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Tractor hydraulic power data acquisition system

J. B.W. Roeber
University of Nebraska-Lincoln

Santosh Pitla
University of Nebraska-Lincoln, spitla2@unl.edu

Michael F. Kocher
University of Nebraska-Lincoln, mkocher1@unl.edu

Joe D. Luck
University of Nebraska-Lincoln, jluck2@unl.edu

Roger M. Hoy
University of Nebraska-Lincoln, rhoy2@unl.edu

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J. B. W. Roeber, S. K. Pitla, M. F. Kocher, J. D. Luck, R. M. Hoy

Department of Biological Systems Engineering, University of Nebraska-Lincoln, Lincoln, NE, USA

Corresponding author — S. K. Pitla, 207 L.W. Chase Hall, Department of Biological Systems Engineering,
University of Nebraska–Lincoln, Lincoln, NE, USA; email spitla2@unl.edu

Abstract
Tractor hydraulic power is used on a wide range of agricultural implements; however, the availability of operational hydraulic data at points other than full engine throttle position is limited. Operators could utilize this hydraulic data to maximize field efficiency and minimize machinery costs when determining suitable machinery for field operations. A field usable hydraulic test apparatus capable of measuring tractor hydraulic pressure and flow rate data was developed. The goal of this study was to determine if a hydraulic flow and pressure measurement device could be installed on the rear of a tractor to provide implement hydraulic power consumption at different hydraulic hose orientations. The measurement system installed allowed hydraulic lines from the tractor hydraulic remote ports to be attached to the flowmeter and pressure sensors at multiple angles of 0°, 45°, and 90° in different configuration layouts. Tests were performed at different flows and pressures for each hose configuration. The pressures were compared across configurations to a base line reading from a hydraulic pressure and flow rate measurement apparatus used by the Nebraska Tractor Test Laboratory (NTTL). Pressure deviations from the base line were small and ranged between 10.56 kPa and 32.2 kPa. Flow rate differences (<167 mL min^{-1}) were determined to be negligible (<0.5%). Calculated power differences (<33 W) were less than 1% full scale power measured. This small power loss suggested that using the hydraulic measurement apparatus developed as part of this study would enable accurate measurements of tractor hydraulic power provided to implements regardless of hydraulic hose bend angles.

Keywords: Data acquisition, Flow rate, Hydraulic power, LabVIEW, Pressure, Tractor

1. Introduction

The Organization for Economic Co-Operation and Development (OECD) Code 2 (OECD, 2016) test requirements for tractor hydraulic power only stipulate flow and pressure to be recorded at maximum engine speed. However, operators utilize hydraulic power at various engine speeds, from low idle to maximum speed, necessitating the determination of hydraulic power usage over a range of engine speeds. When instrumenting an agricultural tractor to obtain actual operational data from the hydraulic system, mounting locations and space requirements are the most important design aspects of the system. Tractor hydraulic systems must endure the stress of intermittent use and frequent on/off cycling and are widely used for powering implements where mechanical systems are too complex or electrical components are cumbersome for the necessary power requirement. Manufacturers install the entire hydraulic system in a relatively small space due to the power take-off shaft (PTO), drawbar, and 3-point hitch in the same area at the rear of the tractor (Figure 1).

Determining hydraulic power available for agricultural implements require few sensors. Implement hydraulic power consumption can be determined by measuring the pressure and the flow rate of the fluid delivered to the implement. Researchers have the option of installing a flowmeter between the main hydraulic pump and the hydraulic remote ports, or as an extension between the remote ports and the connected implement. A recommendation by the flow meter manufacturer (Flotech Activa F6206-AVB-NN, Racine Federated Inc., Racine, Wisc.) states that a minimum upstream conductor length of 10 times the flowmeter port diameter and a minimum downstream conductor length of 5 times the flowmeter port diameter is required (Flo-Tech, 2015). This is typically done to create laminar fluid flow in the measurement region to maximize the accuracy of flow rate readings. In a case where space is the limiting factor, having the recommended lengths of straight tubing in line with the flowmeter can be difficult. Unable to create a straight-lined hydraulic measurement apparatus on the rear of a tractor would introduce tubing bends on the apparatus. Tubing bends are similar to fittings and valves in that they create local energy losses (Larock et al., 2000). A previous study on agricultural tractor performance used a Hydrotecknik RE6 flow turbine installed in the main pump line upstream of the hydraulic remote block at the rear of the tractor (Burgun et al., 2013). This approach limited the hydraulic implement power
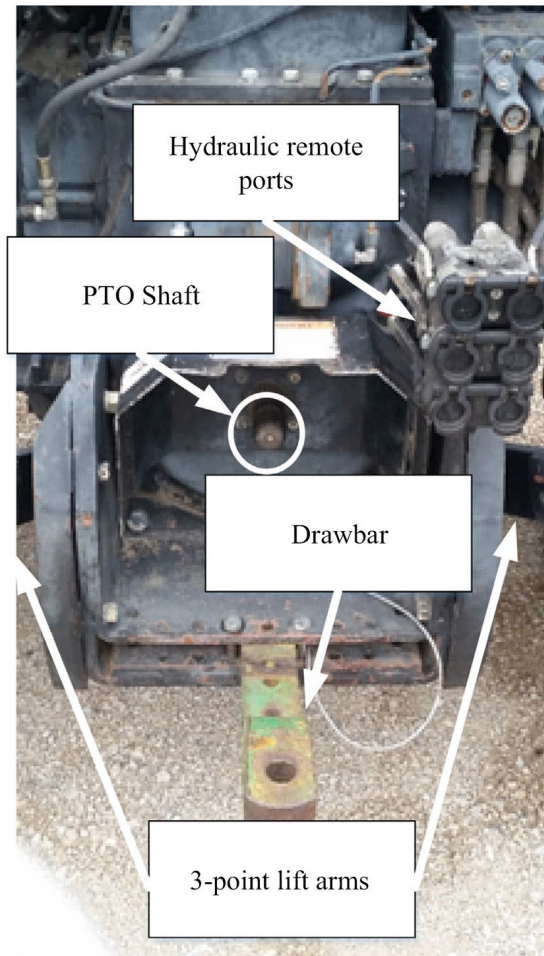
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Figure 1. Typical locations at the rear of an agricultural tractor for delivery of power to implements.

measurement accuracy by inducing the hydraulic systems efficiencies into the measured data. The author's approach also modified a tractor part which would require that the modification be undone after the project has ceased, to ensure the tractor hydraulic system functions to manufacturer specifications after the tractor returned to normal use.

This research presents a different approach for determining the hydraulic power delivered to an implement by a tractor. The goal of this new approach was to minimize modifications to the tractor hydraulic system and allow the hydraulic power test system to mount on any tractor using standard ISO 5675 hydraulic couplers. Guidelines outlined in OECD Code 2 were used for temperature and measurement tolerances. Installing a straight-line flow meter system on the rear of the tractor will be a challenge as the hydraulic test apparatus needed to allow the 3-point lift arms and the PTO shaft to function unobstructed, without adding excessive length to implement hydraulic hoses (Figure 2).

2. Objectives

The goal of this project was to develop a portable hydraulic pressure and flow measurement system. This system would attach to the remote hydraulic ports at the rear of the tractor with minimal modifications to determine the hydraulic power delivered to an attached implement. Specific objectives of the current research work were to:

- Determine which of the six tubing configurations used with a portable hydraulic pressure and flow measurement system

could be mounted without modification to a tractor and provide adjusted pressure and flow rate measurements with differences less than 2% and 0.5%, respectively.
- Determine whether the hydraulic power obtained using the portable hydraulic pressure and flow measurement system had differences less than 1% of full scale hydraulic measurement bench power measurement.

3. Materials and methods

A system to test the effect of tube bend configurations on pressure and flow rate measurement accuracy was established. This system was comprised of an agricultural tractor connected with an in-line Device Under Test (DUT) and a bench hydraulic pressure and flow rate measurement test apparatus.

3.1. Measuring devices

Sensors with analog voltage signal output were selected to allow the most flexibility and compatibility with data acquisition system (DAQ) hardware, and ease of expansion into a higher order system. Following this guideline, a turbine style flowmeter (Flotech Activa F6206-AVB-NN, Racine Federated Inc., Racine, Wisc.) which had the capability of measuring 15–303 L min^{-1} within ±1.0% of the flow reading with an analog output of 0 V DC to 5 V DC was selected to work with the higher flow capacities of hydraulic systems on newer agricultural tractors. The turbine flowmeter measures the flow rate and hence only one sensor was required in the system loop. Additional benefits of the sensor design were: supplementary internal flow straighteners on both sides of the turbine and the availability of ports for installation of temperature and pressure sensors (Flo-Tech, 2015). Analog pressure sensors are widely available in a variety of pressure ranges. The selected pressure sensor (Omega Px309, Omega Engineering Inc.) was capable of measuring 0–34.5 MPa (0–5000 psi) with an analog voltage output range of 0 V DC to 5 V DC (Omega, 2014). The data acquisition interface between the sensor assembly and the data acquisition computer was a National Instruments (NI) myDAQ (National Instruments Corporation, Austin, Texas). NI myDAQ was a portable DAQ with multiple analog/digital inputs and outputs. A single 16- bit analog-to-digital converter was used to sample both analog channels with voltages of 0–5 V DC and sampling rates of 50 Hz per channel. Both analog channels were utilized as differential voltages, one channel for the pressure sensor and the other for the flowmeter on the DUT.

