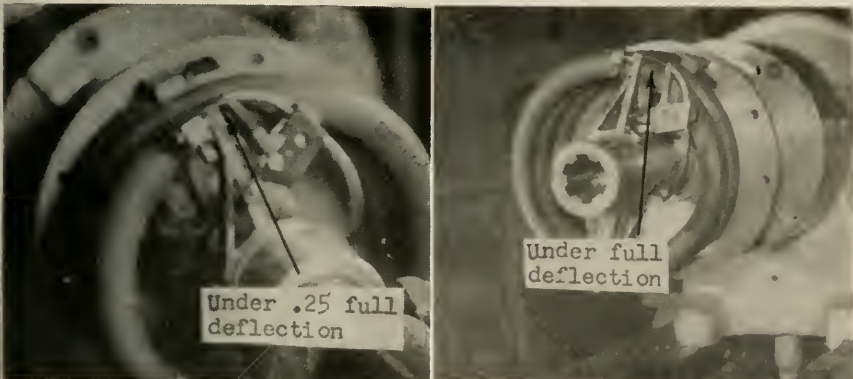
FIGURE 6. Relative positions of the deflection indicator arm and end of the copper tubing when torque is applied. (Displacement has occurred between the arm and copper tubing.)

a load beyond the safe limit of the spring is transmitted, the excess is transferred directly from one shaft to the other through the stops. The inside of the input shaft is slotted as shown in Figure 2. The slot is wide enough to accommodate part of the output shaft with clearance to allow the output shaft to rotate 15 degrees with respect to the input

7
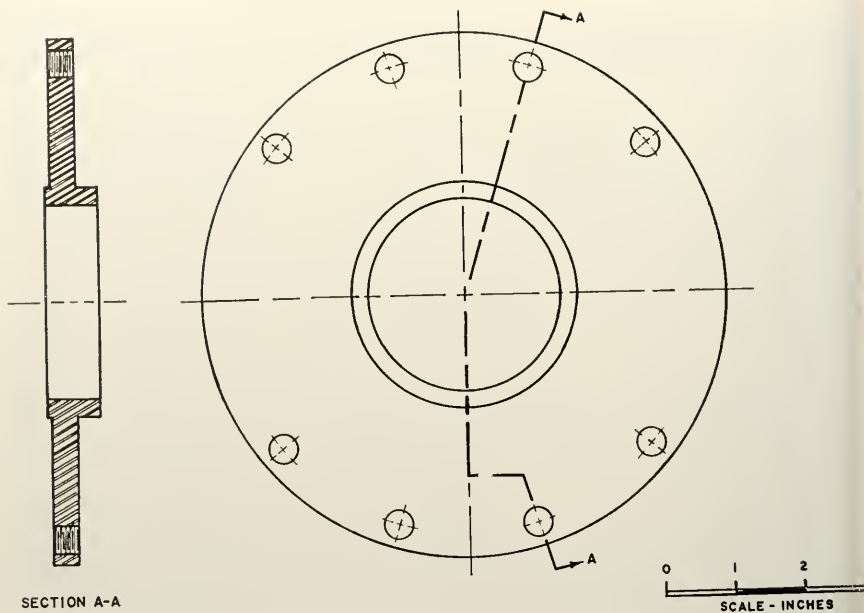
SECTION A-A

FIGURE 7. Rear plate of the torsion dynamometer housing.
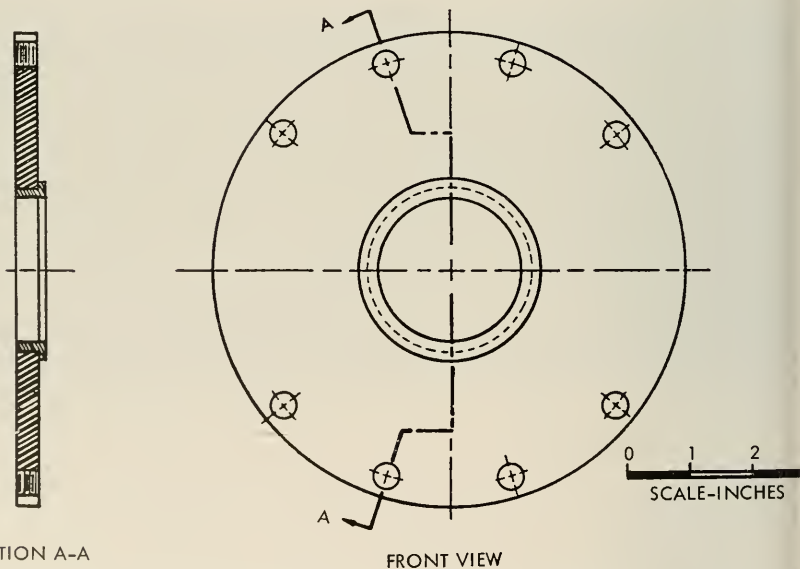


SECTION A-A                    FRONT VIEW

FIGURE 8. Front plate of the torsion dynamometer housing.

8
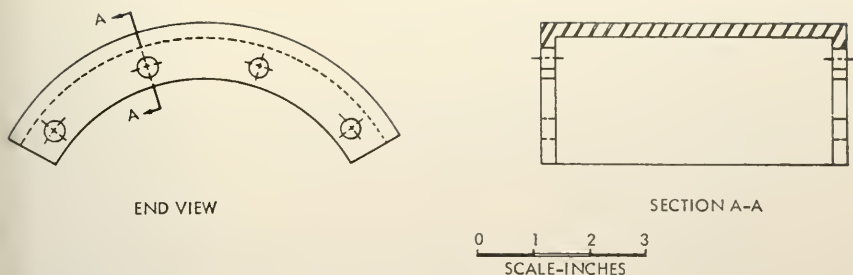
FIGURE 9. Housing sector of the torsion dynamometer.

shaft. When the torque tends to deflect the spring more than 15 degrees, the slot in the input shaft makes contact with the arm of the output shaft. This prevents the spring from deflecting beyond the set value and the system acts as a rigid coupling. With reverse loading, the other side of the slot in the input contacts the output shaft arm and the dynamometer will again work as a rigid coupling.

ASAE recommendations of operating requirements for power-take-off drives include incorporation of a protective device. For an implement driven by a 1⅜-inch power-take-off shaft, a smooth frictional type slip clutch which does not exceed a break-away value of 4,000 lb-in. is suggested. Most farm implements have this or an equivalent safety device. As a precautionary measure, the dynamometer was designed to withstand a much higher torsional overload.

Stress in the spring was calculated thus:

Maximum deflection of spring before the overload stop would prevent the spring from deflecting further = 15° or 0.2618 radians.

