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Design, Development and Calibration of a Drive Wheel Torque Transducer for an Agricultural Tractor

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ABSTRAK

Satu unit penderia sepadan untuk mengukur daya kilas pada roda pacuan sebuah traktor telah direka bentuk, dibangunkan serta ditentukur. Unit tersebut mempunyai reka bentuk satu aci penyambung yang dipasang di antara bebibir gandar roda belakang dan rim gandar roda belakang traktor. Tolok tarikan jenis rintangan dilekatkan pada permukaan lilitan aci serta dijambat dalam litar Wheatstone mengikut konfigurasi piawai bagi pengukuran magnitud daya kilas. Litar jambatan pada setiap sisi aci pemacu belakang disambungkan ke sistem perolehan data dalam traktor melalui gelang terbelah pada hujung aci pemacu. Setiap unit penderia telah direka bentuk untuk julat daya kilas bersamaan 0 hingga 32 kNm dengan kepekaan bersamaan 14.492 (tarikan/kNm. Ujian penentukuran statik ke atas unit penderia menunjukkan kelelurusan dan kejituan pengukuran yang sangat baik dengan masing-masing mempunyai pekali sekaitan bersamaan 0.9994 dan 0.9994. Histerisis unit penderia bagi pengukuran statik didapati tidak bererti. Ralat pengukuran unit penderia tidak melebihi 0.20% daripada magnitud daya kilas untuk pengukuran statik dan 0.1% untuk pengukuran dinamik rangsangan. Ujian demonstrasi di ladang menunjukkan pengukuran bagi magnitud daya kilas pada roda pacuan belakang traktor adalah pada tahap yang sangat memuaskan. Sistem perolehan data berjaya mengimbas serta merekod isyarat yang diterimanya daripada kedua-dua unit penderia. Unit penderia ini merupakan sebahagian daripada sistem peralatan yang lengkap dalam traktor Massey Ferguson 3060 yang akan digunakan untuk pembentukan satu pangkalan data yang lengkap tentang keperluan kuasa dan tenaga bagi traktor-peralatan yang bekerja di ladang.

ABSTRACT

An appropriate transducer unit for measuring the drive wheel torque of a tractor was designed, developed and calibrated. The unit adopts a design having an extension shaft mounted in between the rear wheel axle flange and rear wheel rim of a tractor. Resistance type strain gauges were bonded on the shaft circumferential surfaces into a Wheatstone bridge circuitry for a standard torque measurement configuration. The bridge circuitry on each side of the rear drive axle was interfaced to a data acquisition system on board a tractor via a slip ring at the drive shaft end. Each transducer unit has been designed for a torque range of 0 to 32 kNm with a sensitivity of 14.492 μStrain/kNm. Static calibration tests on the transducer revealed excellent measurement linearity and measurement accuracy with coefficients of correlation or R^2 equal to 0.9994 and 0.9994, respectively. The transducer hysteresis for static measurement was not significant. The transducer measurement errors were not more than 0.20% and 1% of the measured torque magnitudes under static and simulated dynamic measurements, respectively. Field demonstration test showed that the measurement for the torque magnitude at the rear wheel of the tractor was satisfactory. The data acquisition system had successfully scanned and recorded the signals from both transducers. These transducer units are part of the complete instrumentation system in the Massey Ferguson 3060 tractor to be used in the generation of comprehensive database on the power and energy requirements of the tractor-implement working in the field.

Keywords: Tractor, instrumentation, transducer, wheel torque

INTRODUCTION

Increasing populations demand more and more power and energy in agriculture to produce food and fibres. Inefficient tractor operations increase the cost of agricultural production. The need to maintain their profitability have emphasized requirements to maximize the field operation efficiency. Selection and matching of tractors to implement any specific field operation have been largely based on prior experience rather than the understanding of the involved performance factors. Therefore, information on tractor-implement field performance is required for their efficient and effective use.