The flow meter ports (25.4 mm diameter) with SAE 16 threads, were connected to a series of reducers and adapters decreasing the dimensions from SAE 16 to 19 mm National Pipe Thread (NPT), and to 19 mm (¾ in.) medium pressure hydraulic hose (NRP-Jones Hydra-Lite II, 21.4 MPa maximum pressure rating) with ISO 5675 quick-couplers. The sensors and hoses were mounted to a plywood board using U-bolts as illustrated in Figure 3a and b. The hose ends were able to be mounted with the hose in a straight-line configuration (0°), 45°, 90° or any combination of these bends (Figure 3a and b) using the plywood board and U-bolts; however, not all combinations were used for testing. The six tubing configurations selected were: 0-0, 45-0, 45-45, 90-0, 90-45, and 90-90. The reciprocal tubing configurations: 0-45, 0-90, 45-90 were assumed unnecessary due to symmetry. When organizing the tubing configurations as the main treatments, an orientation was selected in which the inlet and outlet were parallel but have opposite direction. For example, the male inlet coupler of the DUT would insert into the rear-facing tractor remote port and the female outlet coupler of the DUT would have the same rear-facing direction as the tractor
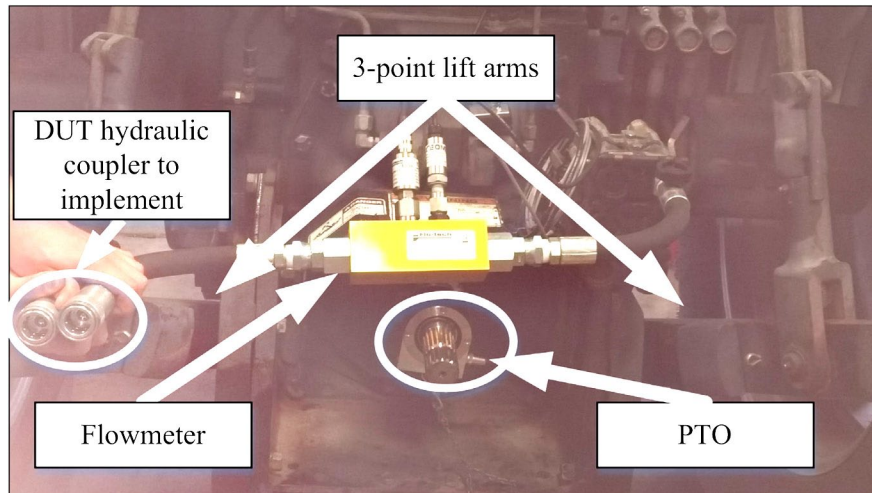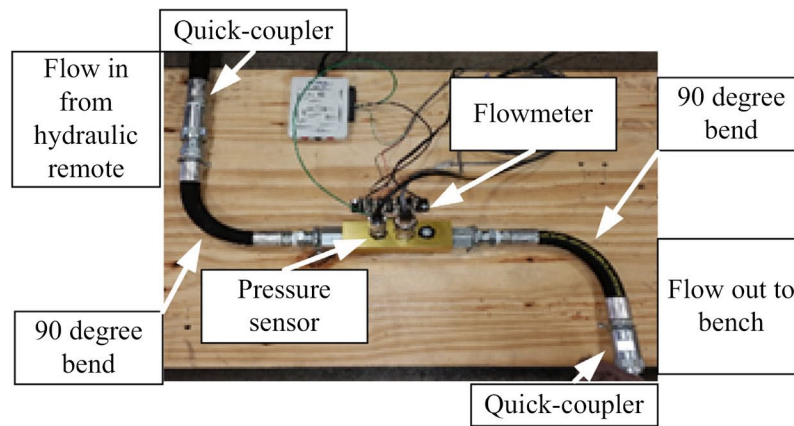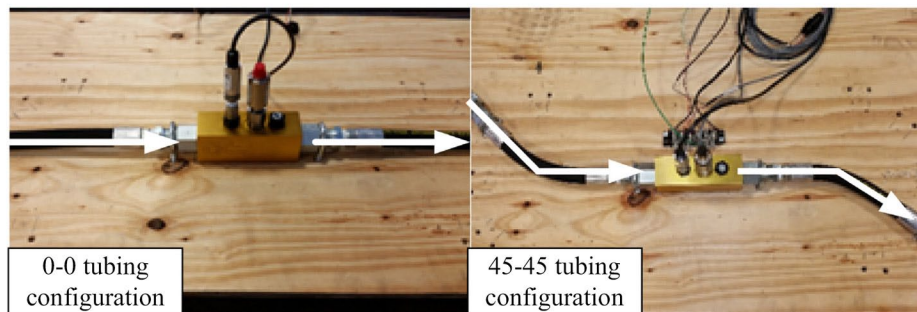
Figure 2. DUT showing mounting location between 3-point lift arms, and PTO shaft.

(a)

(b)

Figure 3. (a) DUT in a 90-90 tubing configuration, (b) DUT in 0-0 and 45-45 tubing configurations.

remote port. This orientation would allow the DUT to function as an extension hose installed on a tractor (Figure 3a and b).

To test if there was an effect of the degree of bending on the accuracy of pressure or flow rate measurements, the flow rate and pressure readings from the DUT were compared to the flow rate and pressure readings from a hydraulic test bench measurement apparatus, hereafter referred to as the Bench (Figure 4).

The Bench used by the Nebraska Tractor Test Laboratory (NTTL) consisted of a Flo-tech flowmeter with the same specifications as the one used on the DUT, strain-type pressure sensors, a thermocouple,

and a needle valve. The sensors are calibrated annually, traceable to ISO 9001. The flowmeter assembly was mounted with a straight steel tubing of 30 cm (12 in.) in length and 19 mm (¾ in.) diameter, connected to hydraulic hoses of the same diameter on both the upstream and downstream sides. The Bench DAQ board (NI cDAQ 9174, National Instruments Corporation, Austin, Texas) with an analog, strain, and thermocouple modules was used for collecting the data. NI modules for data collection on the Bench include an 8-channel universal sink/source digital module (NI 9435, National Instruments Corporation, Austin, Texas) to read the digital signal of the engine speed sensor
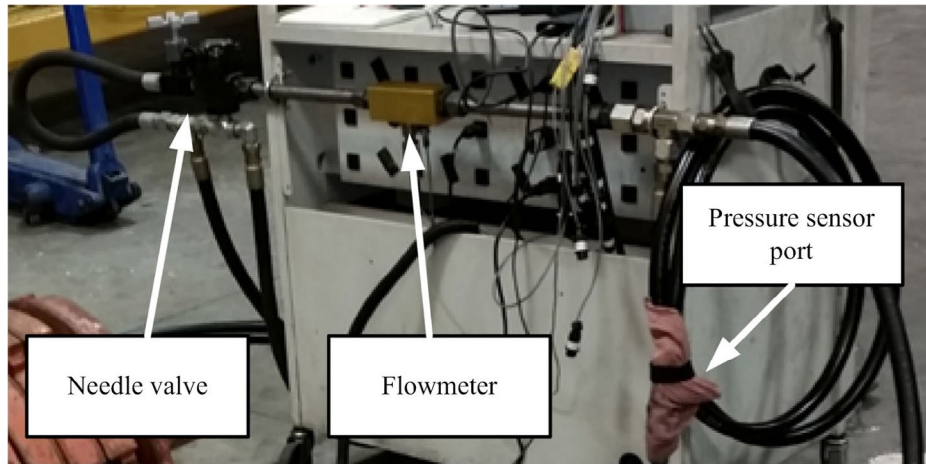
Figure 4. Bench test apparatus used by Nebraska Tractor Test Laboratory (NTTL).

(D12E2P6FV, Banner Engineering Corporation, Minneapolis, Minn.), a 4-channel thermocouple module (NI 9211, National Instruments Corporation, Austin, Texas) for ambient and hydraulic temperatures (OL-703, OMEGA Engineering, Inc., Stamford, Conn.), a 4-channel bridge analog module (NI 9237 D-SUB, National Instruments Corporation, Austin, Texas) to read strain-based pressure sensors, and a universal analog module (NI 9219, National Instruments Corporation, Austin, Texas) which measured the analog voltage output of the flowmeter. For the current tests only two strain channels for pressure sensors, two temperature channels, and an analog voltage channel for the flowmeter were utilized.