Moment of the spring arms when the spring deflection is 0.2618 radians is: $\Theta^* = \dfrac{ML}{EI}$     $M = \dfrac{\Theta EI}{L}$

Where M = moment in inch pounds

L = effective length of the spring = 43.5 inches

I = moment of inertia = 0.01553 for a ¾ inch round section

E = modulus of elasticity in bending = 30 x 10⁶ (approx.) for steel

By substitution

$$M = \frac{0.2618 \times 30 \times 10^6 \times 0.0155}{43.5}$$

= 2,800 lb-in.

*Convential symbol was not available.

9

Overload stops take any moment exceeding 2,800 lb-in. applied to the system. These calculations were:

> Bearing area of overload stop = 0.187 in.²
> Force the overload stop can carry when the maximum allowable stress is 30,000 psi
> = .187 x 30,000
> = 5,610 lbs.
> Moment the overload stop can carry when lever arm of stop is equal to .875 in.
> = 5,610 x .875
> = 4,900 lb-in.
> Total moment or torque the spring and overload stop can carry
> = 2,800 + 4,900 = 7,700 lb-in.

Section four of ASAE recommendations (1) on operating require ments for power-take-off drives states:

> Implements subject to high starting loads or plugging should be equipped with an overload protective device in the power line which will protect the drive against torsional overloads of sufficient magnitude to cause mechanical failure of either tractor or implement parts. In consideration of the foregoing factors, it is desirable for implements to conform to the following conditions:
> a. The instantaneous operating loads should not exceed 7,500 lb-in. for the 1⅜-inch diameter shaft.

The spring used in the dynamometer construction was made from a 49-inch length of ¾-inch diameter V.D. Tool Steel Rod (SAE W2-1.0 Carbon-V). The alloy composition (23) of this steel is: Carbon, 1.00% Manganese, 0.23%; Vanadium, 0.18%; Silicon, 0.25%.

When steel is raised above the 400° to 450° F. range for tempering there is a decrease in yield strength (13). Also, alloy steels tempered between 425° and 750° F. (18) has a low impact strength.

The rod was hot-wound to form a spring of approximately 2½ turns with a mean coil diameter of 6 inches and a pitch of 1⅛ inches. After hot-winding, the spring was hardened by heating to 1450° F. and suddenly quenching in cold water. Tempering was done by removing the quench and reheating to 600° F., holding at this temperature for one hour, and then cooling in air to atmospheric temperature. This complied with recommendations of the Joseph T. Ryerson Company from whom the steel was purchased. A sample piece of the hardened and tempered spring was tested on a Rockwell Hardness Tester and gave the figures shown in Table 1. A spring of different properties can be obtained by tempering at a different temperature. Also, further variations can be obtained by using a steel of different composition.

| Trial No. | Hardness Reading Rockwell "C" Scale | Tensile Strength (18) (Ultimate) in psi for Hardness Observed |
|:---:|:---:|:---:|
| 1 | 52 | 262,000 |
| 2 | 54 | 278,000 |
| 3 | 54 | 278,000 |
| 4 | 50 | 245,000 |
| 5 | 53 | 269,000 |
| 6 | 52 | 262,000 |

The following calculations indicate that the working stress of the designed spring is 75,000 psi.

Maximum moment required to deflect the spring by 15 degrees = 2,800 lb-in. (previous calculations)

Stress in the spring when 2,800 lb-in. is applied is:

$$S = \frac{M \times C \times K}{I}$$

Where S = stress in pounds per square inch

C = distance of remotest fiber from neutral axis = 0.375 in.

I = moment of inertia in in.$^4$

= 0.01553 in.$^4$ for a ¾-inch circular section

K = stress concentration factor = 1.11 for a $\frac{D}{d}$ ratio of 8

Where D = mean coil diameter of spring

d = diameter of spring wire

By substitution

$$S = \frac{2,800 \times .375 \times 1.11}{.01553}$$

= 75,100 psi.

The tensile strength of the spring as observed by hardness testing was found to be satisfactory. The original design was based on the yield strength of the spring being 150,000 psi.

# Static and Dynamic Load Testing

### STATIC LOAD TESTING

The relationship between torque and dial gauge readings was obtained by static tests. These were conducted by clamping the input shaft of the dynamometer in the chuck of a lathe. The chuck was held stationary by engaging both the back gear and direct drive. Torque was applied by placing a series of known weights on a lever fastened to the

11

output shaft. For torque applied, the corresponding dial gage readings were made. Before taking the dial gage reading, the sliding collar was tapped lightly to eliminate static friction beween the collar and the output shaft, and to insure that the collar was as near as possible to an equilibrium position. The average of dial gage readings for the same value of torque applied (Table 2) was plotted against torque to obtain the calibration curve in Figure 10.

### Table 2. Static Load Test Results (Dial Gage Reading for Torque Applied)

| Torque Applied in ft-lbs. | Dial Gage Reading in Thousandths of an Inch | | | | | |
|---|---|---|---|---|---|---|
| | Test 1 | Test 2 | Test 3 | Test 4 | Test 5 | Average |
| 10 | * | * | * | * | * | * |
| 19 | * | * | * | * | * | * |
| 30.4 | 32 | 28 | 31 | 31 | 30 | 30 |
| 38.4 | 65 | 61 | 63 | 64 | 64 | 64 |
| 47.9 | 106 | 105 | 103 | 106 | 103 | 104 |
| 56.5 | 150 | 152 | 145 | 145 | 143 | 147 |
| 64.6 | 187 | 194 | 187 | 185 | 183 | 187 |
| 73.6 | 229 | 229 | 224 | 226 | 225 | 227 |
| 82.6 | 265 | 263 | 266 | 268 | 264 | 265 |
| 90.2 | 303 | 297 | 305 | 307 | 307 | 304 |
| 98.8 | 343 | 342 | 345 | 346 | 345 | 344 |
| 107.3 | 383 | 387 | 386 | 387 | 384 | 385 |
| 115.9 | 423 | 421 | 424 | 426 | 425 | 423 |
| 123.8 | 462 | 462 | 461 | 465 | 463 | 463 |
| 132.5 | 501 | 501 | 499 | 503 | 501 | 501 |
| 141.0 | 540 | 540 | 540 | 544 | 539 | 541 |
| 149.5 | 578 | 582 | 579 | 583 | 579 | 580 |
| 160.5 | 625 | 625 | 624 | 628 | 625 | 625 |
| 170.0 | 669 | 667 | 668 | 671 | 669 | 669 |
| 178.6 | 705 | 707 | 707 | 705 | 707 | 706 |
| 197.6 | 792 | 792 | 790 | 790 | 790 | 791 |

*Exceeded low range of dynamometer.

