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Investigation of Petroleum Use in Off-road Agricultural Machinery and Analysis of J1939 Controller Area Network (CAN) Data for Advanced Machinery Testing

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INVESTIGATION OF PETROLEUM USE IN OFF-ROAD AGRICULTURAL MACHINERY AND ANALYSIS OF J1939 CONTROLLER AREA NETWORK (CAN) DATA FOR ADVANCED MACHINERY TESTING

By

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INVESTIGATION OF PETROLEUM USE IN OFF-ROAD AGRICULTURAL 
MACHINERY AND ANALYSIS OF J1939 CONTROLLER AREA NETWORK (CAN) 
DATA FOR ADVANCED MACHINERY TESTING 
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University of Nebraska, 2017 
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Nearly all modern off-road agricultural machinery are powered by petroleum, primarily 
diesel fuel. Equipment manufacturers work to improve fuel efficiency of their equipment 
to provide better value to their customers. Benefits of reducing petroleum use include 
lower operating costs, improved operational efficiency, improved energy security, and 
environmental benefits. Agricultural tractor testing focuses primarily on characterizing 
maximum performance but there is a need for advanced test procedures to better measure 
fuel efficiency for load cases representative of in-field operations. This would help 
farmers make better equipment purchasing and utilization decisions and help 
manufacturers quantify efficiency improvements; ultimately driving continued progress 
toward better efficiency. This research explored historic diesel fuel use for off-road 
agricultural machinery, current trends in improving diesel efficiency, and potential to 
better assess fuel efficiency with advanced test procedures. Additional work evaluated 
the accuracy of machine-reported data that has the potential to be used for creating 
advanced test procedures. Results showed that machine-reported engine speed is a good 
predictor of actual engine speed but there was statistically significant difference between 
measured torque and torque calculated from machine reported parameters. Tools for
collection and analysis of in-field machine data were also explored and an application was developed as proof-of-concept for SAE J1939 data logging and real-time analysis. Each of the software and hardware tools explored demonstrated advantages for individual use cases and provided practical options for J1939 field data collection.

Chapter one is composed of information from a report titled Agricultural Industry Advanced Vehicle Technology: Benchmark Study for Reduction in Petroleum Use that was prepared for the United States Department of Energy and published by Idaho National Laboratory. Chapters two and three are being prepared for publication in Transactions of ASABE (American Society of Agricultural and Biological Systems Engineers) and Computers and Electronics in Agriculture, respectively.
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CHAPTER 1: PETROLEUM USE IN OFF-ROAD AGRICULTURAL MACHINERY

1 Farm Diesel Use in the United States

On-farm diesel use in the U.S. has remained relatively constant since 1985, but decreased slightly in 2009, which may be attributed to price increases and economic recession (Figure 1). During this time, the U.S. harvested area has also remained relatively constant at roughly 300 million acres. In 2010, farm diesel use was 5.4% of total U.S. diesel use. Crops accounting for an estimated 58% of U.S. farm diesel use include corn, soybean, wheat, hay, and alfalfa, respectively, based on harvested crop area and a recent analysis of estimated fuel use by crop (Figure 2)\(^1\). Diesel use in these cropping systems is primarily from tillage, harvest, and various other operations (e.g. planting, spraying) (Figure 3). Diesel efficiency is markedly variable due to machinery types, conditions of operation (e.g. soil type, moisture), and operator variability. Farm diesel use per acre has slightly decreased in the last two decades (Figure 4) and diesel is now estimated to be <5% of farm costs per acre (Figure 5).

This chapter will explore current trends in increasing diesel efficiency in the farm sector. The report combines a review of literature and data analysis to identify nascent technologies for increasing diesel efficiency.
**Figure 1** Historical U.S. diesel consumption: “Farm” includes all on-farm diesel use, and “Off-Highway” includes forestry, construction, and industrial. Source: DOE/EIA Annual Energy Review 2011 Table 5.15.