3.2. Test setup

The DUT used the fixed position flowmeter with variable position coupler locations as described earlier. The systems were connected so that the DUT was connected to the tractor's extend remote port via a 19 mm (¾ in.) diameter hydraulic hose with a length of 1.8 m. Flow exiting the DUT went through the Bench system and returned to the tractor's retract remote port. This setup placed the DUT and the Bench flowmeters and pressure sensors in series before the needle valve. A schematic illustrated in Figure 5, depicts the connections and sensor

locations of the DUT and the Bench in relation to the tractor providing the hydraulic flow.

A Case IH tractor (DX55, CNH America LLC, Racine, Wisc.) with an engine rating of 35.8 kW at an engine speed of 2700 rev min^{-1} was used to generate fluid flow for the tests ranging from approximately 20 L min^{-1} to 44 L min^{-1} measured by the Bench flow meter, corresponding to different engine speeds set by the tractor tachometer (Figure 6). These values were the average of 3 tests at each engine speed setting in the 0-0 tubing configuration with the needle valve fully open.

3.3. Data acquisition hardware and software program

A LabVIEW graphical user interface (GUI) was created to read and log the signal data from the DUT using the LabVIEW application programming interface (API). The LabVIEW GUI allowed the user to specify the channel of the pressure and flow sensors via the DUT Channels array (Figure 7).

Scaled engineering values allowed the flow rate, pressure, and power to be displayed in real-time to the user. "Continuous Data" push button control allowed the user to start and stop collection of mean 1 Hz raw data gathered while the control was in the "On" state, and omit data when the control was "Off". The "Data Point" push button control
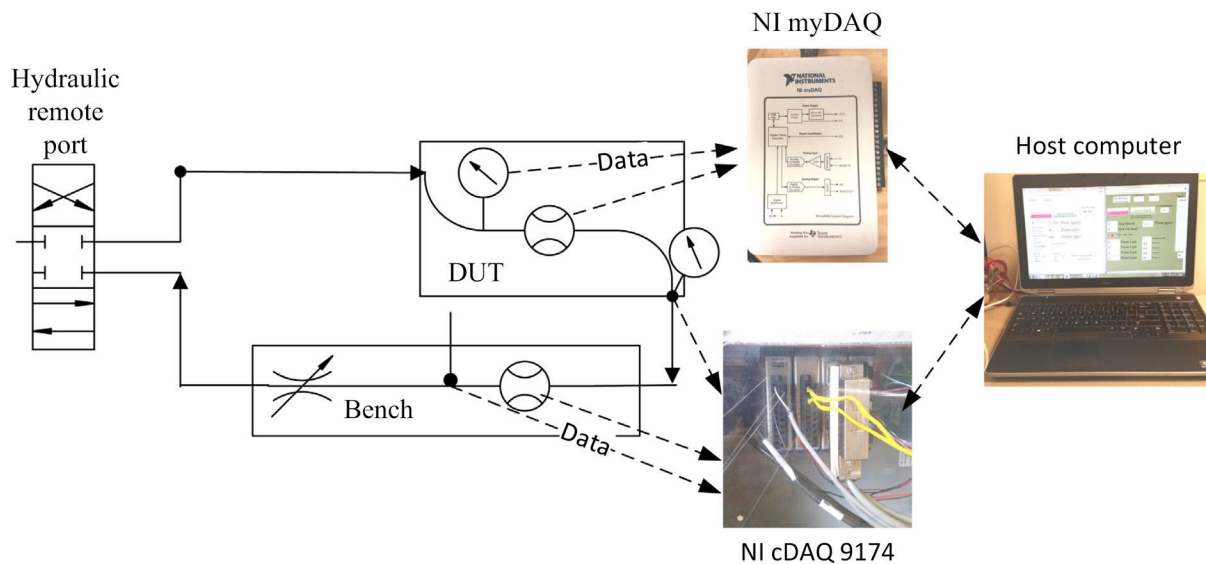

Figure 5. Schematic diagram showing system flow direction and sensor locations.
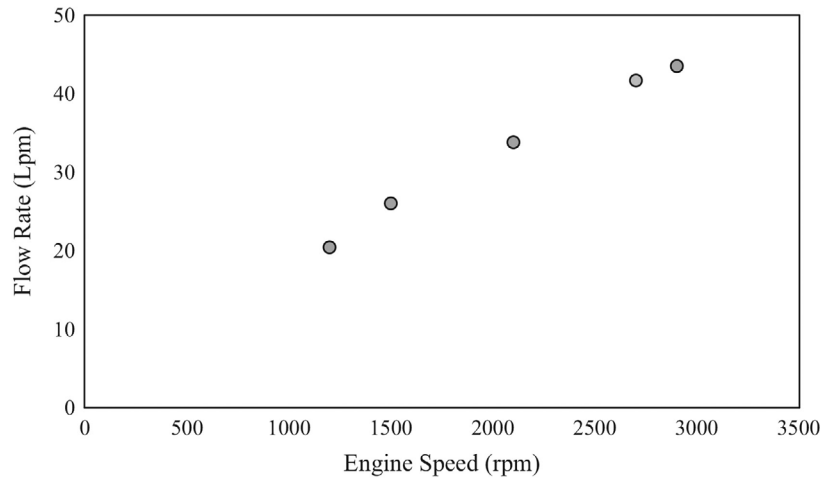
Figure 6. Average hydraulic flow rate from one hydraulic remote versus engine speed with the tractor's hydraulic remotes adjusted for full flow.
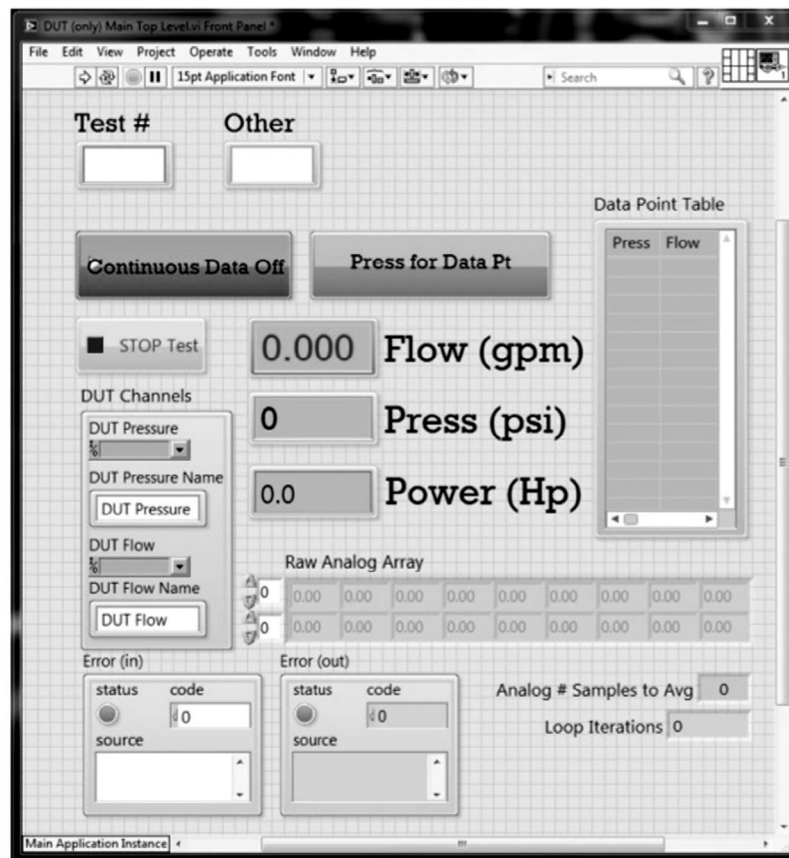

Figure 7. LabVIEW Front Panel for DUT testing.

allowed the user to collect a single 1 s mean data sample, which was helpful collecting the necessary OECD Code 2 required hydraulic performance parameters. The raw analog data were presented in an array of values at the sampling frequency of 1000 Hz.

A schematic drawing was presented in a block diagram (Appendix A) to depict the flow of data. LabVIEW had pre-generated DAQ virtual instruments (VIs) which simplified development of the block diagram program. The main components of any LabVIEW VI are initializing, reading/writing values, and closing. In order to save the data that was read, the data needed to be logged to a file.

The API used to gather the Bench results was developed for the official testing by the NTTL engineers. In the NTTL version, up to four pressure sensors could be used along with a flowmeter, engine speed sensor, and a thermocouple. The channels were set up in NI Measurement and Automation Explorer (NI MAX) as tasks that could be called by the LabVIEW API.