### Dynamic Load Testing

A General Electric Cradled Dynamometer was used for dynamic load tests. The torsion dynamometer mounted on a tractor was connected with a shaft and universal joints to the electric dynamometer. The tractor transmitted power through the torsion dynamometer with the electric dynamometer serving as the load. By adjusting the throttle of the tractor with the governor disconnected and by operating the controls of the electric dynamometer, torques of different magnitudes were transmitted. Dial gage and electric dynamometer scale readings (Table 3) for different throttle openings were taken with the RPM being held at 550.

Horsepower indicated by the electric dynamometer for different dynamometer scale readings were calculated from the formula:

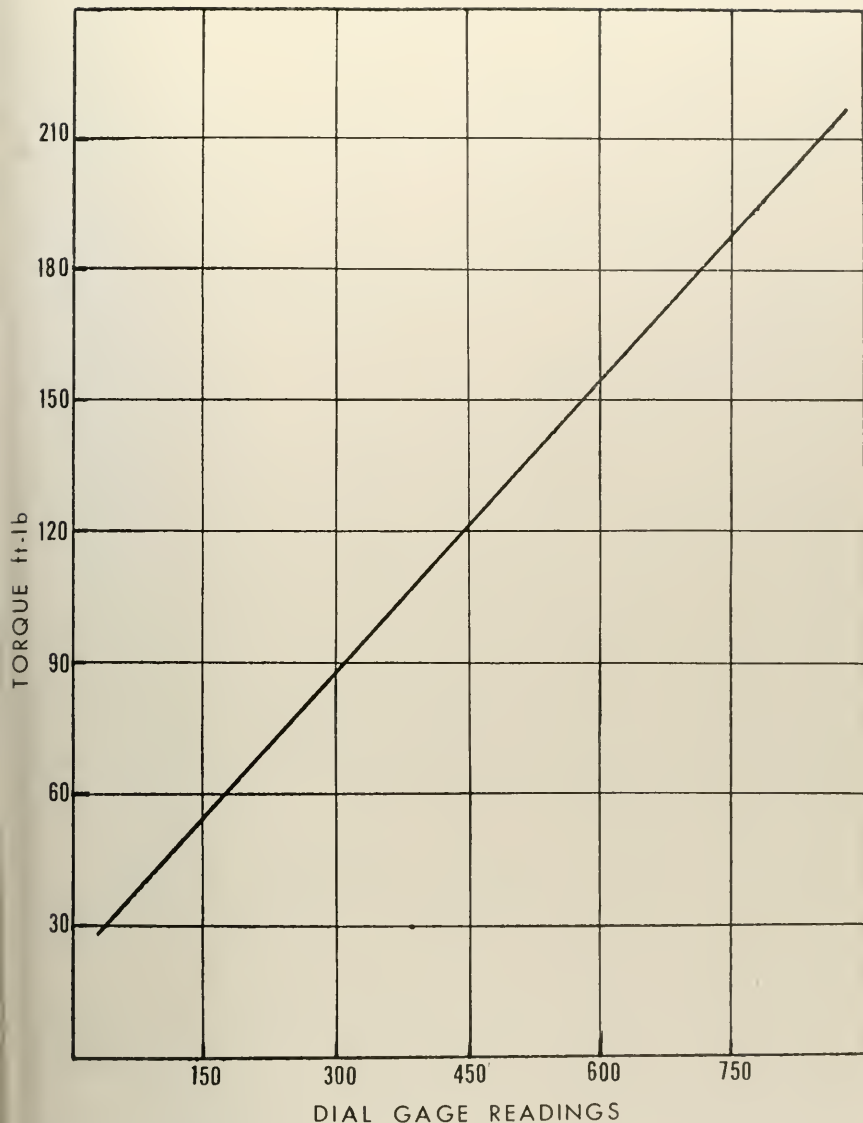FIGURE 10. Static calibration curve of the torsion dynamometer.

$$\text{hp} = \text{FN} \times \frac{1}{3,500}$$

Where F = force in pounds from dynamometer scale reading

N = revolutions per minute

$\frac{1}{3,500}$ = constant of the electric dynamometer used

13

TABLE 3. RESULTS OF DYNAMIC LOAD TESTING 550 RPM

| ELECTRIC DYNAMOMETER SCALE READINGS POUNDS | DIAL GAGE READINGS | TORQUE IN FT-LB FROM GAGE READING AND FIG. 10 | HP FROM TORSION DYNAMOMETER | H.P. FROM ELECTRIC DYNAMOMETER | DIFFERENCE IN HP |
|---|---|---|---|---|---|
| 22.5* | 05* | 25.5* | 2.67* | 3.54* | * |
| 32.5 | 120 | 50.0 | 5.23 | 5.11 | +.12 |
| 41.4 | 180 | 63.3 | 6.62 | 6.52 | +.10 |
| 42.5 | 190 | 65.5 | 6.85 | 6.68 | +.17 |
| 52.5 | 250 | 78.5 | 8.21 | 8.24 | +.03 |
| 56.5 | 270 | 83.0 | 8.68 | 8.88 | —20 |
| 60.5 | 315 | 92.5 | 9.67 | 9.50 | +.17 |
| 67.5 | 370 | 105.0 | 10.98 | 10.61 | +.37 |
| 70.5 | 380 | 107.0 | 11.19 | 11.08 | +.11 |
| 74.5 | 400 | 111.5 | 11.66 | 11.70 | —.04 |
| 84.5 | 465 | 126.0 | 13.17 | 13.27 | —.10 |
| 87.5 | 480 | 128.5 | 13.44 | 13.74 | —.30 |
| 93.5 | 540 | 142.0 | 14.85 | 14.69 | +.16 |
| 94.5 | 540 | 142.0 | 14.85 | 14.85 | +.00 |
| 102.5 | 600 | 155.0 | 16.21 | 16.11 | +.10 |
| 108.5 | 635 | 163.0 | 17.05 | 17.05 | +.00 |
| 112.5 | 670 | 168.3 | 17.60 | 17.67 | +.07 |
| 115.5 | 690 | 175.0 | 18.30 | 18.15 | +.16 |
| 117.5 | 700 | 177.0 | 18.51 | 18.46 | +.05 |
| 122.5 | 730 | 183.5 | 19.20 | 19.25 | —.05 |

*Below limits of dynamometer.

To determine the power transmitted through the torsion dynamometer, torque values for dial gage readings were taken from the calibration curve (Figure 10). When the torque was known, power transmitted was calculated from the formula:

$$hp = \frac{2 \pi NT}{33,000}$$

Where T = torque in ft.-lbs. from static calibration curve

N = revolutions per minute

## DISCUSSION OF TEST RESULTS

Static load tests indicated that torque loads with a maximum error of ± 1 ft-lb. could be determined. Variation in dial gage readings for the same torque was probably due to friction between the sliding collar and the shaft and in the bearings of the input and output shafts.