Information on tractor-implement field performance in ASAE standard and published literature is based on cropping practices that differ greatly from those in Malaysia. A regional agricultural machinery management database is badly needed to help planners, managers and engineers in making decisions concerning the management of the agricultural field machinery. It is therefore necessary to have detailed information on power demand and energy requirements of any machinery utilized in the locality area.

Measurements of wheel torque have received considerable attention in tractive performance studies. Instrumentation that measures the wheel torque and angular velocity would enable the tractive of the drive wheel to be investigated. The common method for wheel torque measurements is strain gauges with a slip ring mounted at the outer end of the axle to stationary recording equipment (Tompkin and Wilhelm 1982; McLaughlin *et al.* 1993; El-Janobi *et al.* 1997). The other method, telemetry is that of actually transmitting the strain gauge signal through the use of radio-frequency transmitters mounted on shaft and picking up the signal by means of a receiver placed nearby (Palmer 1985; Watts and Longstaff 1989; Synder and Buck 1990).

The Department of Biological and Agricultural Engineering, Universiti Putra Malaysia has initiated a research work to develop an instrumentation and data acquisition system on-board a Massey Ferguson 3060 tractor for comprehensive information on the performance of the tractor and its working implement in the field. The intended system would have the capacity of measuring and monitoring engine speed, PTO speed, forward speed, drive wheel slippage, acres worked, fuel consumption per hour, fuel consumption per hectare, acres worked, cost factor, fuel consumed, fuel remaining, and distance, also at the same time have the capacity of measuring and recording horizontal pull at the tractor drawbar, wheel torque at both tractor rear wheels, PTO torque at the tractor PTO output, and both horizontal and vertical forces at the point hitches (Kheiralla and Azmi 2001). Extensive field testing will be conducted with this instrumented tractor on the aspects of power demand and energy requirements of various agricultural field operations in Malaysia. Ultimately, the obtained data would be processed, analyzed, and transformed into agricultural machinery management data for Malaysia.

This paper describes the design, development, calibration and demonstration of one of the available transducers, namely driver wheel torque transducer.

MATERIALS AND METHODS

General Description

The design of wheel torque transducer is based on an extension shaft that is mounted between wheel axle flange and tire rim of a tractor (*Fig. 1*). A RBE-4A Kyowa slip ring and a special made adapter are fitted to one end of the extension shaft. Two sets of KFG-5-120-D16-11-L1M-2S Kyowa, 90° rossette, 120 Ohm, 2.1 gauge factor strain gauges are bonded at 45° shear planes into a into full bridge configuration on opposite sides of the extension shaft. A 2.5 mA constant current excitation source is supplied from the data acquisition system to each wheel torque transducer via their slip rings. Two L shaped steel conduits are mounted on each side of the tractor mudguards to carry the cables from slip rings to the data acquisition system inside the tractor cab. *Fig. 2* shows the arrangement of wheel torque transducers on the drive axle shaft. The block diagram shown in *Fig. 3* illustrates the complete system.