**Figure 2** Estimated diesel use by harvested crop in 2010 for reduced tillage: A) harvested area by crop. (Note: “other crops” consists of 28 individual crops.) B) estimated diesel use by crop (Source: diesel use for each crop estimated by multiplying harvested area by FEAT efficiencies and subtracted from total fuel use to estimate fuel use for other crops).
Figure 3 Cropping cycle and diesel use per operation for corn with mulch tillage practice.\textsuperscript{4}

Figure 4 Farm diesel use and harvested acres 1993-2010.\textsuperscript{2, 3}
Figure 5 Cost per acre for crop inputs.5
1.1 Tractor Diesel Efficiency

Although most mechanized agricultural operations include a tractor as a primary power unit, the tractor itself is not particularly useful without an implement attached. The implement has a task-specific design, which engages the soil or crop to carry out tillage, cultivation, harvest, and other operations. Most modern tractors provide power for implements via a drawbar, power take-off (PTO) shaft, and/or fluid power hydraulics. Much effort and focus has been directed at tractors because that is where the fuel is consumed to generate mechanical power. Innovations and efficiency improvements in tractor engines, powertrains, and auxiliary power systems have been ongoing since tractors were invented a century ago, and significant gains have been realized.

Specific fuel consumption (SFC) (horsepower-hours per gallon, Hp-h/gal; or kilowatt-hours per liter, kWh L⁻¹) for tractors tested at Nebraska Tractor Test Laboratory (NTTL) from 1958 to 2012 improved by 19.7% for PTO power and 23.4% for drawbar power when comparing data averaged over the last five years of this period versus the first five years of this period (Figure 6). It should be noted that trends based on NTTL data do not necessarily include all tractor models produced by industry (although for tractors sold in the U.S. there would be few exceptions); and minimum tractor power requiring an official test has increased over the years to eliminate the necessity of testing small tractors not intended for use in commercial agriculture (e.g. garden tractors).
Figure 6 Annual average specific volumetric fuel consumption for PTO and drawbar power for diesel tractors. Source: NTTL data.

Increased SFC was observed in tractors with higher PTO power levels, which may be due to parasitic loads being a smaller fraction of gross power as power levels increase (Figures 7 & 8). Reasons for the overall trend in improved SFC for PTO and drawbar operations are not well documented, but contributions include improvements in engine & powertrain efficiency, fuel systems, turbocharging, manufacturing (tighter tolerances, advanced materials, etc.), fuel & lubricants, reduction in parasitic loads (variable fans, closed center hydraulics, etc.), tire design (bias vs. radial tires), and machine setup & operation (optimal ballasting, shift-up-throttle-back, etc.). It should be noted that some high power tractors are intended primarily for high draft drawbar applications, and the PTO may be designed to transmit only a portion of available engine power; therefore, SFC for PTO power may be skewed as it does not reflect full engine power efficiency in every case.
Figure 7: Annual average specific volumetric fuel consumption for PTO power by PTO power level for diesel tractors. Source: NTTL data.

Figure 8: Annual average specific volumetric fuel consumption for drawbar power by drawbar power level for diesel tractors. Source: NTTL data.

Standard test procedures (such as OECD Code 2, Standard Code for the Official Testing of Agricultural and Forestry Tractor Performance) are used to characterize tractor performance but do not evaluate efficiency for in-field operations where loads can vary...
significantly (due to implement design and operator style, as well as, differences in crop, field, and soil characteristics) and have a combination of simultaneous power demands (e.g. drawbar, PTO, and hydraulic). An example of an engine torque curve and associated load points, as measured with OECD Code 2 procedures, along with a theoretical load distribution is shown in Figure 9. The actual distribution of loads for typical agricultural tractor operations is not known. Idle time is thought to be up to 20% to 30% of tractor run time and many processes do not require sustained operation at full load. Remaining operations are a variety of partial loads that are not captured in existing test procedures. If actual load distributions were known, advanced test procedures could be developed to better evaluate loads and related efficiencies that are more representative of in-field operations.