The tractor to which the dynamometer was mounted for dynamic testing transmitted vibrations to the dynamometer. The vibrations of the system under dynamic tests tended to reduce the effects of friction. Power variations of the tractor pulsated the dial gage indicator, making it slightly difficult to secure precise readings. The small difference between the calculated horsepower (Column 6, Table 3) from the torsion and electric dynamometer indicates that the pulsations of the dial gage indicator were of no serious consequence.
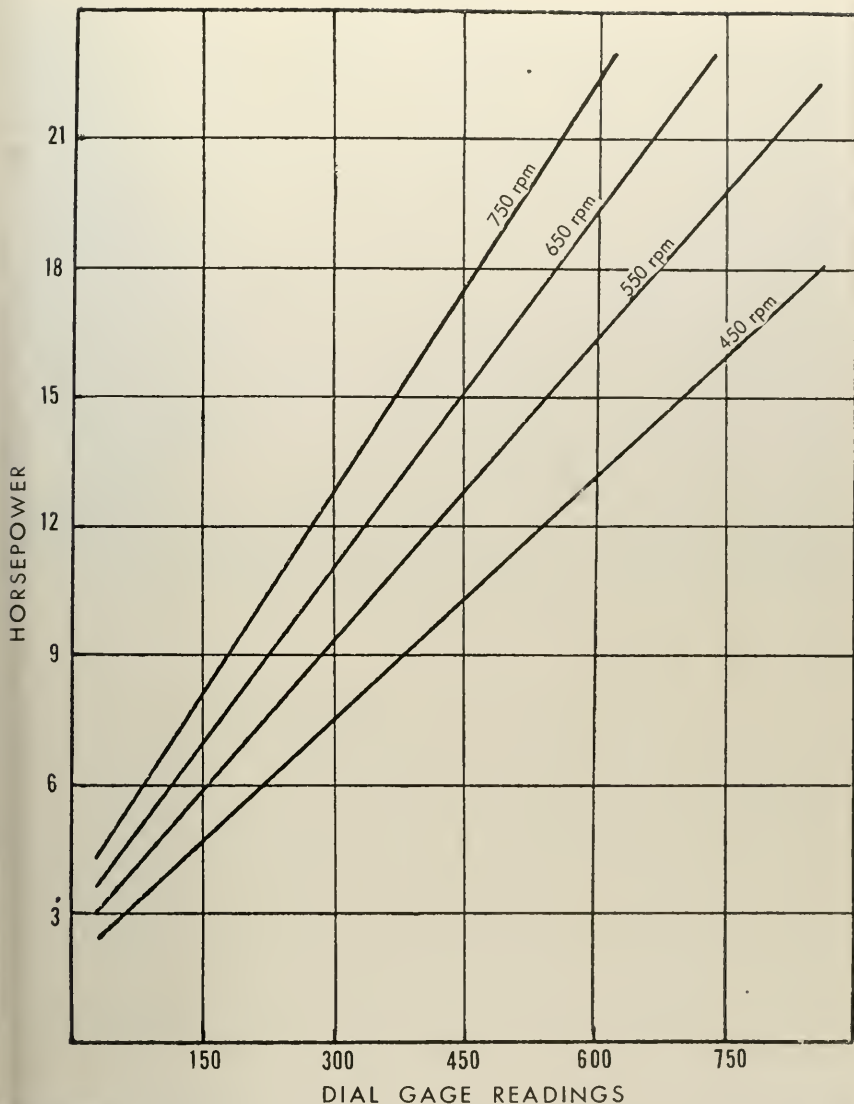
FIGURE 11. Dynamic calibration curve of the torsion dynamometer (horse-power is calculated from static torque reading in Fig. 10).

No dampening device was used either between the driver and driven shafts or between the sliding collar and the dial gage. Hence, the dynamometer was very sensitive to torque changes.

The probable error in reading the two dynamometers determined statistically was 0.11 horsepower. The average difference of 0.04 horse-power would result as a matter of chance at slightly above the 50 per

FIGURE 12. The PTO dynamometer mounted on a tractor and coupled to a power-take-off drive.

cent level. This indicates that there was no significant difference in the power determined from the two dynamometers. The probable error of reading the torsion dynamometer is less than $\frac{1}{2}$ of 1 per cent of maximum power that could be transmitted.

The dynamometer spring was pre-loaded by tightening the spring with nuts against the shaft arm. The pre-load eliminated any play between the spring ends and the shaft arms to which the spring was connected. The pre-load on the spring can be avoided by having collars on the spring ends. The output and input shaft arms can then be tightened against the collars to form a rigid unit. For simplicity in construction, collars were not provided on the spring. Due to this pre-load, the spring did not deflect until a torque of 23.5 ft-lb. was applied. The relationship between torque and deflection, as obtained from Figure 10, shows that for every foot-pound of torque applied, 4.56 thousandths of an inch was indicated on the dial gage.

Horsepower can be calculated by substituting values of torque taken from corresponding values of gage readings and speed of the shaft in the formula:

$$ \text{hp} = 1.90 \times 10^{-4} \frac{N\ (C - 107)}{4.56} $$

This formula was obtained from the fundamental formula:

16

$$hp = \frac{2 \pi NT}{33,000}$$

Torque T as obtained from a gage reading is:

$$T = \frac{K + C}{4.56} \text{ foot-pounds}$$

Where C is the dial gage reading in thousandths of an inch and K is the constant to correct for the pre-load in the spring. Pre-load in the spring was 23.5 lb-ft. or equal to .107 of an in. on the dial gage.

By substituting values for $\pi$ and T, the equation becomes:

$$hp = 4.168 \times 10^{-5} CN$$

The constant (4.168) in the above equation can be made a multiple of 10 or any other desirable constant by making the appropriate changes in the lever arm connecting the sliding collar and dial gage.

# Summary and Results

Objectives were to design, build, and test a torsion dynamometer with the following features:

1. Suitable for testing torsional power requirements of mounted as well as drawn power-take-off driven machines.
2. Compact.
3. Light in weight (32 lb.).
4. Simplicity of design.
5. Reasonable accuracy.

The dynamometer is based on the principle that a torsional load applied to a helical spring will cause deflection proportional to the torque. This deflection, when transmitted to an indicating dial on a stationary member, can be used to determine torque.

A design of the various component parts of the dynamometer was made. Construction was accomplished in the Agricultural Engineering Laboratory. Component parts of standard design were used when they could be readily adapted. Suitable grades of steel and other materials were obtained to make the remaining component parts.