3.4. Test procedure

The tractor used in the study had a rated engine speed of 2700 rev min^{-1}, and high idle speed of 2900 rev min^{-1}, so both speeds were chosen for the high flow rate tests. Engine speeds for lower flow rates included: 1200 rev min^{-1}, which was assumed to be a transitional engine speed for transport to field operational mode and vice versa; 1500 rev min^{-1},
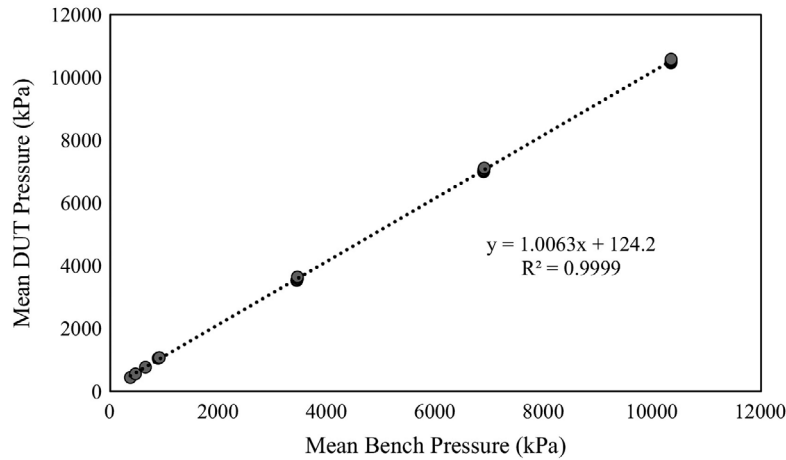
Figure 8. Average pressure value comparison between Bench and DUT in the 0-0 tubing configuration at each engine speed, and pressure setting.

Table 1. Adjustment terms (P'_{DUT}) applied to DUT pressure measurement based on 0-0 configuration.

Engine speed (rev min^{-1})	Bench pressure setting (MPa)			
	Needle valve fully open P'_{DUT} (kPa)	3.45	6.90	10.34
1200	72	94	112	127
1500	94	114	134	150
2100	122	148	169	191
2700	173	190	210	224
2900	172	198	221	242

The DUT pressure was assumed to be higher than the Bench pressure at the 0-0 tubing configuration due to a pressure drop because of the friction losses in the hose and adapters, and the orifices of the quick-couplers. After determining that the pressure drop across the coupler orifice was approximately 227 kPa (33 psi) at the maximum flow rate, adjustment terms were developed for the DUT pressure measurements to minimize the differences in the system measurements, i.e. differences between the DUT in the 0-0 tubing configuration and the Bench. The adjustment terms calculated (Equation (1)) were the difference between the DUT pressure and the Bench pressure at 0-0 tubing configuration for each engine speed and pressure setting:

$$P'_{DUTe,p} = P_{DUTe,p} - P_{Be,p} \qquad (1)$$

where

P'_{DUT} = Mean DUT pressure adjustment term (kPa) at the 0-0 tubing configuration
P_{DUT} = Mean DUT pressure (kPa) from 0-0 tubing configuration
P_B = Mean Bench pressure (kPa) from 0-0 tubing configuration
e = engine speed setting
p = pressure setting

The adjusted DUT pressure (Equation (2)) was the pressure after applying the adjustment terms (Equation (1)) for each engine speed/pressure setting.

representing a ½ throttle position in most geared transmission tractors used during headland turns; and 2100 rev min^{-1}, representing a full throttle condition during working loads; these engine speeds cover the range of typical engine operating speeds for tractors with rated engine speeds of 2100 or 2200 rev min^{-1}. Using the Nebraska Tractor Test Report 1837 (NTTL, 2004) for the tractor model Case IH DX55, a pressure of 17.58 MPa (2550 psi) was listed as the maximum sustained by the pump; therefore, Bench pressure settings at minimum pressure, 3.45, 6.90, 10.34, 13.79, 17.58 MPa (500, 1000, 1500, 2000, 2550 psi) were used during tests.
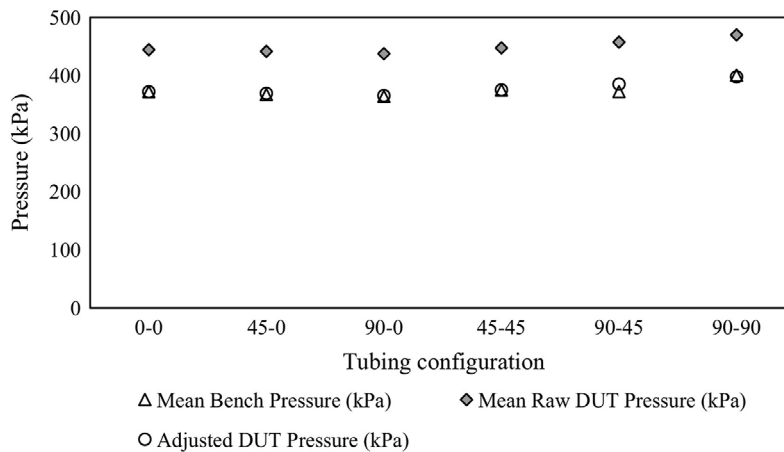

Figure 9. Average pressure values from test arrangements with an engine speed of 1200 rev min^{-1} and the needle valve fully open.
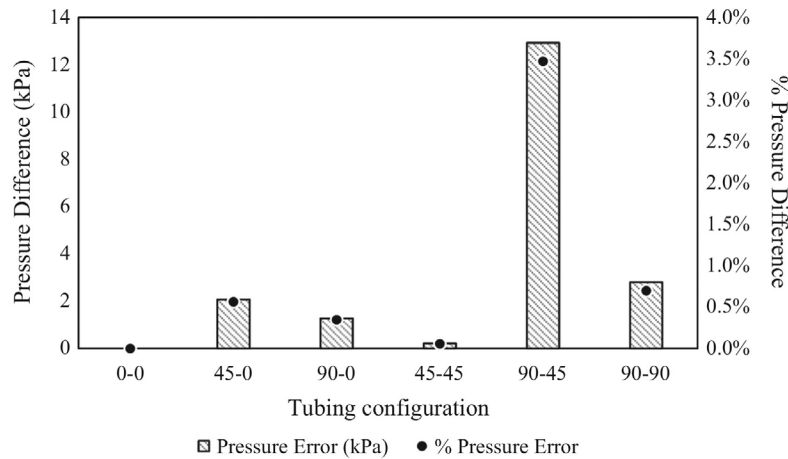
Figure 10. Average pressure differences of the tubing configuration with an engine speed of 1200 rev min^{-1} and the needle valve fully open.
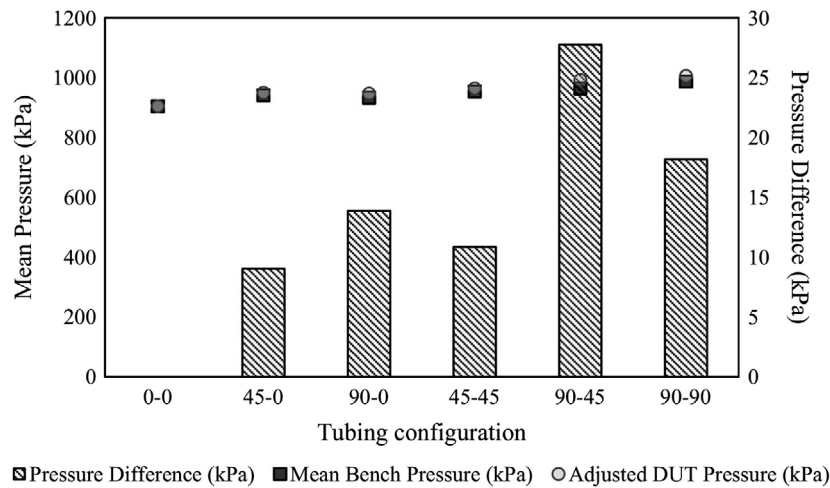

Figure 11. Pressure results with tubing configurations at 2900 engine rev min^{-1} and the needle valve fully open.

Table 2. Pressure results with the tubing configurations at 1200 engine rev min^{-1}, and the needle valve fully open (*Capital letters in superscript indicate significant differences in pressure among tubing configurations).

Treatment	DUT tubing configuration	Mean Bench pressure (kPa)	Mean DUT pressure (kPa)	Adjusted DUT pressure (kPa)	Pressure difference (kPa)*	% Pressure difference
1	0-0	372	444	372	0.0[A]	0.00
2	45-0	367	442	369	2.1[A]	0.57
3	90-0	364	438	365	1.3[A]	0.35
4	45-45	375	448	375	0.2[A]	0.06
5	90-45	372	458	385	12.9[B]	3.47
6	90-90	401	470	398	2.8[A]	0.70

Table 3. Pressure results with the tubing configurations at 2900 engine rev min^{-1}, and the needle valve fully open (*Capital letters in superscript indicate significant differences in pressure among tubing configurations).