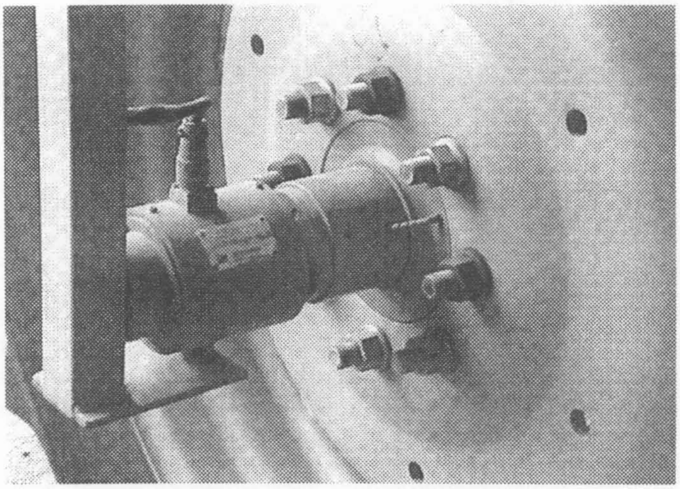

Fig. 1: Wheel torque transducer

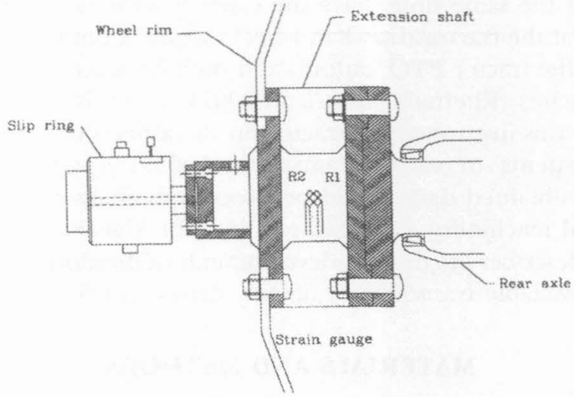

Fig. 2: The arrangement of wheel torque transducer on drive axle shaft

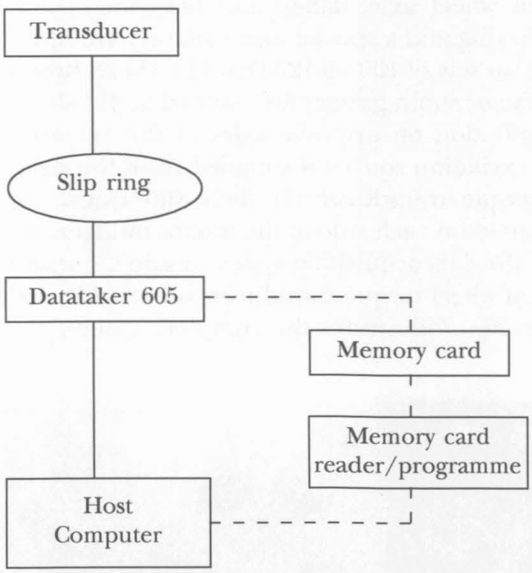

Fig. 3: Block diagram of wheel torque transducer measurement

Extension Shaft Design

The difficulties faced in mounting strain gauges on the rear drive axle to measure the direct torque led to the adoption of an extension shaft. The installed extension shaft on the drive axle should be able to support the tractor drive wheel torque, dynamic loading and bending moment at the same time sensitive to shear strain.

The expected wheel torque, which the transducer was designed to measure, was estimated from tractor rated engine power. A Massey Ferguson 3060 tractor having an engine power of 64 kW and a rated engine speed of 2200 rpm would produce flywheel torque of 277.8 Nm. There would be only a slight lost of transmitted power from the engine to drive wheels for the mechanical rotating system. The torque magnitudes at drive wheels are equal for a tractor traveling straight on uniform surfaces. Therefore, the torque magnitude at each drive wheel is given as (Goering 1990):

$$T_w = \frac{\eta}{2}\left(\frac{N_e}{N_w}\right)T_e \tag{1}$$

where T_w is the drive wheel torque in Nm, T_e is the flywheel torque in Nm, N_w is the drive wheel speed in rpm, Ne is the engine speed in rpm and η is transmission efficiency in percentage. The wheel torque magnitude for a tractor traveling at lowest gear (i.e. 2.3 km/hr and running at rated engine speed of 2200 rpm) is equal to 32.26 kNm.

The significance of bending is an important criteria in the design of the extension shaft. The dynamic load at the drive wheel is made up of the static load and the weight transfer to the wheel as a result of mounted implement on the tractor. The weight transfer on the tractor is given as (Liljedal *et al.* 1989):

$$\Delta R_r = \frac{P.h}{L_1} \tag{2}$$

where P is the maximum drawbar pull in kN, h the drawbar height in m and L_1 the tractor wheelbase in m. The weight transfer magnitude is 8.63 kN at a maximum drawbar pull of 50 kN for the specified tractor having a drawbar height of 0.41 m and wheelbase of 2.38 m. Knowing the weight transfer magnitude of 8.63 and the rear wheel static load of 23.343 kN, gives each wheel a dynamic load of 15.987 kN for a simple moment calculation. Thus, computed maximum bending moment on the extension shaft is 14.228 kNm for a load of 15.987 kN at a span of 1.78 m.