Testing of the torsion dynamometer consisted of two phases: static load testing and dynamic load testing. Static tests were performed by rigidly clamping the input shaft and by applying weights on a lever arm connected to the output shaft. Dial gage readings for different torques gave the relationship of 4.56 thousandths of an inch for one ft-lb. of torque.

Dynamic load tests were performed by connecting the torsion dynamometer between the power-take-off of a tractor and a cradled electric dynamometer. Horsepower calculated from the torsion dynamometer

at different loads was compared to that taken from the electric dynamometer. There was no significant difference in the power calculated from readings on the two pieces of equipment. Actual differences indicated a probable error of $\frac{1}{2}$ of 1 per cent of the maximum capacity of the torsion dynamometer.

FIELD TESTS

Field use of the dynamometer was made during the harvest. A mower, hay crusher, and rotary forage harvesters were checked for power consumed under various conditions.* Results of duplicate power tests indicated that the accuracy of the dynamometer was quite favorable.

The sensitivity of the indicator was too great to be practical with the fluctuating loads encountered. This can readily be corrected by changing the dimensions of the linkage. A recording mechanism in place of the indicator would also be an improvement. The recorder would show maximum and minimum power used and furnish a means of more accurately determining the average power.

*Performance of Forage Crushers by P. John Zachariah. K. C. Elliott, and R. A. Phillips, West Virginia University Agricultural Experiment Station Bulletin (in print), Morgantown, West Virginia.

# APPENDIX

## Development of Power-Take-Off Shafts
## and Torsion Dynamometers
# Review of Literature

DEVELOPMENT OF POWER-TAKE-OFF SHAFTS

The versatility of a tractor in supplying power to stationary as well as to hitched equipment was increased by the incorporation of power-take-off shafts. This development began a few years after some progress had been made in the use of tractors for draft purposes. The term "power-take-off" refers to the mechanical means of transmitting power in the form of rotary motion, without the use of a belt, from a tractor engine to a machine unit which may be attached directly to the tractor or coupled and pulled behind the tractor (32).

In the early stages of development, some manufacturers were equipping machines with individual power units. Combined harvester-threshers were used with steam tractors in 1904 and 1905 (32). A steam engine was mounted on the harvester-thresher, just as internal combustion engines are presently mounted, but this engine was operated by steam brought from the steam tractor boiler through a hose. In a broad sense, this constituted a power-take-off. In 1906, Gougis, in France developed a power-take-off to replace the bull wheel drive on a grain binder (8). Benjamin of the International Harvester Company is credited with developing the first power-take-off in this country.

A good percentage of manufacturers produced machines which were operated through ground wheel drives. Ground wheel drives were very popular, but their efficiency was 50 per cent or less. Eighty per cent or higher efficiency could be obtained by applying power through a power-take-off (32). This improved drive was adapted to combines, balers, field forage choppers, mowers, and other machines of this general type. Though the use of power-take-off was demonstrated as early as 1878 (17) at a universal exposition held at Paris, it was about 1923 before tractor manufacturers offered the power-take-off as an accessory or as a built-in unit.

The need of transmitting power even when the driven machine is stationary, and the possibility of eliminating built-in weights of both tractor and drawn machine encouraged manufacturers to make power-take-off driven machines instead of the ground wheel drive type. As the power-take-off became a common feature of all tractors, it was necessary for the manufacturers of machinery to standardize the power-take-off to

facilitate interchangeability of both tractor and driven implement. Tractor power-take-off specifications were first published in the August 15, 1929, issue of *Farm Implement News*. Later, W. L. Zink (33) presented a paper at the Chicago meeting of the ASAE making recommendations on:

1. Size and type of power-take-off shaft ends.
2. Method of retaining the front fitting to the spline shaft.
3. Power-take-off speed and rotation.
4. The power-take-off shaft location with respect to height.
5. Center line and position relative to point of hitch.
6. Various safety features.

In order that any power-take-off machine can be quickly connected to any make of tractor without the necessity of special equipment, the ASAE has published recommendations for the standardization of power-take-off drives (1). The torsion dynamometer developed at West Virginia University conforms to these power-take-off specifications.

DYNAMOMETERS DEVELOPED IN THE PAST

Dynamometers, as their name implies, are instruments for measuring power. They can be divided into two main classes—traction dynamometers and trosion dynamometers.

Traction dynamometers are intended to measure the power produced by a direct pull or thrust. An example would be power required of a tractor to pull a trailer or that required of a team of horses to pull an implement. These dynamometers consist of some kind of weighing device (16) together with equipment to measure the rate of travel. The weighing device may be a spring, a hydraulic cylinder and piston, or some other method by which the force or pull exerted is measured. A speedometer or a stop watch in conjunction with a measured distance could be used to determine rate.

Torsion dynamometers can be classified into two groups: the absorption type and the transmission type. In an absorption dynamometer, the power to be measured is converted into some form of energy, usually heat, and dissipated. This type of dynamometer can be used to test only equipment which develops power. Transmission dynamometers (16) are those that are designed to measure the power transmitted from one rotating shaft to another, either directly or indirectly through a belt gearing, or other suitable mechanism.

In 1836, a form of transmission dynamometer was introduced into this country by Bachelder of Sanco, Maine (9). Hopkins was one of the early users of transmission dynamometers (9). The dynamometer designed by Hopkins was used for tests with Siemen's Dynamo Electric Machines. The principle used was the weighing of resulting stress from

a deflected belt, and by this means ascertaining the direct stress in the belt. From rate of travel and force transmitted through the belt, horsepower could be calculated.

Another form of dynamometer was one in which the difference in tensions of the tight and slack side of the belt was arranged to move a system of levers. A dynamometer of this kind was designed by Tatham (9) of Philadelphia for use in the Franklin Institute.

The Tatham Dynamometer (Figure I) consists of a lever arm and weights to balance the torque transmitted from one shaft to another. The torque on the transmission shaft of the Tatham Dynamometer was equal to the difference in tension in the belt $T_1 - T_2$ multiplied by the radius of the pulley, (r). Neglecting the friction on the two idler pulleys P and $P_1$, their reactions at their bearings will be $2T_1$ and $2T_2$ respectively. Taking moments about their fulcrum (f), then:

$$WR + 2T_2a = 2T_1a$$

from which $T_1 - T_2 = \dfrac{WR}{2a}$

and torque $= r(T_1 - T_2) = \dfrac{rWR}{2a}$

the horsepower transmitted $= \dfrac{rWR \times 2\pi N}{2a \times 33,000}$

where N is the revolutions per minute of the transmission pulley, and units of r, R and a, are in feet, and W in pounds.