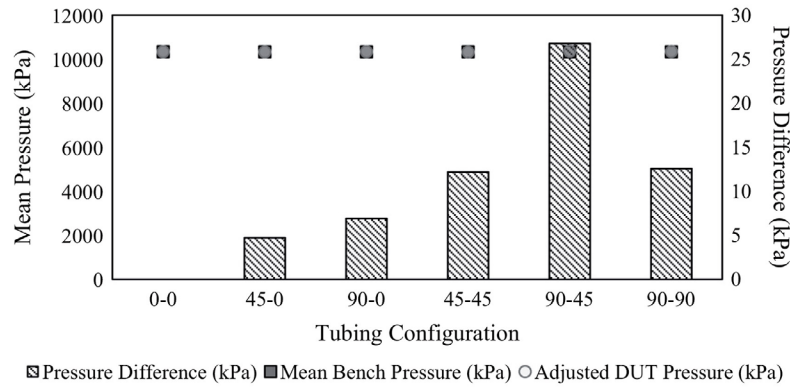| Treatment	DUT tubing configuration	Mean Bench pressure (kPa)	Mean DUT pressure (kPa)	Adjusted DUT pressure (kPa)	Adjusted pressure difference (kPa)*	% Pressure difference
1	0-0	905	1077	905	0.0[A]	0.00
2	45-0	941	1122	950	9.0[AB]	0.96
3	90-0	933	1119	947	13.9[BC]	1.49
4	45-45	954	1137	965	10.9[BCD]	1.14
5	90-45	965	1164	993	27.8[E]	2.88
6	90-90	988	1178	1006	18.2[BCDE]	1.84

Figure 12. Pressure results with tubing configurations and needle valve resistance of 10.34 MPa (1500 psi) at 2900 rev min^{-1}.

$$P''_{\text{DUTn,e,p}} = P_{\text{DUTn,e,p}} - P'_{\text{DUTe,p}} \qquad (2)$$

where

P''_{DUT} = Mean adjusted DUT pressure (kPa)
P_{DUT} = Mean DUT pressure (kPa)
P'_{DUT} = Mean DUT pressure adjustment term (kPa) from 0-0 tubing configuration
n = nth tubing configuration
e = engine speed setting
p = pressure setting

Starting with the DUT in a 0-0 tubing configuration, the hydraulic oil temperature was brought up to 60 °C. OECD test procedure for hydraulic power performance steady-state laboratory test settings requires a temperature range of 65 °C ± 5 °C (OECD, 2016). With the tests starting at 60 °C, as subsequent tests were performed the temperature would be within OECD tolerance for a longer duration without the system becoming overheated. The engine speed was then set to 1200 rev min^{-1} with the needle valve fully open. Thirty seconds of the 1000 Hz data averaged over 1 s periods were collected, and then the needle valve was adjusted until the pressure at the Bench was 3.45 MPa (500 psi). This process was repeated for the subsequent pressure levels in increasing order to minimize the rate at which the oil temperature increased. A safety relief in the tractor operator's hydraulic controls, which disengaged the hydraulic lever detent, limited maximum system pressure to around 12.8–13.2 MPa (1850–1920 psi). With this upper limit on the hydraulic system pressure, test pressure levels were reduced to: needle valve fully open, 3.45, 6.90, and 10.34 MPa. This procedure was repeated for each of the engine speeds before proceeding to the other hose configurations (45°, 90°, etc.). Three replications were made at each hose configuration (5 engine speeds × 6 tubing configurations × 4 pressures × 3 repetitions = 360 data points). The order of the tubing configuration treatments was randomized for each

replicate. Within each tubing configuration, the order of the engine speed treatments was chosen randomly. The pressure level treatments within each engine speed treatment were applied in order from lowest to the highest pressures to avoid overheating the hydraulic oil. This randomization approach was used to avoid excessive delays (caused by the time required to change tubing configuration and engine speed) in completing measurements within each replicate.

Since the Bench and DUT data were logged in two independent files on the same host computer, for each individual test run, the two files were merged into one file with the file timestamps used to confirm which two files to combine for each test run. The replications for each pressure/engine speed/tubing configuration were averaged together to determine each treatment mean.

Two differences were determined as results for each treatment combination: the difference between the pressure measured by the DUT and the pressure measured by the Bench, and the difference between the flow rate measured by the DUT and the flow rate measured by the Bench. ANOVA was employed to determine if there were any differences among the treatment means. The Least Significant Difference (LSD) tests were used to determine which (if any) differences among the treatment means were significant. The pressure differential was the difference between the adjusted DUT pressure and the Bench pressure. Percent difference was calculated based on the adjusted pressure difference relative to the overall Bench pressure:

$$P_{\text{En}} = \frac{(P''_{\text{DUTn}} - P_{\text{Bn}})}{P_{\text{Bn}}} * 100 \qquad (3)$$

where

P_{E} = Pressure difference (%)
P''_{DUT} = Mean adjusted DUT pressure (kPa)
P_{B} = Mean Bench pressure (kPa)
n = nth tubing configuration

Table 4. Pressure results with the tubing configurations at 2900 engine rev min^{-1}, 10.34 MPa (*Capital letters in superscript indicate significant differences in pressure among tubing configurations).

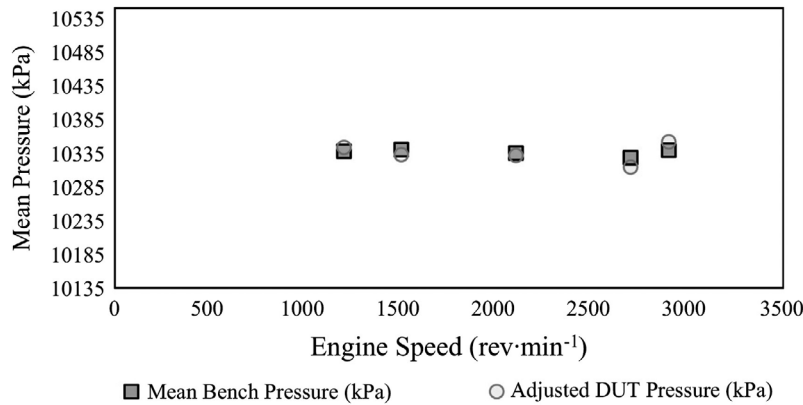| Treatment	DUT tubing configuration	Mean Bench pressure (kPa)	Mean DUT pressure (kPa)	Adjusted DUT pressure (kPa)	Adjusted pressure error (kPa)*	% Pressure error
1	0-0	10,342	10,584	10,342	0.0[A]	0.00
2	45-0	10,337	10,574	10,332	4.7[AB]	0.05
3	90-0	10,336	10,585	10,343	6.9[AC]	0.07
4	45-45	10,337	10,566	10,324	12.2[B]	0.12
5	90-45	10,338	10,606	10,365	26.8[D]	0.26
6	90-90	10,339	10,594	10,352	12.6[C]	0.12

Figure 13. Pressure results by engine speed with the 90-90 tubing configuration and needle valve resistance of 10.34 MPa.

4. Results and discussion

The mean DUT pressure was higher than the Bench pressure at the 0-0 tubing configuration as predicted. Figure 8 plots the Bench pressure versus the DUT 0-0 tubing configuration pressure, the higher pressure variation was noticed at the needle valve fully open pressure setting. The linear regression indicates a strong correlation between the Bench and DUT pressures ($m = 1.0063$), with a high coefficient of fit ($R^2 = 0.9999$). Pressure values outside of the measured engine speed/pressure settings were calculated using the regression equation (Figure 8).

The adjustment term for each engine speed and pressure setting ranged between 72 and 242 kPa (10–35 psi) (Table 1). Pressure adjustment terms (P0 DUT) had a direct relationship with pressure and engine speed (flow rate) which was consistent with fluid mechanics theory.

Figs. 9–11 show comparisons between the mean pressures of the Bench and the adjusted DUT pressures (P00 DUT) for engine speeds and tubing configurations. In comparing the pressures between different tubing configurations within the 1200 rev min^{-1} engine speed, a larger difference was seen at the 90-45 tubing configuration (Figure 9). This higher pressure difference pattern was present in all the engine speed/pressure settings. A least significant difference value of 10.56 kPa was calculated to be statistically significant pressure differences. The 90-45 tubing configuration had statistically significantly pressure differences compared to the other tubing configurations at all engine speed and pressure setting combinations.

Table 2 outlined the pressure differences at a low engine speed, low pressure setting. The adjusted DUT pressures (Equation (2)) were the pressure after applying the adjustment terms (Equation (1)). Pressure difference was the deviation of the adjusted DUT pressure (Equation (2)) from the bench pressure. The percent pressure difference (Equation (3)) used the adjusted pressure terms. There was no statistically significant pressure differences between treatments 1, 2, 3, 4, and 6, and treatment 5 had statistically significant difference relative to other treatments.

The highest pressure difference in Table 2 of 12.9 kPa (3.47%) occurred at the 90-45 configuration when compared to other tubing configurations. OECD Code 2 allows a ±2.0% tolerance in hydraulic system pressure (OECD Code 2, 2016).

When comparing the pressure differences between the lowest engine speed (Figure 10) and the highest engine speed (Figure 11) with the "needle valve fully open", the pressure difference increased with engine speed. As an example, at 1200 rev min^{-1} engine speed the pressure difference at 90-45 configuration was approximately 13 kPa, whereas for the same tubing configuration the pressure difference

Table 5. Pressure results by engine speed with 90-90 tubing configuration and needle valve resistance of 10.34 MPa.

