A COMBINED DRAWBAR PIN AND FORCE TRANSDUCER

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This paper describes the design and performance of a drawbar pin which also serves as a force transducer. The transducer consists of a split, elongated outer ring and an inner ring on which are mounted a total of eight strain gages. The pin transducer produces a linear output independent of the load application point within the tractor drawbar clevis. The unit has been successfully used for a number of field tests on four-wheel drive tractors.

INTRODUCTION

In field testing of agricultural tillage equipment it is necessary to measure the implement draft from which the power requirements can be determined. The usual procedure is to measure the tractor pull or draft by inserting a dynamometer (force transducer) between the implement and the tractor hitch points. Draft is the horizontal component of pull in the direction of travel.

Early drawbar dynamometers of the spring-type were difficult to read due to rapid fluctuations of the undamped pointer. Hydraulic dynamometers which consist of a cylinder-piston system connected to a Bourdon tube gage may be damped considerably by placing a restriction valve in the line to the gage. Because of general acceptance by tractor manufacturers, this type of dynamometer is still used to measure draft in the drawbar tests at the University of Nebraska. In recent years there has been an increased use of strain gage dynamometers of various configurations. These units, when operated with electronic readout instruments, can provide multiple ranges and avoid the oil-leakage problem inherent in hydraulic dynamometers. Clyde (1955) described a “homemade” strain gage dynamometer with associated circuitry. Harrison and Reed (1961) have described drawbar draft transducers with strain gages applied to ring (o-type) members. The ring transducer was incorporated into a special fixture which would allow measurement of only the draft component of pull. The fixture was rigidly mounted on the end of the tractor drawbar. A strain gage dynamometer which measured the draft component of pull was developed by Zoerb (1963). By exciting the strain-gage bridge circuit with the voltage output from a ground-driven tachometer generator, this dynamometer read drawbar horsepower directly.

The major objection to placing a dynamometer fixture at the end of the tractor drawbar is that the tractor-implement hitch-point configuration is altered. The horizontal and vertical angles of pull are altered with the insertion of the dynamometer between the tractor and the implement. Integral drawbar force transducers have been built by Jensen (1954) and by Zoerb and Howse (1976). An improved version of the latter dynamometer has been described by Musonda and Bigsby (1982). While the integral dynamometer has the advantage of not changing tractor-implement hitch configuration (since no apparatus is required between the tractor and implement), a disadvantage is that of not being “portable” from one tractor to another. The integral dynamometer becomes a permanent part of a given tractor.

This paper describes the design of a drawbar pin which also serves as a force transducer. Thus, the unit overcomes the drawback of the built-in integral dynamometer because the pin can be used with any tractor. In addition the pin dynamometer does not in any way change the normal tractor-implement hitch configuration.

CONSTRUCTION DETAILS

The objective of this project was to design and construct a pin force transducer which could be used as the drawbar pin for four-wheel drive tractors. Measurements were taken of the drawbar clevis (hitch points) of seven four-wheel drive tractors. The principal dimension restriction was pin diameter. To fit all tractors, the drawbar pin transducer must not be more than 38 mm in diameter. The various dimensions of the hitch clevises dictated a pin length of at least 190 mm.

Basically the pin transducer consists of an inner ring and a split outer ring. The inner ring represents the sensing member with attached strain gages. The outside diameter of this ring is 20.0 mm and the inside diameter is 14.0 mm. The purpose of the outer ring is to protect the gaged inner ring and to provide sufficient strength to resist bending and shear loads. In operation, the implement and tractor hitch members compress the outer ring halves against the inner ring. Thus the inner ring is placed in compression when a drawbar load is applied. Compressive stresses are developed on the outer surface of the ring in the plane parallel to the direction of the load, while tensile stresses are developed in a plane at right angles to the load direction. A photograph of the transducer disassembled, and before strain gages were mounted, is shown in Fig. 1. A cross-section of the transducer is given in Fig. 2.

A few special precautions were necessary in the design and construction of the unit. For example, it was necessary to keep the inner ring oriented in a given direction with respect to the load. The small diameter pin (Fig. 1) keeps the inner ring located within the outer ring halves. One-half of the outer ring was fastened to the upper plate by cap screws. During field tests, the torque produced on the pin while turning and pulling a large load sheared these cap screws. This half of the outer ring was then welded to the upper plate. The upper plate with the angle iron brackets ensured that the drawbar transducer was kept aligned with respect to load direction. The width between the angle brackets was adjustable to accommodate drawbars of various widths. The lower hole accommodated a large cotter pin, above which one or more flat washers could be inserted to take up vertical slack.