The shaft diameter design equation for applied torque and bending moment of under static loading condition is described as (Hindhede 1987):

$$D = \left(\frac{32 f_s}{\pi S_{yt}}\sqrt{\left(\frac{S_{yt}}{S_e}M\right)^2 + \frac{3}{4}T^2}\right)^{\frac{1}{3}} \tag{3}$$

where D is the shaft diameter in mm, T is the applied torque in kNm, M is the bending moment in kNm, S_{yt} is the yield tensile strength in MN/m^2 and S_e is the endurance strength in MN/m^2 and f_s is the dimensionless factor of safety.

The material used for the shaft is mild steel SAE 1020 having a yield strength of 300 MN/m^2. Based on shaft material strength, practical constraints and tractor requirements, an extension shaft having 100 mm diameter and 120 mm length with two flanges at both ends was designed as a compromise between transducer strength and sensitivity. A special adapter having 88 mm external diameter and 60 mm length was made and fitted at one end of the extension shaft to hold the slip ring. The final extension shaft design is shown in *Fig. 4*.

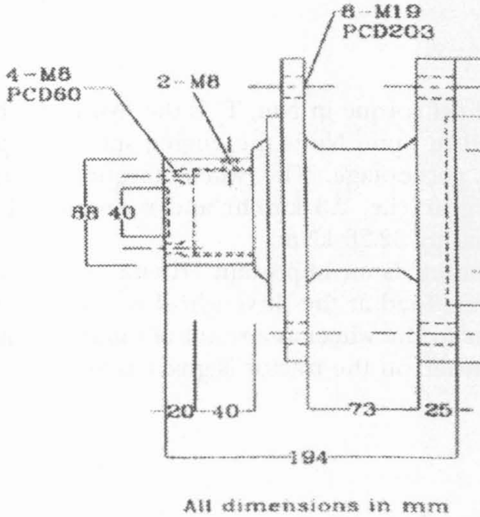

Fig. 4: Dimensions of the extension shaft and adapter

The strain due to torque loading at 45° location to the shaft axis is given as (Timoshenko and Gere 1990):

$$\varepsilon_{45^\circ} = \frac{8T}{\pi D^3 G} \tag{4}$$

where ε_{45° is the output strain in μStrain, T is the applied torque in Nm, D is the shaft diameter in mm and G the modulus of rigidity in GPa. Knowing the shaft diameter of 100 mm, torque magnitude of 32260 Nm and shaft modulus of rigidity of 80 GPa, the maximum strain ε_{45° was calculated to be 981μStrain.

Strain Gauges Configuration

The wheel torque sensing element design had 2 sets of Kyowa KFG-5-120-D16-11-L1M-2S, 90° rossette strain gauges installed on opposite sides of the extension axle shaft and positioned at 45° to axle axis to give the maximum sensitivity to torsional strains. The selected gauges have α gauge resistance of 120 Ohm, α gauge factor of 2.1, and one-meter length lead wires. The available 4 gauges are wired to a constant current full Wheatstone bridge configuration. A full bridge 2.5 mA constant current supply from data acquisition system is used to excite the strain gauges bridge on the wheel torque transducer.

The gauge installation was carried out after smoothing the extension shaft surface by fine silicon carbide abrasive paper. Accurate markings for the gauge locations on the shaft were made possible with the aid of a Digimetic Height gauge. The strain gauges were positioned at 180° apart on the predetermined surface of the shaft and glued permanently using an epoxy adhesive. Finally, SG280 protective coating was applied to the installed gauges to provide protection against water, humidity and mechanical abrasion.

The sensitivity of the transducer is expressed in terms of bridge output strain per unit torque applied. The predicted output strain for the wheel torque transducer circuit under design torque is calculated to be 981μStrain. Since the output of constant current for the data acquisition system is measured in μStrain, the predicted channel sensitivity is calculated to be 30.41 μ Strain/ kNm.