Engine speed	Mean Bench pressure (kPa)	Mean DUT pressure (kPa)	Adjusted DUT pressure (kPa)	Pressure difference (kPa)	% Pressure difference
1200	10,338	10,471	10,343	5.5	0.05
1500	10,340	10,498	10,333	7.5	0.07
2100	10,335	10,529	10,332	3.7	0.04
2700	10,328	10,566	10,314	13.9	0.14
2900	10,339	10,594	10,352	12.6	0.12

increased to approximately 28 kPa at 2900 rev min^{-1} (Figs. 10 and 11). The higher engine speeds also accounted for more significant differences in the mean pressures due to a larger pressure drop across the DUT outlet/Bench inlet orifice.

A summary of pressure differentials for different DUT tubing configurations at an engine speed of 2900 rev min^{-1} is presented in Table 3. The significant pressure differences in treatment means were between treatments 1 and treatments 3, 4, 5, and 6, and between treatment 5 and treatments 2, 3, and 4. It should be noted that the 90-45 configuration had the highest pressure error (2.88%) of approximately 27.8 kPa.

Figure 12 presents the pressure differentials at the maximum operating pressure of 10.4 MPa and maximum engine speed of 2900 rev min^{-1}. When pressure settings are changed from the lowest (needle valve fully open) to the highest system pressure (10.34 MPa), there were significant differences between the mean pressures (Figs. 11 and 12).

A summary of the pressure differentials for different tubing configurations at the highest system pressure (10.34 MPa) and high idle engine speed of 2900 rev min^{-1} is presented in Table 4. It can be noted that the 90-45 configuration had the highest pressure error of 26.8 kPa (0.26%) relative to other tubing configurations.

The 90-90 configuration of the DUT was the most likely configuration for tractor hydraulic power data acquisition, given the restricted space at the rear of the tractor. This tubing configuration also was considered as an extreme case where there was significant bending in the hydraulic hoses of the DUT. Mean bench pressures and mean adjusted DUT pressures are shown in Figure 13 for each engine speed at the 10.34 MPa pressure with the 90-90 tubing configuration, the mean pressure axis was scaled to indicate a maximum range of ±2.0% from the bench pressure setting. Based on Figure 13, it was observed that as the engine speed (and the flow rate) increased the pressure difference was relatively small and well within OECD Code 2 tolerances. This trend was consistent at other operating pressures.
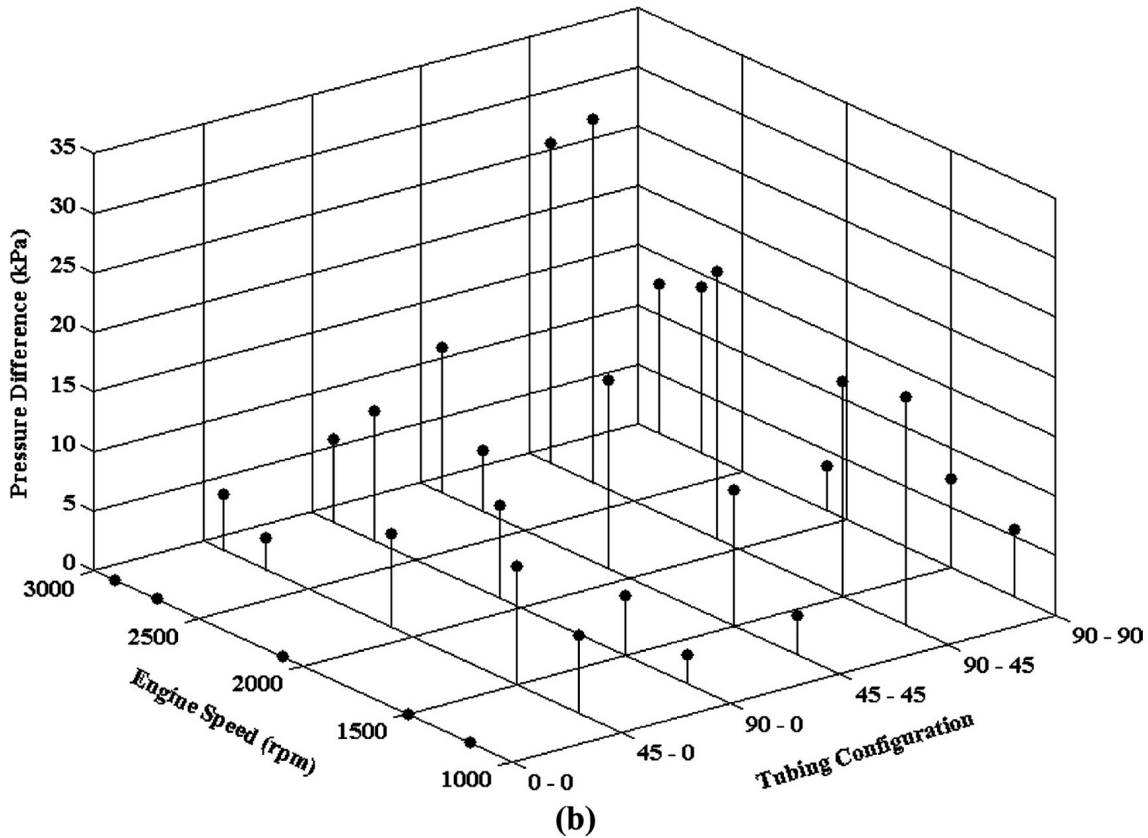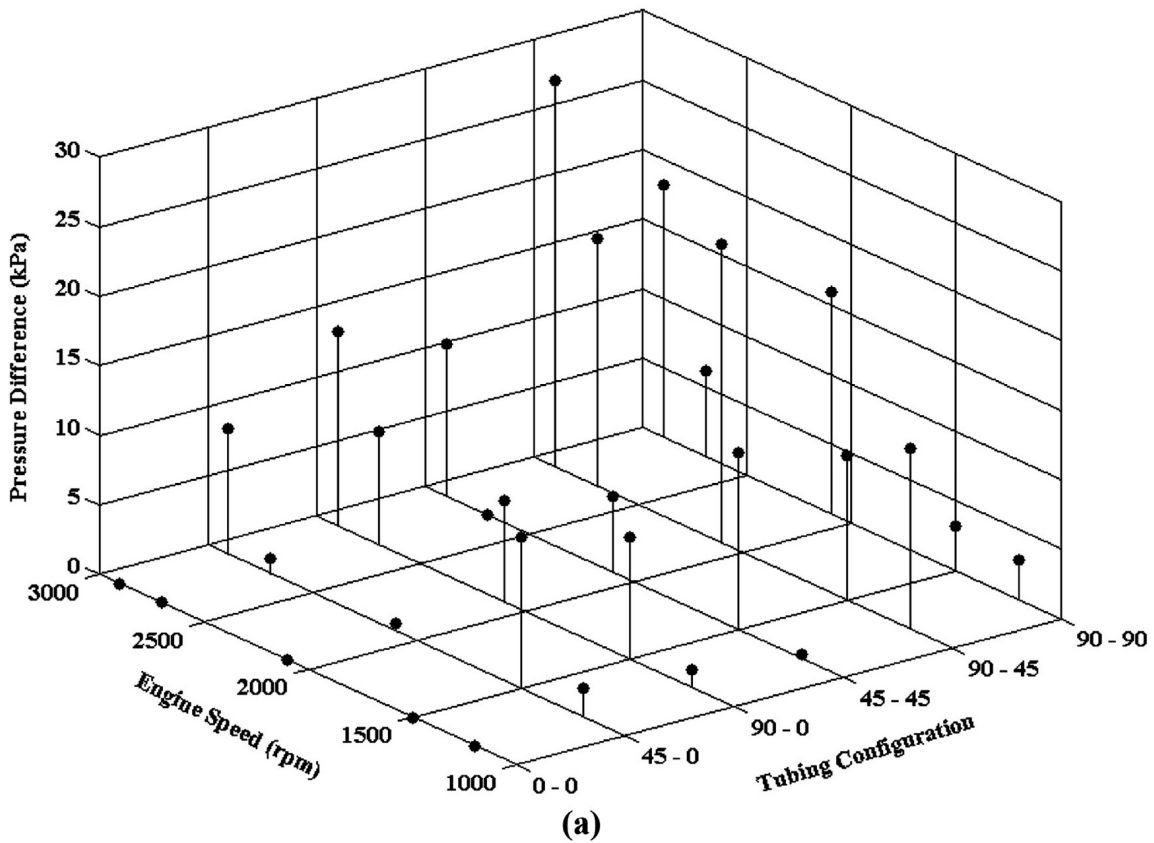
Figure 14. Pressure errors (%) for engine speed by tubing configuration combinations at pressure levels of (a) needle valve fully open, and (b) 10.34 MPa.
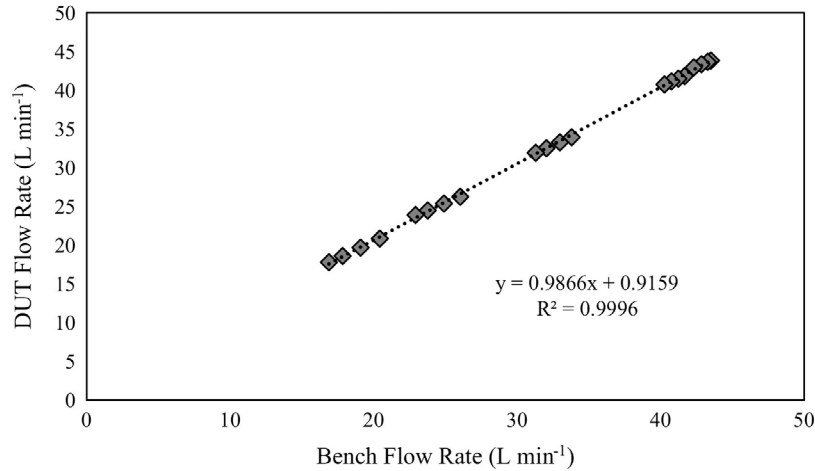
Figure 15. Mean Bench vs. mean DUT flow at the 0-0 tubing configuration.