### The Power-Take-Off Dynamometer Developed by McCall*

McCall (20) built a recording transmission dynamometer in 1940 at Ohio State University. Its operating principle was similar to the dynamometer built by Tatham. This dynamometer consists of a driving shaft B, and a driven shaft A, connected by means of sprockets and a roller chain. The chain passes over two idler sprockets C and D mounted on a balance arm frame which is pivoted at A. The balance arm also contains a counter balance weight and a connecting link to the piston as shown in Figure II. When power is transmitted from shaft A to shaft B through the roller chain, the force on the connecting link acts against the hydrostatic piston and causes pressure to be built up in the hydrostatic unit. The pressure is transmitted through an oil line to a Gulley recording unit where it is recorded. By using calibration factors, horsepower transmitted is calculated.

The design of this power-take-off dynamometer was such that it required a trailer to transport the unit. The complete dynamometer
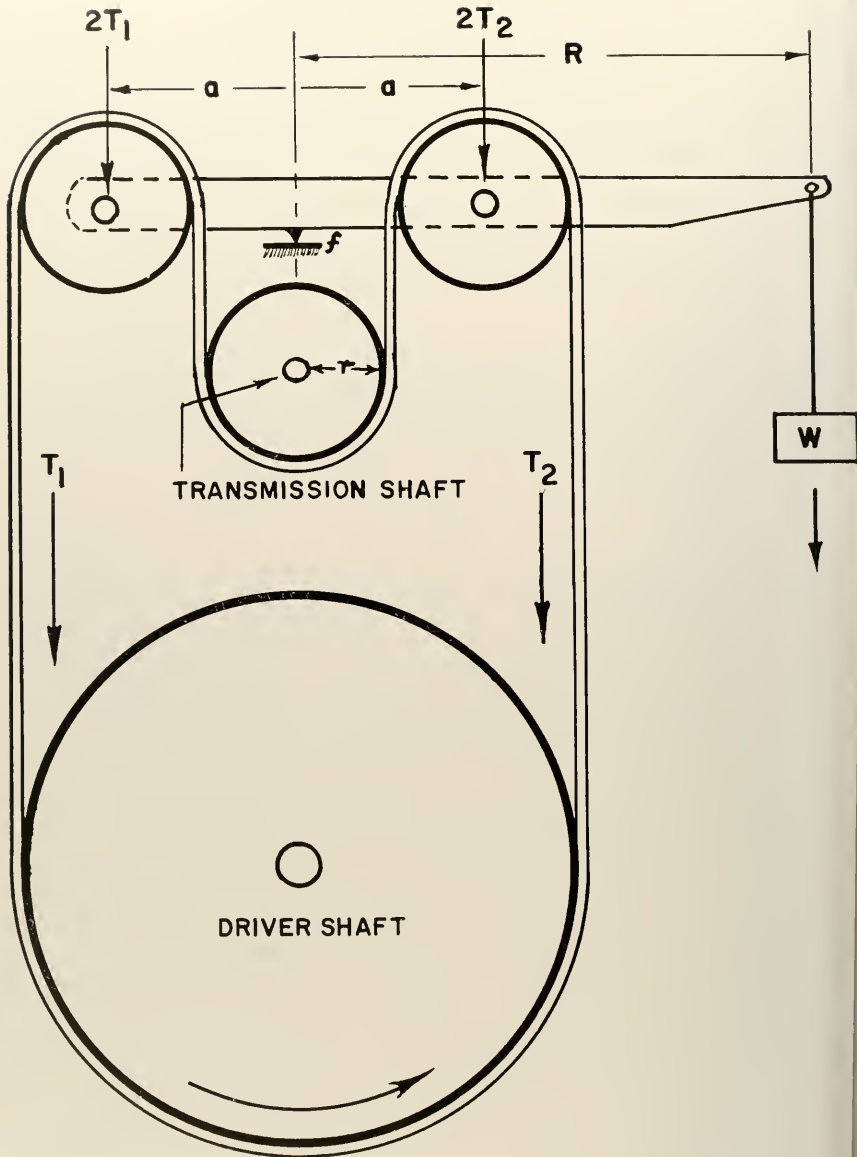
*See Figure II.

FIGURE I. Belt dynamometer.

without operator weighed 1,000 pounds. As the unit was mounted in
trailer, it could not be used for testing driven machines which wer
mounted on tractors. The normal hitch position of a trailing implemer
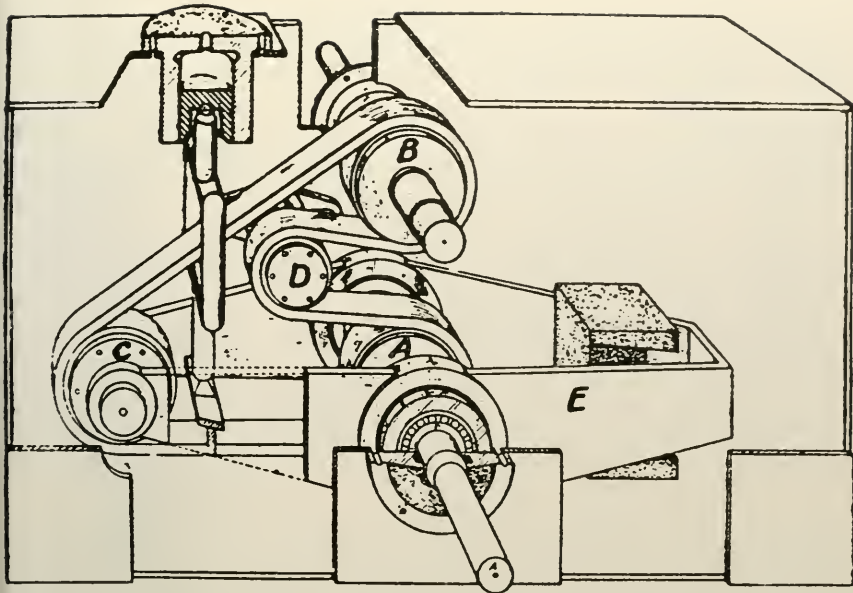to a tractor would be disrupted during tests with this unit.

22

FIGURE II. Working parts of the power-take-off dynamometer developed by McCall. (In the actual machine a ¾-in. pitch double width roller chain was used.)

## CRADLE DYNAMOMETER

A widely used form of transmission dynamometer is the cradle type. In it, the power unit is usually an electric motor or an internal combustion engine. The motor or engine is mounted in a cradle, and the system balanced on its centroidal axis. To give free rocking action when power is transmitted, the power unit is usually supported on ball bearings. The engine or electric motor is mounted so that the axis of the driving shaft of the power unit and the axis of the cradle bearings coincide.