**Figure 9** Discrepancy between existing performance tests (black) and probable in-field load distributions (red).
1.2 Tractor Mechanics

Described below are innovations still under development that are intended to further improve machine efficiency.

**Engine**

Nearly all modern tractors used in commercial agriculture are powered with diesel fuel. Although fuel economy is important to engine and machinery manufacturers, much effort and resources in recent years have been dedicated to meeting exhaust emissions regulations. Engine calibrations have been optimized to reduce exhaust pollutants in accordance with EPA emissions tiers. This was accomplished through several means, including in-cylinder combustion optimization, exhaust gas recirculation (EGR), etc., but did not include exhaust after-treatment (e.g. EPA Tiers 1-3). With the addition of exhaust after-treatment systems for the Tier 4 Interim stage, some engines require diesel exhaust fluid (DEF) to catalyze pollutants in the after-treatment system (e.g. urea), while other after-treatment systems inject diesel fuel into the exhaust stream to regenerate a diesel particulate filter (DPF) that traps particulate matter.

Some manufacturers claim as much as 5% greater fuel efficiency for their Tier 4 Interim engines than that of Tier 3 models. Yet, when evaluating fuel efficiency, variations in exhaust after-treatment systems should be considered due to trade-offs between consumption of diesel and DEF.
With the development of hybrid machines and electric powertrains, some manufacturers are implementing electrically powered, variable speed water pumps to reduce coolant flow with the intent of saving energy and fuel when full coolant flow is not needed. An alternate design has been researched and demonstrated a 1.7% improvement in fuel economy for a clutched two speed water pump (standard drive speed and 65% of that speed) with a planet gear drive and 4% improvement in fuel economy for a clutched on/off water pump when compared to a conventional belt driven water pump. These designs were evaluated on a chassis dynamometer with a test vehicle using the New European Driving Cycle (NEDC). While the effect of these clutched water pump designs is expected to be similar to an electric water pump with regard to improved cooling system performance, Shin, et.al. argues that inefficiencies in the conversion of energy between mechanical and electrical systems would hinder the efficiency of an electric water pump.8

Engines can spend a notable amount of time at idle. A report of on-highway heavy duty diesel engines cites a near doubling of fuel consumption for an increase from 600/750 rpm to 1000 rpm.9 Advanced engine controls are being introduced to reduce fuel consumption by lowering engine idle speeds and even shutting the engine off during extended idle periods. Examples of these strategies are cited in this report and are found in existing patent applications that show intentions of further development in these strategies.10

Some efficiency gains are the result of changes in machine set-up and operation. Proper maintenance (clean filters, correct lubricants), adjustments (proper tire pressure), and ballasting (appropriate weight distribution for the conditions) affect fuel efficiency. One
study suggests that maintaining clean fuel and air filters can provide an average fuel savings of slightly more than 100 gallons annually for a farm tractor.\textsuperscript{11}

Cooling fans can be a significant parasitic load. Therefore, many modern tractors use variable speed or variable pitch fans that continually adjust to provide only the cooling needed and avoid unnecessary parasitic loads. However, if the radiator is not clean, the fan is not effective, and coolant temperature remains high; therefore commanding a higher fan speed. One cooling fan manufacturer reports, “A marginal increase in fan speed of 600 rpm due to a clogged radiator leads to a doubling of necessary fan drive power. If the fan drive power rises from 9.5 to 19 kW, the resulting fuel consumption increases to about 0.92 gph (3.5 liters per hour).”\textsuperscript{12} Simply maintaining a clean radiator can have a direct impact on fuel consumption.