Table 6. Adjustment terms applied to DUT flow measurement based on 0-0 configuration.

Engine speed (rev min^{-1})	Bench pressure setting (MPa)			
	Needle valve fully open Flow adjustment (L min^{-1})	3.45	6.90	10.34
1200	0.42	0.59	0.77	0.91
1500	0.22	0.47	0.72	0.98
2100	0.13	0.29	0.47	0.64
2700	0.15	0.26	0.39	0.48
2900	0.32	0.39	0.52	0.64

A summary of the pressure differentials at the 90-90 tubing configuration for different engine speeds is presented in Table 5. A maximum difference of 13.9 kPa (0.14%) is observed at an engine speed of 2700 rev min^{-1}. Less than 0.15% pressure difference was observed at all engine speeds for the 90-90 configuration indicating that this tubing configuration can be used for hydraulic flow and pressure data collection.

Figure 14a illustrates the pressure differentials with all combinations of tubing configurations and engine speeds when the needle valve was fully open. As discussed previously, the 90-45 tubing configuration consistently had the largest significant differences in pressure. The

pressure differences ranged from 0 kPa at 2700 rev min^{-1} in the 45-45 tubing configuration to 27.8 kPa (2.88%) at 2900 rev min^{-1} in the 90-45 tubing configuration. As engine speed changed, there was no discernable pattern in pressure differences for every tubing configurations at the needle valve fully open condition. The 45-0, 90-0, 45-45, and 90-90 all increased in difference from 1200 to 1500 engine rev min^{-1}. All the tubing configurations had higher pressure differences at the highest engine speed of 2900 rev min^{-1} (Figure 14a). At the system pressure of 10.34 MPa (Figure 14b), the pressure differences ranged from 2.4 kPa (0.02%) at 1200 rev min^{-1} in the 90-0 tubing configuration, to 30.4 kPa (0.29%) at 2700 rev min^{-1} in the 90-45 tubing configuration. The pressure differences increased as engine speed increased to 2700 rev min^{-1} in the 90-0, 90-45, and 90-90 tubing configurations. Pressure differences in the 45-0 tubing configuration remained below 10 kPa. The pressure differences in the 45-45 tubing configuration increased as engine speed increased to 2100 rev min^{-1}; all the pressure differences in higher engine speeds were higher than the lowest engine speed setting.

The observed mean DUT flow was lower than the Bench flow at the 0-0 tubing configuration ($m = 0.9866$) indicating that an adjustment term was needed for the DUT flow measurement (Figure 15) due to possible flow restrictions in the hydraulic couplers. The approach was the same as the pressure adjustment (Equation (1)). Flow rates outside of the measured engine speed/pressure settings were calculated using the regression equation.
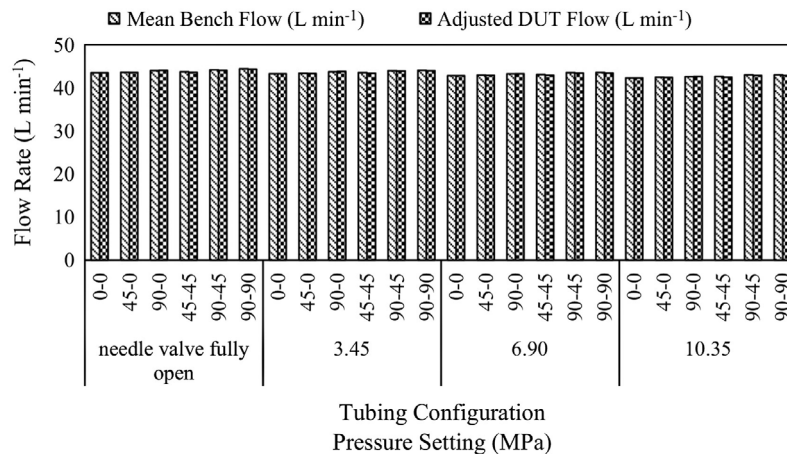

Figure 16. Differences in flow rate between the Bench and the adjusted DUT at 2900 rev min^{-1}.
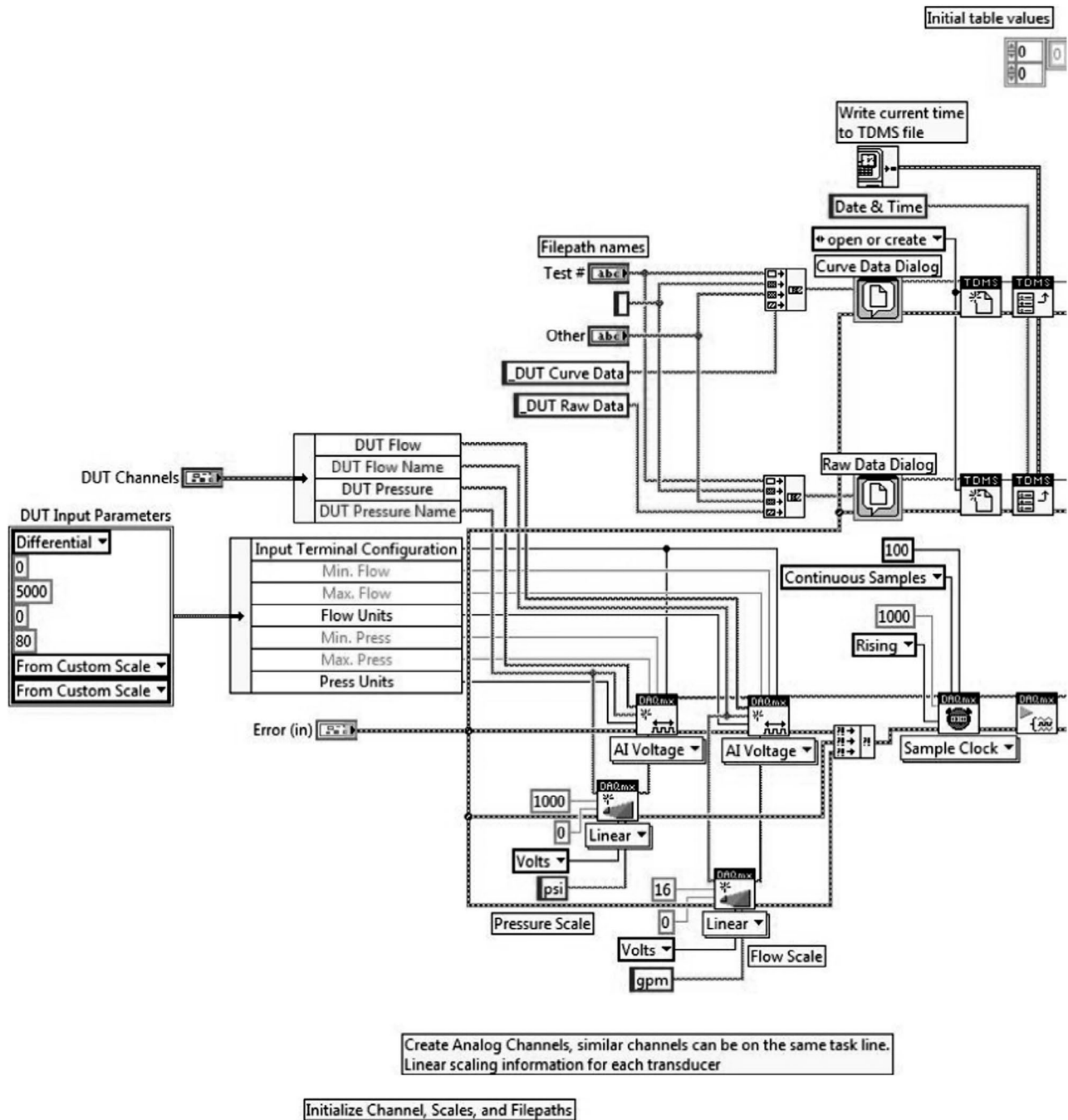
Figure 17. Block Diagram of LabVIEW program. Illustrates how channels are created and initialized.