Natural Frequency and Dynamic Response

The designed transducer should be able to measure dynamic mode response of the tractor wheel torque. The dynamic response of wheel torque transducer and tractor drive shaft can be modeled as torsional vibration of two mass-shaft. The model used to simulate a tractor and wheel torque transducer is shown in *Fig. 5.* The applied torque at the shaft end causes the axle and transducer to vibrate and the complete transducer unit to respond to torsional oscillatory motion (Timoshenko *et al.* 1990). In order for the recorded torque to be influenced by any vibration motion, its natural frequency should be larger than the frequency of exciting vibrations. The natural frequency of transducer and drive axle shaft can be expressed as (Liljedal *et al.* 1989):

$$f = \frac{1}{2\pi}\sqrt{\left(k_t \frac{J_1 + J_2}{J_1 J_2}\right)} \tag{5}$$

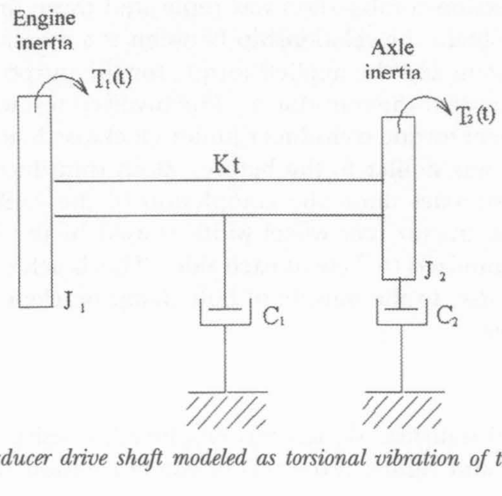

Fig. 5: Transducer drive shaft modeled as torsional vibration of two mass shafts

where f is the natural frequency in Hz, J_i is the effective torsional inertia at engine in kgm^2, J^2 is equivalent torsional inertia at the axle shaft converted from vehicle mass and rolling resistance and is given as $J_2 = mr^2$ in kgm^2, and k_t the stiffness of the drive shaft given as $k_t = \pi GD^4/32L$ in Nm/rad.

A medium-sized tractor based from Crolla, (1978) could develop an effective torsional inetria of 2 kgm^2 at its engine. Knowing the tractor mass of 3898 kg, rolling radius of 0.79 m, axle shaft diameter of 82 mm, shaft length of 0.70 m and shaft modulus of rigidity of 80 GPa, the calculated axle inertia and stiffness were respectively 2495 kgm^2 and 507279 Nm/rad. Then, the natural frequency of the transducer and tractor drive shaft system was calculated to be approximately 80 Hz.

Calibration

Comprehensive static calibration tests were carried out to determine the measurement linearity between applied torque with output strain and measurement accuracy between applied torque and measured torque. The test was conducted by fixing the transducer between the rim the concave disk and tractor rear axle simulator. A rectangular bar was bolted horizontally to wheel rim concave disc. The whole assembly was mounted to a loading frame that is equipped with manual hydraulic power pack. A load cell that was located in between the piston end of the hydraulic cylinder of the power pack and the free of the rectangular bar to measure the applied torsional force. A known torque was applied to the transducer and shaft simulator by manually operating the hydraulic pack on the loading frame. A data acquisition system running on DeCipher plus command program was used to scan and record the output strain from the drive wheel torque transducer. The test was conducted under the loading range from 0 to 5.0 kNm at 0.5 kNm intervals. Each drive wheel transducer was calibrated under two loading directions (i.e. clockwise and anticlockwise) and two loading conditions (i.e. increasing and decreasing). Each measurement at any applied load was repeated four times and each test at any loading direction-combination was replicated three times. A similar test was conducted to obtain the relationship between the measured torque by the data acquisition system and the applied torque for the purpose of checking the measurement accuracy of the transducer. The involved test was only conducted to the left drive wheel torque transducer under clockwise loading direction and the test procedure was similar to the before. Both transducers were mounted to tractor rear drive axles upon the completion of the calibration test. The displacement of the tractor rear wheel width caused by the installation of the transducers was minimized to 7 cm at each side. This is achieved by bolting the wheel rim concave disc to the outside of bolt clamp brackets on the wheel rim for both rear wheels.