**Waste Heat Recovery**

Air conditioning systems powered by waste heat from engine exhaust gas and EGR coolers have been studied and are claimed to be capable of reducing fuel use and engine idling.\textsuperscript{13} ClimateWell’s Verdacc heat-driven air conditioning system is advertised to be available for new vehicle designs and retrofit for existing vehicles. According to their claims, this technology “makes it possible to reduce fuel cost used for cabin comfort by up to 90%.”\textsuperscript{14}

A patent filing exists regarding a device for recovering energy from an engine’s exhaust stream with an electric machine that may consist of a generator or motor/generator that
may be part of a turbo charger. The recovered energy may be stored for later use or used directly to power the engine or machine functions and therefore improve the overall fuel efficiency of the system.¹⁵

Behr has demonstrated a prototype waste heat recovery system that showed up to 5.2% efficiency improvement on a test rig. Their system, shown schematically in Figure 10, is conceptually similar to a small steam engine that converts thermal energy from engine exhaust into mechanical power that can be used directly or stored for later use. Efficiency gains were highest for the steady state portion of their long-haul truck test cycle.¹⁶ For agricultural operations, these energy recovery and efficiency gains for steady state operation may lend themselves to applications such as tillage with sustained heavy draft loads.

Figure 10 Prototype waste heat recovery system developed by Behr¹⁶
**Powertrain**

Traditional discrete gear transmissions provide a number of manually selectable gears for low speed, high draft field operations and for high speed, low load transport operations. While these transmissions provide an adequate range of operation for typical farm activities, their manual operation, and sometimes large step ratios, can force the engine to be operated at suboptimal speeds with respect to efficiency.

Shift-up-throttle-back operation is known as an effective way to improve fuel efficiency. Procedures used by NTTL (based on OECD Code 2) include a reduced engine speed sequence at various drawbar loads that demonstrates the benefits of shift-up-throttle-back operation. For a load case of 75% of pull at maximum drawbar power, shift-up-throttle-back operation typically results in 5% to 15% reduction in fuel consumption while still producing the same drawbar power. For a load case of 50% of pull at maximum drawbar power, shift-up-throttle-back operation typically results in 15% to 30% reduction in fuel consumption while still producing the same drawbar power. Even though the benefit of shift-up-throttle-back operation is clear, it can be difficult for an operator to manage the throttle setting and gear selection for constantly changing loads during field operations.

As powertrain designs evolve, additional gears with smaller step ratios have been added to transmissions to narrow the required engine operating range. By properly matching the engine and powertrain, and with the aid of advanced controls, the engine can operate in a relatively narrow range where it is most efficient.
Many transmissions require pressurized oil for clutch actuation and lubrication. Although the minimum required oil pressure may be different for each set of clutches that are actuated, and for the torque being transmitted at a given time, transmission oil systems often maintain constant pressure required for worst-case loads. A portion of this oil may be directed through a pressure reducing valve to lower the pressure for lubrication. Energy required to pressurize the portion of flow used for the low pressure lubrication circuit may be converted to heat as the pressure is reduced. Additional energy may be lost if the cooling fan has to run faster to reject this additional heat from the cooling package. Some manufacturers have separated these circuits to avoid pressure drop and associated energy loss for the lubrication flow. A patent exists for a strategy to regulate transmission charge pressure in order to reduce pressure when it is not needed; for example, to regulate to high pressure for transport and regulate to low pressure for stationary operations when transmission clutches are not engaged.

John Deere’s 24 speed dual-clutch transmission offered on their 6R series of tractors is expected to provide a “4 percent cut in fuel” or a savings of up to 10 grams of diesel per kWh compared to an IVT transmission. Although a manual mode is available that will allow the operator to select the desired gear, it is the enhanced controls of the automatic mode that leads to efficiency optimization. In automatic mode, the machine optimizes efficiency by selecting the appropriate gear to keep the engine in an efficient operating range for the desired speed and load. Deere is also implementing their e23 transmission on their 7R and 8R Series tractors with 23 forward and 11 reverse speeds. Much like the 24 speed transmission described above, the e23 also offers a control feature that manages
the transmission for best fuel economy by automatically shifting up and throttling back while maintaining an operator selected ground speed.22