When power is transmitted to the machine which is to be tested, the cradle tends to rotate on the centroidal axis, but it is prevented from rotating by means of a lever arm. Some form of force-measuring device is used at the end of the lever arm. From the force necessary to hold the system in balance, the length of lever arm and the speed of the power unit, the horsepower required to operate the driven machine can be determined.

$$\text{Horsepower} = \frac{F \times 2\pi\, LN}{33,000}$$

Where    F = Force in pounds required to balance the system

       L = Length of lever arm in feet

       N = Revolutions per minute

In 1922-23 McCuen (21) developed a recording cradle-type dynamometer in which a Midwest engine was used as the prime mover. The engine was mounted on a frame with trunnion ends to give free swinging action on the ball bearings. To measure torque, a lever arm was connected to the engine housing with the other end of the lever arm connected to a hydrostatic coupling which transmitted oil pressure to a recording mechanism.

Cradle-type dynamometers can be used for measuring power produced by engines and motors as well as for determining power required to operate driven machines. The wide range of power that can be measured with this dynamometer, and the ease with which it can be calibrated, constitute its advantages. However, its excessive weight, size, and the need for a suitable belt or coupling limits its use to the testing of stationary machines. This would not be a feasible unit for testing power-take-off machines under field conditions.

## Shaft-Type Torsion Dynamometer

When a torsional load is applied to a shaft, an angular twist or deflection proportional to the moment is produced. By measuring this twist in a calibrated rotating shaft and the speed at which the shaft rotates, power transmitted through the shaft can be determined. Devices used for measuring the twist or deflection can be mechanical, optical, or electrical (6). Hiren (16) is credited with making the first dynamometer of this kind. When the modulus of elasticity of the shaft material is known, and the angle of twist and speed of shaft measured, power transmitted can be mathematically determined as follows:

$$hp = \frac{2\,\pi\,N\,\Theta^*\,G\,J}{12 \times 33{,}000\,L} = \frac{2\,\pi\,N\,M}{12 \times 33{,}000}$$

J = polar moment of inertia of the shaft section in inches⁴

L = length of shaft in inches

Θ = angle of twist of shaft in radians in length L of the shaft

N = revolutions per minute of the shaft

G = modulus of elasticity of the shaft material

M = moment in inch-pounds

The Central Laboratory Company (7), in the first quarter of this century, used electrical means to measure torque in shafts. The dynamometer they designed consisted of two shafts connected by a helical spring (Figure III). Discs fitted to each shaft, and insulated from the shaft except for a narrow radial portion, served as commutators. The two brushes when contacting the radial portion of the discs simultaneously,

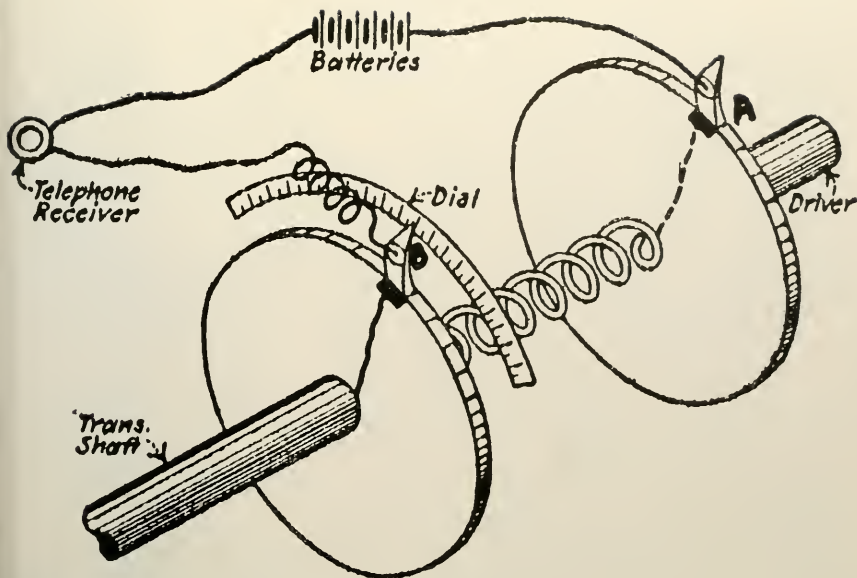*Conventional symbol was not available.

24

FIGURE III. Torsion dynamometer with electrical device to measure angle of twist. (When no torque was transmitted, brushes A and B were adjusted to hear clicks in the telephone receiver. Upon transmission of power, angular lag of the transmission shaft with respect to the driven shaft was measured by adjusting brush B along the calibrated scale, till a click was heard. The scale reading indicated the torque transmitted.)

completed the electrical circuit, causing a click in the telephone receiver. Brush A was fixed, and brush B could be moved along a dial.

Before torque was transmitted, the brush B was adjusted so that a click was heard, and the dial set at zero. As torque was transmitted through the system, due to deflection of the spring, the disc under brush B would have an angular lag with respect to the disc under brush A. To measure the torque, brush B was adjusted until a click was heard, indicating that the contact pieces were passing at the same time below the brushes. The angular adjustment of brush B, as shown on the calibrated dial, indicated the torque delivered.

USDA Recording Torquemeter

In 1953, Reed (24) and Berry of the United States Department of Agriculture (USDA) developed a recording torquemeter. The design was such that torque or rotation force was transformed into a force parallel to the driver and driven shafts, where it was resisted by liquid filled bellows. Pressure of the liquid in the bellows was proportional to the force acting or torque, and the power transmitted could be calculated from the RPM and pressure readings.
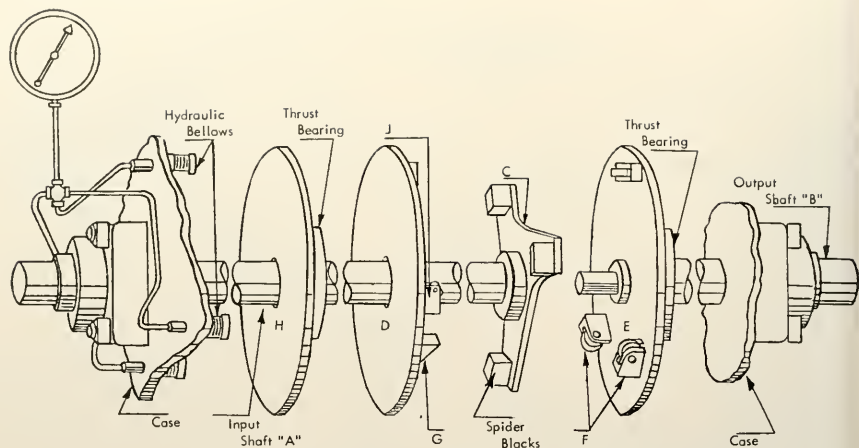
**FIGURE IV.** Diagrammatic sketch of USDA torquemeter showing relationship of working parts.