Statistical Analysis

A four way factorial statistical design was employed, consisting of two sides of the wheel (i.e. left and right), two levels of wheel rotation (i.e. clockwise and anticlockwise), two modes of loading (i.e. loading and unloading), eleven levels of applied torque and four replications; resulting in a total of 352 tests. For simplicity, the following linear additive model was utilized (Steel and Torrie 1996):

$$Y_{ijklm} = \mu + \alpha_i + \beta_j + \gamma_k + \delta_l + (\alpha\beta)_{ij} + (\alpha\gamma)_{ik} + (\alpha\delta)_{il} + (\beta\gamma)_{jk} + (\beta\delta)_{jl} + (\gamma\delta)_{kl} + (\alpha\beta\gamma)_{ijk} + (\alpha\beta\delta)_{ijl} + (\alpha\gamma\delta)_{ikl} + (\beta\gamma\delta)_{jkl} + (\alpha\beta\gamma\delta)_{ijkl} + e_{ijklm} \quad (6)$$

where Y_{ijklm} is the output strain made in ith level of wheel, jth level of rotation, kth level of loading, lth level of applied torque, and in the mth replications, μ is the mean, α_i, β_j, γ_k and δ_l are the main effects of wheel, rotation, loading and applied torque, respectively, $(\alpha\beta)_{ij}$, $(\alpha\gamma)_{ik}$,................. $(\beta\gamma\delta)_{jkl}$ and $(\alpha\beta\gamma\delta)_{ijkl}$ are first and higher order interaction effects of wheel, rotation, loading and applied torque, and ε_{ijklm} is the random error.

The General Linear Model procedure in PC SAS software package (SAS 1996) was employed to determine the relationship of output measured strain with applied torque.

Demonstration Test

Field demonstration trials on complete instrumentation and data acquisition system were conducted for verifying wheel torque measurements of the Massey Ferguson 3060 tractor. The tractor was used to pull at its drawbar a FIAT 540 tractor under brake on a dry asphalt surface at an average traveled speed of 2 km/hr (*Fig. 6*). During the test, the tractor differential lock was engaged to ensure equal torque and travel reduction at each wheel. The instrumentation system and data acquisition system on the tractor was set to run at sampling means at one-second intervals. The collected data during field demonstration was downloaded into the host computer with the aid of Datataker 605 memory card reader for further analysis.

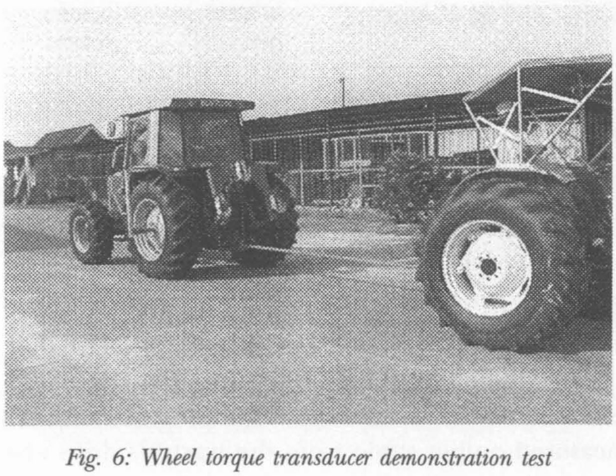

Fig. 6: Wheel torque transducer demonstration test

RESULTS AND DISCUSSIONS

ANOVA on the calibration for wheel torque transducer in Table 1 shows that the entire tested main effects and interactions with the exception of applied torque had no significant (P>0.05) effects on measured output strain. The insignificance of rotation implies no differences on the measured output strain between clockwise and anticlockwise wheel rotation. The insignificance of loading indicates no hysteresis effect on the measured output strain under loading and unloading modes. Again, the insignificance of wheel explains similar measured output strain from both wheel sides. All these obtained findings supported the use of one calibration equation for wheel torque transducer measurement.