Flow adjustment terms applied at different engine speeds are presented in Table 6. The maximum adjustment term of 0.98 L min⁻¹ (4.27%) was applied at 10.34 MPa system pressure and the engine speed of 1500 rev min⁻¹. With increase in operating pressures the magnitudes of the flow adjustment pressures increased.

After applying the adjustment terms an ANOVA table was developed with a LSD value of 0.067 L min⁻¹ (0.018 gal min⁻¹). Configurations which had the most significant differences between the means appeared within the 2900 rev min⁻¹ range, with the largest significant difference being 0.17 L min⁻¹ (0.04 gal min⁻¹, 0.38%). However, these differences are small compared to the overall flow rate (Figure 16), so it was assumed that the flow was within a reasonable error of 0.2 L min⁻¹ or approximately 0.5% of full scale.

The power measured at the Bench and the DUT was calculated using Equation (4), the adjusted pressure and flow values were used to calculate the DUT power.

Hydraulic power is a calculated value of pressure and flow:

$$\text{Power (kW)} = \frac{P \text{ (kPa)} * 0.001 \text{ m}^3 * Q \text{ (L min}^{-1})}{60 \text{ s}} \tag{4}$$

Power = mean hydraulic power (kW)
P = mean hydraulic pressure (kPa)
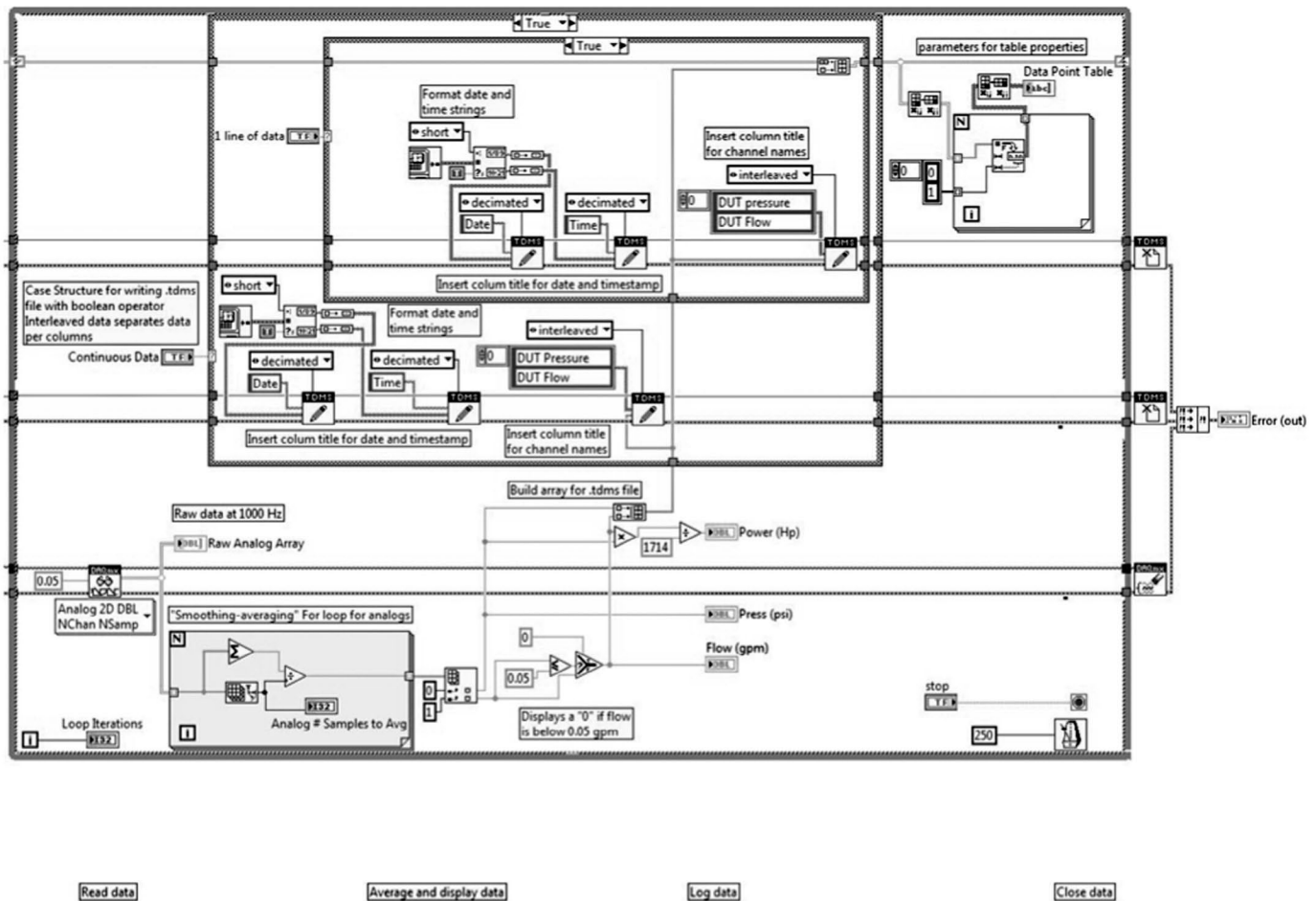Q = mean hydraulic flow (L min⁻¹)

Figure 18. Block Diagram of LabVIEW program. Illustrates the reading and logging of the data.

The largest differences in power occurred at the same tubing configuration/engine speed/pressure setting as the significant pressure differences. Overall, the largest difference in power occurred at the 45-45 tubing configuration at the highest engine speed setting (33 W). When considering as a percentage of the power measurement, the 45-45 tubing configuration maximum power difference was 0.46% of the Bench power.

5. Summary and conclusions

A data acquisition system which was instrumented without modifying the tractor to measure and record hydraulic pressure and flow rate was successful. Using the OECD Code 2 procedure for hydraulic power measurement, tests were conducted at typical engine speeds other than the governor maximum speeds. The results showed that the DUT pressure was higher than the Bench pressure as anticipated due to the pressure drop across the hydraulic fittings. Adjustment terms were made to correct for these system differences at the 0-0 tubing configuration. After the adjustment terms were made, the largest differences occurred in the 90-45 tubing configuration with a pressure differential range of 10.4 kPa (2.24%) to 32.2 kPa (0.93%) throughout all the engine speed and pressure combinations. Higher engine speed (flow rate) settings showed larger pressure differences as expected, in the most extreme tubing configuration (90-90) with the largest difference of 21.3 kPa (0.62%). The largest differences in pressure measurements were at

the higher engine speed settings as you would expect. These pressure differences were within OECD Code 2 permissible measurement tolerances of 2.0%; however, the percent difference was above 2.0% at low pressure settings due to the lower Bench pressure. Flow differences between the Bench and DUT were determined to be below 0.2 L min^{-1} (0.5%) which was considered negligible. Significant differences in the flow rate means happened more often at the higher engine speed settings, indicating possible flow restriction through the DUT coupler. The calculated power measurement difference was also negligible (<33 W, 0.46%). When instrumenting onto the rear of a tractor in the extreme bending case of 90-90, the differences are less than 22W (0.44%). With the largest power difference of 33 W, any tubing configuration could be applied. As this system will be used in field conditions and OECD Code 2 presents procedures for laboratory tests, it was determined that the differences were within the necessary measurement accuracy for field use. With these findings, it was concluded that bending in the tubing before and after the flowmeter in this system did not affect the accuracy of the power measurements.

Appendix A. LabVIEW Block Diagram See Figures 17 and 18.

References

Burgun, C., Lacour, S., Delacroix, A., Descombes, G., Doyen, V., 2013. Computing time and fuel requirements to assess efficiency of a field work from conventional laboratory tests: application to a plowing operation. Agric. Food Sci. 22 (2), 247–261, http://ojs.tsv.fi/index.php/AFS/article/view/7629/6310 (Retrieved July 1, 2014).

Flo-Tech, 2015. Installation and Operating Instructions. Racine Federated Inc., Racine, Wisc. http://www.badgermeter.com/Badger-Files/PDFs/Industrial-Products/SEN-UM-00987-EN.pdf

Larock, B.E., Jeppson, R.W., Watters, G.Z., 2000. Local and minor losses. In: Hydraulics of Pipeline Systems. CRC, Boca Raton, FL, p. 15.

NTTL, 2004. Test 1837: Case IH DX 55 Diesel. Nebraska Tractor Tests. Paper 243. http://tractortestlab.unl.edu/documents/DX55.pdf

OECD, 2016. Code 2 OECD Standard Code for the Official Testing of Tractor Performance. OECD, Paris, France http://www.oecd.org/tad/code/02%20-%20Code%202%20-%20Final(February%202016).pdf (accessed 15 March 2016)

OMEGA, 2014. Specifications Sheet. OMEGA Engineering Inc. http://www.omega.com/pressure/pdf/PX309-5V.pdf