As more gear ratios are added to a transmission, and step ratios become smaller, the opportunity to maintain operation at peak engine efficiency can grow. This concept leads to the evolution of a step-less transmission with an infinite number of ratios. Infinitely (or continuously) variable transmissions (IVTs or CVTs) allow the engine to work in a narrow, yet highly efficient, operating range while still providing full range of speed and torque to the powertrain. A number of CVTs have been marketed, and manufacturers are claiming notable fuel savings. Although mechanical power transmission is more efficient, it is the continuous variable characteristic of hydraulic and electric powertrains, and advanced integrated engine controls, that lead to overall improved system efficiency.

Several manufacturers have developed hydro-mechanical CVTs. A study by Howard has shown that for partial load conditions (loads below 76% - 81% of maximum drawbar power at respective speeds) a CVT was more fuel efficient than a discrete gear transmission operated at full throttle; however, it was less efficient for loads near max power. When a shift-up-throttle-back strategy was used, the gear transmission had significantly lower fuel consumption at power levels 37% - 52% of maximum drawbar power at respective speeds. Howard’s study indicated that, in general, the CVT was more efficient than the full throttle operation of the gear transmission, but the gear transmission with shift-up-throttle-back operation was more efficient than the CVT in the load and speed range tested. The gear transmission is inherently more efficient at transmitting
power as it lacks certain parasitic losses that accompany the CVT, but at the system and machine level, the CVT and system level controls can achieve efficiency improvements for some load conditions.\textsuperscript{23}

CNH claims their Puma Series Tractor can achieve as much as 25% reduction in fuel use when equipped with a CVT transmission and Diesel Saver Auto Productivity Management (APM). This integrated control system maintains an operator selected working speed at the most efficient operating points by automatically adjusting the engine and transmission.\textsuperscript{24}

Machine controls react to external loads but cannot anticipate future loads. By the time an engine or powertrain system reacts to a load, the event may have passed and a new condition is present that requires the machine to operate in a different way for optimum efficiency and performance. In the future, GPS technology may play a role in advanced powertrain controls for improved efficiency and performance. A patent for control of vehicular systems based on geo-referenced maps gives consideration to the idea that using geo-referenced data to anticipate operating conditions can improve efficiency and performance for tractors, combine harvesters, sprayers, and other agricultural machinery by preemptively adjusting transmission ratio, differential locks, and other machine settings prior to changes in slopes, crop conditions, and soil conditions. This concept may also be applied to hybrid systems where, for example, energy storage could be managed in anticipation of an upcoming downhill slope; energy currently stored can be used up prior
to reaching the start of the downhill slope to free up storage capacity for energy that can be recovered while traveling down the slope.\textsuperscript{25}

Modern examples of electric CVTs are described below:

The Belarus 3023 (Figure 11) tractor’s electro-mechanical drivetrain, with a 300 hp engine powering a 220 kW generator, is claimed to have a fuel consumption reduction of 15\% to 20\%.\textsuperscript{26} Optional equipment includes an electric cooling fan for the radiator, an electromechanical front PTO shaft that can operate at speeds independent of engine speed, and an autonomous electric power station for 172.5 kW of auxiliary power.\textsuperscript{27}

\textbf{Figure 11} Belarus 3023 tractor with electro-mechanical powertrain.\textsuperscript{28}

The Rigitrac EWD120 tractor (Figure 12), a project of the Technical University of Dresden and supported by Rigitrac Traktorenbau AG and other companies, includes independent electric wheel motors powered by a diesel-engine-driven generator. This
generator can also provide electric power for external implements. Because of the continuously variable characteristic of the electric powertrain, the diesel engine always works in its “best fuel consumption map,” and prototype documentation states that “the electric drive system has a higher efficiency than the conventional drivelines.” Electric brakes and braking resistors allow braking energy to be converted to “useful heat,” (however, no description is provided on how this recovered energy is utilized). No advertised claims regarding improvements in fuel efficiency have been found, although one source stated “the transmission runs at 85% efficiency, compared with the 65%-70% of the hydrostatic units used on Rigitrac’s standard machines.”