TABLE 1
ANOVA on the calibration for wheel torque transducer

Source of variation	SS	MS	F-value
Replication	4.8933	1.6311	2.25NS
Wheel	12.5340	12.5340	17.32NS
Rotation	0.4305	0.4305	0.60NS
Loading	0.0707	0.0707	0.10NS
Torque	179916.7807	179916.7807	24867.65**
Wheel (Rotation	0.0041	0.0041	0.01NS
Wheel (Loading	3.2359	3.2359	4.47NS
Wheel (Torque	14.3287	14.3287	1.98NS
Rotation (Loading	0.0001	0.0001	0.00NS
Rotation (Torque	5.2415	0.5241	0.72NS
Loading(Torque	18.4511	1.8451	2.55NS
Wheel(Rotation(Loading	1.4114	1.4114	1.95NS
Wheel(Rotation(Torque	6.2343	6.2343	0.86NS
Wheel (Loading(Torque	3.4713	0.3471	0.48NS
Rotation(Loading(Torque	1.9759	0.1975	0.27NS
Wheel (Rotation(Loading(Torque	3.5036	0.3503	0.48NS
Error	188.8328	0.7235	

** Significant at 1% probability level
NS Not Significant

The plotted calibration graph in *Fig.* 7 shows that applied torque and measured output strain were highly correlated. The linearity equation is expressed by:

$$S = 14.492T \quad \text{with} \quad R^2 = 0.9994 \tag{7}$$

where S represents the measured output strain in μStrain and T is the applied torque in kNm. The equation was used in the programming of Datataker 605 to read the measured output strain from the tractor's drive wheels in kNm.

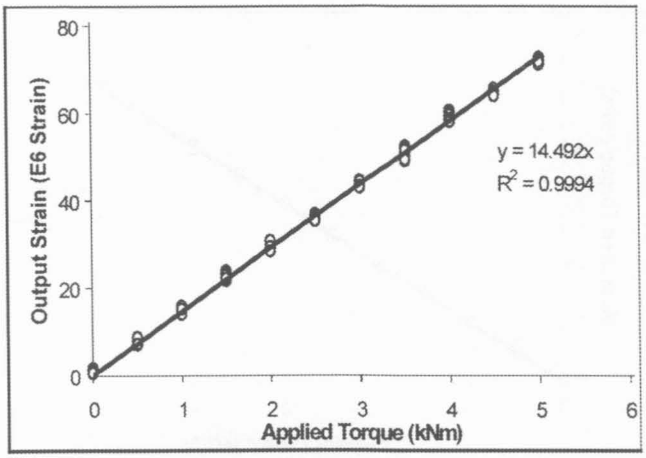

Fig. 7: Calibration curve for wheel torque transducer

The measured strain gauge sensitivity was 14.492 μStrain/kNm. This value was 0.45 times lower than earlier computed theoretical sensitivity (i.e 30.41μStrain/kNm). This difference was due to gain multiplier effect that was set automatically during autoranging by the Datataker 605.

The plotted measurement accuracy graph in *Fig. 8* shows a high degree of linearity between applied torque and measured torque. Their relationship is best expressed by the following formula:

$$T_M = 0.998T \quad \text{with} \quad R^2 = 0.9994 \qquad (8)$$

where T_M represents the measured output torque in kNm and T is the applied torque in kNm. The transducer is rated to give a measurement accuracy within ± 0.20% range. This factor was used for computing and documenting the measured output torque by the transducer.

The natural frequency of wheel torque transducer based on torsional vibration of two mass shaft systems was estimated to be 80 Hz while the working frequency of a typical tractor operating in the field as indicated by Claar *et al.* (1982) and Erickson and Larsen (1983) was around 2 Hz. Fortunately, with this frequency ratio of less than 0.1 (i.e. 0.025 to be exact) resulted with a transmissibility value of less than 1.01 and torque measurement error of no more than 1% of the excitation torque (Timoshenko *et al.* 1990). Consequently, the conducted static calibration on the transducer was acceptable for dynamic torque measurement. Besides that, it also ensured that the dynamic torque measured by the transducer would not be attenuated or distorted.