As a result the top plate of the pin assembly would be held in contact with the drawbar under dynamic or jerky load conditions. Figure 1 shows four recesses milled into the outer ring halves to provide clearance for the four strain gages mounted on the inner ring in the plane of load application. As indicated in Fig. 3, four gages (Nos. 3, 4, 5, 6) were mounted on the center portion of the pin and at right
Figure 1. Transducer before assembly and installation of strain gages.

angles to the load direction. From Fig. 2 it can be seen that ample clearance space was provided for these gages on either side of the inner ring. These annular clearances also provided a passage for the gage lead wires up to the Cannon socket. Figure 4 shows the pin transducer with spacer washers. In Fig. 5, the transducer is shown connected to the drawbar of a four-wheel drive tractor. For calibration purposes a strain-gage drawbar dynamometer was connected in series with the pin force transducer. The pin was then calibrated against the precalibrated dynamometer. This arrangement is shown in Fig. 6.

Figure 2. Transducer cross-section.

ANALYSIS OF LOAD SENSING RING

The analysis of circular rings in terms of deflection and stress has been presented by several authors. In most textbooks specific deflection and strain formulas are given for circular rings and in some cases also for octagonal and extended rings. Hoag and Yoerger (1975) have given a complete mathematical development of equations for displacement and moment at any section for the simple ring and the extended ring. Their analysis considers the general case where a ring section is subjected to normal force, shear force and a bending moment.

The following equations for the simple ring are based on the ring being subjected to a compressive load only. Based on Castigliano’s theorem and referring to Fig. 7, Seely and Smith (1952) have developed the following equation for the bending moment at any section in the ring:

\[ M = -0.318FR + \left( \frac{FR}{2 \cos \theta} \right) \]  

(1)

Figure 3. Physical and electrical locations of strain gages.
where $M$ = bending moment (N·m)
$F$ = applied load (N)
$R$ = mean radius of ring (mm)

Thus, the maximum bending moment occurs when $\theta = 90^\circ$, that is, at the points of application of the load.

At $90^\circ$ to the direction of load application ($\theta = 0^\circ$), the bending moment is:

$$M_0 = 0.182FR$$  \hspace{1cm} (3)

Thus, the moment on the ring will vary from $0.182FR$ at $\theta = 0^\circ$, to $-0.318FR$ at $\theta = 90^\circ$. Note that when $\theta = 50.5^\circ$, the moment is zero.

An analysis of the circular ring by Cook and Rabinowicz (1963) indicates a ring deflection of:

$$\delta = 1.79FR^2/EBt^3$$  \hspace{1cm} (4)

where $\delta$ = ring deflection (mm)
$b$ = axial length of the ring (190 mm).
$E$ = modulus of elasticity (GPa)
$t$ = ring thickness (mm)

The strain in a “thin” ring is:

$$\varepsilon = M/El = 6M/EBt^2$$  \hspace{1cm} (5)

The steel used for the construction of this pin transducer is classed as AISI C1144. The yield strength is 690 MPa (100 000 psi) as received. The pin transducer was expected to handle a drawbar load of 60 kN. Strain gages used were Micro Measurements, type EA-06-090DH-350.

**STRESS CALCULATIONS**

1. **Outer Shell**

   Although the inner ring can take a relatively small portion of the load due to bending, it was assumed that all bending stress would be taken by the outer shell. Inside and outside diameters of 25 mm and 38 mm, respectively, were selected, to provide a moment of inertia of $83.2 \times 10^3$ mm$^4$. The maximum bending moment would occur if the implement tongue applied the load at the midpoint between the tractor hitch clevis. Normally the implement tongue will rest on the lower clevis plate where the moment would be relatively small. Assuming a clevis span of 100 mm and point loading, the maximum possible moment is:

$$M_{max} = 30 \times 10^3 \times \frac{100}{2} = 1500 \text{ N·m}$$

then

$$\sigma_{max} = \frac{Mc}{I} = \frac{1.50 \times 10^3 \times 19}{83.2 \times 10^3} = 342.5 \text{ MPa}$$

As a result of the segment of the outer shell which is in contact with the inner ring, the actual moment of inertia is larger than that calculated above. In practice the
point loading would not be achieved. These two factors ensure that the actual stress developed under maximum load would be considerably less than 342.5 MPa. The maximum shear stress in the outer shell for a 60-kN load was computed as 37 MPa.

2. Inner Ring
An outer diameter of 20 mm was selected. Thus the lateral clearance between the inner ring and the outer shell (except at the contact points) is 2.5 mm (see Fig. 2). An inner diameter of 14 mm was chosen giving a wall thickness, $t$, of 3 mm. From Eq. 5,

$$F_{\max} = \frac{e_{\max} Eb^2}{1.91R} = \frac{\sigma_{\max} b^2}{1.91R} = \frac{690 \times 190 \times (3)^2}{1.91 \times 8.5} = 72.7 \text{ kN}$$

This value represents a factor of safety of 72.7/60 = 1.2 based on yield strength, which may be low for cases where impact loads are encountered. A greater factor of safety would be achieved by decreasing the inner diameter (increasing the ring thickness).