![Figure 12 Rigitrac EWD120 tractor with electric powertrain.](image)

An electric hybrid powertrain, called the ZF TERRA+, was developed by ZF Friedrichshafen AG for agricultural tractors and self-propelled harvesters (Figure 13). A starter generator module integrated into the transmission provides electric energy for auxiliary functions and implements. With the addition of a battery, the ZF TERRA+ can
become a hybrid system able to recover braking energy, provide short-term engine load relief, and allow the engine to work in a more efficient operating range; this system includes an electric transmission-oil pump. This company claims that when used in a tractor “the optimum operation of electrified auxiliaries can achieve consumption benefits of about 5% on the average.” According to ZF, the ZF TERRA+ can be used to power electric drives on implements with improved efficiency and has up to 10% more power available as compared to conventional hydraulic drives.

![Figure 13 ZF TERRA+ starter generator in combination with the continuously variable S-Matic transaxle.](image)

A patent application for a “Tractor with Hybrid Power System” outlines a concept for a diesel-over-electric powertrain that includes energy recovery with battery storage. The concept also includes provisions for electrically powered variable speed PTO and an electrically powered hydraulic pump. The patent application stated that simulations of this hybrid system show 5% to 20% fuel savings depending on the type of implement used with the tractor.
AGCO has developed a prototype electrically-powered high-clearance sprayer that is claimed to deliver 35% more power to the ground than its conventional counterpart with the same engine. AGCO says the experimental ElectrRoGator 1386 electric sprayer (Figure 14) has 36% higher torque, 6% more power, 20% better fuel efficiency, and better performance than their conventional 1386 sprayer. On the prototype, braking energy is dissipated as heat through resistor grids, but work is being done to recover and store this energy for use by other machine functions. AGCO indicates that combine harvesters are also good candidates for electric drive technology where they can benefit from the improved efficiency and independent control of multiple machine functions.\textsuperscript{37}

\textbf{Figure 14} AGCO ElectRoGator 1386.\textsuperscript{37}

In 2016 John Deere revealed an all-electric concept tractor (Figure 15) equipped with two 150 kW electric motors powered by 130 kWh capacity battery packs.\textsuperscript{38} The independent motors are used for propulsion and to power PTO and hydraulic systems. Full battery charge can work about four hours in normal use.\textsuperscript{39}
Electric and hybrid powertrains have been developed for several models of off-road industrial and construction equipment introduced for sale. Equipment types include crawler dozers, wheeled loaders, and excavators.

Caterpillar’s D7E, a crawler dozer, introduced for sale in 2009, has a diesel-over-electric drivetrain for primary propulsion and electric accessories including the AC compressor and water pump. Fuel savings of 10%-30% are being advertised, with some customers reporting larger savings in certain applications. Advertised claims also include 25% more material moved per liter of fuel. Fuel use can be reduced further on machines configured with an engine idle shutdown timer that can shut the engine off after a predetermined period of time at idle.
Deere is marketing a 644K hybrid electric wheeled loader with claims of up to 25% average reduction in fuel consumption (based on Deere’s internal tests, including stockpiling, truck loading, and transport). Although this machine does not have energy storage capability, the diesel-over-electric powertrain is capable of recovering energy that can reduce engine load and fuel consumption.

Komatsu’s H205-1 hybrid excavator is claimed to have fuel savings of 25%-41%. Kinetic energy is captured during braking and is stored in an ultra-capacitor to be used for power assistance during the next machine cycle. Komatsu’s literature suggests that the success of their hybrid system is due in part to the ability of ultra-capacitors to charge and discharge quickly for the high energy demand of the excavator, whereas battery storage would be too slow to be effective for this application. Hyundai reportedly has also been working with ultra-capacitors for hybrid excavator designs.