Fig. 9 indicates a sample data of the measured wheel torque at the tractor left drive wheel. The computed 95% confidence interval for the measured torque was found to be 3.46 ± 0.63 kNm. The dynamic variability was reasonably synchronized with the sampling frequency pattern of instrumentation and data

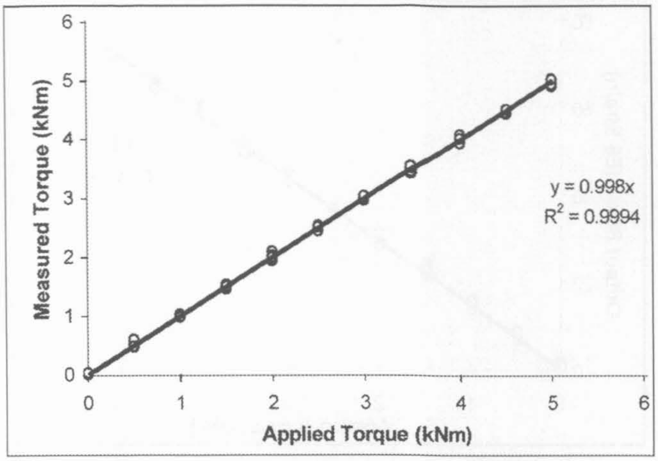

Fig. 8: Verification curve for measurement accuracy of wheel torque transducer

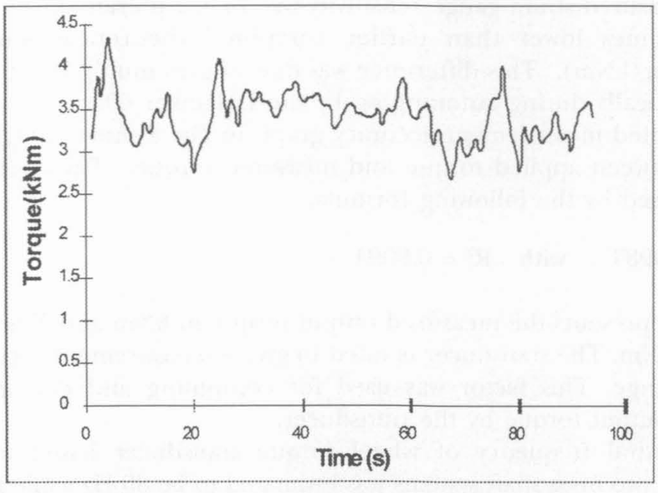

Fig. 9: Torque measurements at left rear drive wheel

acquisition system. The instrumentation and data acquisition system was able to scan the transducer signals without difficulties. The stored data in the memory card of the Datataker 605 was downloaded and recovered into the hard disk of the computer at the end of the trials.

CONCLUSION

A wheel torque transducer was designed and developed to measure the tractor's rear wheel torque output up to 32 kNm at a sensitivity of 14.492 μStrain/kNm. The transducer unit adopts a design having extension shaft mounted between wheel axle flange and wheel rim. Static calibration of the transducer revealed excellent linearity between applied torque with output strain with coefficient of correlation equal to 0.9994 and excellent accuracy between applied load with measured torque with coefficient of correlation equal to 0.9994. The transducer's measured sensitivity was favorably comparable to its computed theoretical sensitivity. Natural frequency of the transducer is estimated to be 80 Hz by modeling the complete system as a torsional vibration of two mass-shafts. The static and dynamic responses of the transducer are within the acceptable error limit. Measurements field demonstration trials on the transducer and acquisition system were satisfactory. The data acquisition system was able to scan and record the signal measured by the transducer. The stored data in the memory card of the acquisition system that was collected from the conducted field demonstration trials was successfully downloaded into the hard disk of a computer at the laboratory.

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