The inner ring was also checked for compressive stress in the walls, and for contact stress (Faires 1965). These values were calculated to be 50 MPa and 70 MPa, respectively.

The deflection of the inner ring was determined from Eq. 4:

$$\delta = \frac{1.79FR^3}{Ebt^3} = \frac{1.79 \times 60 \times 0 \times (8.5)^3}{206.8 \times 10^3 \times 190 \times (3.0)^3} = 0.062 \text{ mm}$$

This calculation indicated that clearance between the outer shell halves had to be greater than this value.

CALIBRATION
As mentioned above, the equipment shown in Fig. 6 was used to calibrate the drawbar pin transducer. A precalibrated strain gage dynamometer was connected in series with the pin transducer and these units were chained to a post. The tractor was placed in gear and eased forward to apply a load. Only a maximum of 41.4 kN (9300 lb) could be applied in this manner. The results of the calibration tests are shown in Fig. 8. In addition to checking the linearity of the pin transducer, the effect of various vertical load positions was investigated. In the first test (position A), the member which simulated the implement tongue, rested on the lower plate of the tractor hitch clevis. This tongue member was approximately 25 mm thick. Thus, the point of load application was considered to be 12.5 mm above the lower clevis plate. Similarly, loads were applied for positions B and C. In position B, the center of the tongue was 37 mm above the lower plate; in position C, the center of the load was 53 mm above the lower plate. Equations for output versus pull were determined for positions A, B and C. Statistically there was no significant difference in the relationships for the A, B and C positions. Therefore all points for the three positions were combined to give the straight line shown in Fig. 7. This line has a slope of 16.27 $\mu$E/kN and a correlation coefficient of 0.998.

This analysis illustrates a unique design feature of the transducer, namely, that the electrical output is independent of the point of load application within the tractor clevis. The physical locations of the strain gages on the inner ring and the electrical connections of the eight strain gages into the bridge circuit account for this independent relationship. For loads producing horizontally angled pulls with a swinging drawbar, the output of the transducer would have to be multiplied by the cosine of the angle to obtain draft. Probably the most common type of load for four-wheel drive tractors is that from a heavy-duty cultivator where the straight, horizontal pull is also draft.

During calibration the strain indicator reading was in microstrain (Fig. 8). It is interesting to compare the strain indicated by the instrument with the theoretical values from Eqs. 5 and 6. From Fig. 8, the indicated average strain was 488 $\mu$E for a load of 30 kN. On the ring, four gages are subjected to strains as given by Eq. 5 ($\theta = 90^\circ$) and four are subjected to strains as given by Eq. 6 ($\theta = 0^\circ$). The negative and positive values are additive in the bridge so that the average strain per bridge arm is

$$\frac{1.91 + 1.09 + 1.91 + 1.09}{4} = 1.5 \frac{FR}{Ebt^2}$$
Figure 8. Pin transducer calibration. Points A, B and C represent the vertical position of the center of the implement tongue above the tractor lower clevis plate; A = 12.5 mm, B = 37 mm and C = 53 mm.

The bridge circuit averages the load. Since a total of eight strain gages are used, with two connected in series in each bridge arm, the effective load is $F/2$ in the above case.

The average strain for a 30 kN load is therefore:

$$\varepsilon = \frac{1.5 \times 15000 \times 8.5}{206.85 \times 10^3 \times 190 \times (3)^2} = 541 \mu \varepsilon$$

This value checks well with the indicated strain of $488 \mu \varepsilon$ when one considers that the strain gages have a finite gage length of 2.3 mm. In other words, the strain given by Eqs. 5 and 6 represents the strain at the gage midpoints. The strain at the gage ends is less than that at the midpoint due to the physical location on the ring. Thus, one would expect the indicated strain ($488 \mu \varepsilon$) to be less than the calculated value ($541 \mu \varepsilon$).

This pin transducer has been used for a number of field tests on four-wheel drive tractors. To provide the necessary ruggedness for four-wheel drive tractors, other drawbar transducers used to date have been relatively massive, difficult to handle, and have of course altered the tractor-implement hitch point configuration. This pin transducer simply replaces the normal drawbar pin that is required.

CONCLUSIONS

1. The pin transducer produced a linear output with drawbar load and the output was independent of load application position in the vertical plane.

2. The measured strain values compared favorably with the theoretical values as given by the equations in the paper.

3. The pin transducer serves as both a drawbar hitch pin and as a force transducer without changing the tractor-implement hitch configuration.

REFERENCES