This torquemeter consists of two shafts, an input shaft A, and an output shaft B (Figure IV). A plate E, having three rollers F, is rigidly connected to shaft B. The input shaft has a rigidly fixed spider plate C which turns a transmission plate D having three inclined blocks. The transmission plate operates against the thrust-bearing of the nonrotating plate H. The spider plate rotates the transmission plate J by contact. The three inclined blocks G on the transmission plate are in contact with rollers F on plate E. When torque is transmitted through the inclined blocks and rollers, and axial force equal to the torque is set up. This force, acting on plate D, is transmitted through a thrust-bearing to the nonrotating plate H. Movement of the nonrotating plate is resisted by three hydraulic bellows, each having a capacity of 373 pounds. These bellows are connected by tubing to a pressure indicator or recorder. Torque can be obtained from the pressure readings, and when the speed of the shaft is known, power being transmitted can be determined. The torquemeter is relatively simple and can be used to test mounted PTO driven machines. Its weight is 150 pounds.

STRAIN-GAGE TECHNIQUE OF MEASURING TORQUE

Use of strain-gages for measuring torque transmitted through a shaft is relatively new. Within the last few years, strain-gage technique of measuring torque has become popular. These torque measurements have been used for power calculations.

The strain gage is a fine piece of resistance wire mounted in an insulating medium and bonded to the part to be tested. Gages are so connected that changes in the resistance of the gage wire, due to change
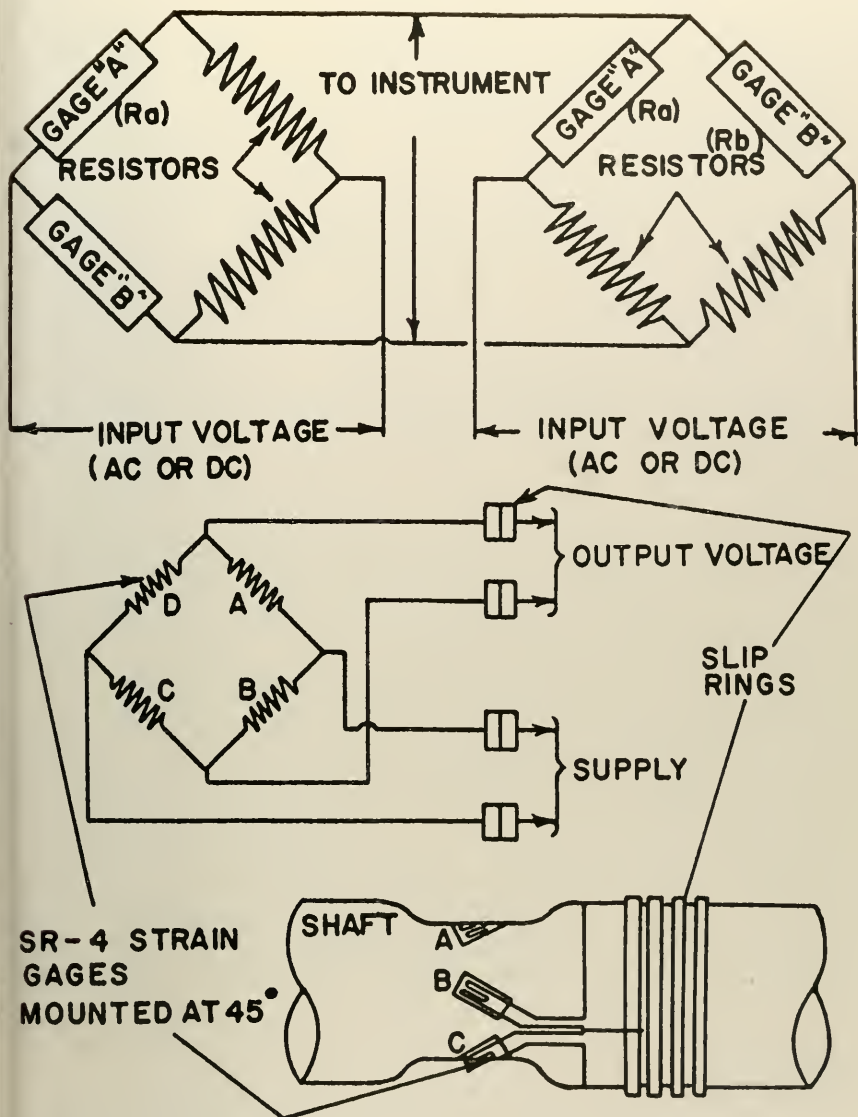
FIGURE V. Strain-gage connections in a SR-4 torque pick-up.

in dimension of the wire caused by the strain induced, can be read or recorded through suitable equipment. As strain is a straight-line function of stress (within the elastic limit) resistance of the strain gage can be mathematically converted into stress.

Different types of gage connections are used for measuring strain. Baldwin-Lima-Hamilton Corporation manufactures torquemeters cap-

able of measuring from 10 inch-ounces to 750,000 foot-pounds (27). Their SR-4 Torquemeter consists of a torque pick-up and an oscillograph or oscilloscope. The pick-up provides an electrical signal which is proportional to the torque. The instrument may be used to translate the signal into indicated or recorded units of torque. The pick-up is made of an elastic member, usually in the form of a short length of shaft, to which a group of special SR-4 strain gages are bonded and connected to form a Wheatstone bridge circuit (Figure V). On transmission of torque through the shaft, a change in strain causes a corresponding change in the resistance of the gages. The wire gages bonded to the shaft are in such a position and are so connected into a bridge circuit as to cancel any possible bending or thrust strains. Slip rings on the shaft and a nonrotating brush assembly permit the strain gage bridge to be energized and its unbalance measured by any suitable indicating or recording instrument.

When the gages and instruments used are in good condition, they will give accurate results. Heth (12) has mentioned that it is difficult to get satisfactory results when gages are used outdoors under unfavorable weather conditions, such as extreme humidity and very low temperatures. During the last five to six years, to eliminate the effects of moisture, temperature and humidity variance, considerable improvements have been made in the design and application of gages.

Gage mechanisms and associated indicating and recording instruments are very delicate for use under field conditions. If accurate results are to be obtained, the equipment must be handled by engineers especially trained in its use. Above all, it is not a low-cost unit. A torque pick-up unit alone, having a capacity of 2,000 in-lb., costs $1,175 (28) and a direct reading indicator costs $900. An installation with a recording device and suitable accessories would cost approximately $10,000. In view of the high cost and delicate mechanisms involved strain gages are not extensively used for power measurement of power take-off driven machines.

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