Ricardo has developed a hybrid excavator demonstrator to show the benefits of their flywheel energy storage system and is claiming “10% fuel savings overall, with 30% fuel savings in some duty cycles.” The composite flywheel, rotating at 60,000 rpm in a vacuum, can store 0.25 kW-h and deliver up to 28 Nm of torque. Ricardo is developing other flywheel storage models up to 1.25 kW-h. With 65% lower cost than battery hybrid systems, and still delivering 80% of the fuel economy, the flywheel hybrid system may be a good fit for machines with low volume production.
Caterpillar claims its 336E H hydraulic hybrid excavator uses up to 25% less fuel than a standard 336E with up to 50% greater efficiency in terms of tons of material moved per gallon of fuel.\textsuperscript{45} The hydraulic hybrid design captures braking energy in hydraulic accumulators that is used to accelerate the machine in subsequent machine cycles. In addition to the hydraulic hybrid design, contributions to fuel savings also come from advanced engine controls, including engine idle shutdown and on-demand engine power that reduces engine speed when less power is needed.\textsuperscript{46}

A characteristic shared by these machines, which makes them good candidates for hybrid drive and energy recovery systems, is that they are cycling machines, meaning segments of their typical work cycle involve dissipation of kinetic energy during direction changes, braking/deceleration, lowering of implements, etc., to stage the machine for the next work cycle. Many agricultural field operations (e.g. tillage, planting, and harvesting) are primarily steady state with infrequent cycles and therefore have limited opportunity to recover kinetic energy. However, there may be potential for energy recovery strategies during braking/deceleration events for self-propelled chemical applicators or tractors utilized for transporting material.

**Remote Power (Hydraulic, Mechanical, Electric)**

Early tractors were essentially mechanical replacements for horses with the primary purpose of providing drawbar power to pull implements through the field. As tractors and farm machinery evolved, tractors took on additional capabilities to provide other types of power and controls for implements. Early belt drives were eventually replaced with a
power-take-off (PTO) shaft. A PTO consists of a rotating shaft to transmit mechanical power from the tractor to the implement at a standard speed of 540 rpm or 1000 rpm. Generally, PTOs operate at a fixed ratio of engine speed, although some modern tractors include an economy PTO mode that allows PTO operation at a reduced, but still fixed ratio of, engine speed for improved fuel efficiency. For example, PTO tests for a John Deere 5115M show fuel efficiency improvement of 7% to 24%, depending on load, when the PTO is operated on economy mode (540 rpm PTO speed at 1645 rpm engine speed) versus normal mode (540 rpm PTO speed at 2100 rpm engine speed).47

Patents exist for variable speed PTO drives that can provide constant speed PTO at reduced or varying engine speeds. The variable speed concept allows for shift-up-throttle-back transmission operation while still maintaining constant PTO speed and also enables controlling PTO speed as a ratio of ground speed, which may improve the efficiency of certain PTO driven implement operations. If the tractor configuration is such that the hydraulic pump drives are downstream of the variable PTO drive, this can enable the tractor’s hydraulic pump(s) to be driven at constant speed regardless of engine speed and therefore maintain hydraulic system performance at reduced engine speed.48 Another patent application describes the potential to use a similar variable speed PTO system; in this case, the PTO speed may be defined by GPS mapping and/or feedback from implements via ISO-bus with the goal of improving tractor and implement performance and efficiency.49 Although PTOs have high power density and efficient mechanical power transfer, the rigid nature of the mechanical PTO shafts can also be difficult to package in new implement designs.
Hydraulic systems were later added to tractors to control basic implement functions such as raising and lowering implements and later rotary power for functions such as fans, augers, etc. Early hydraulic systems were quite simple, with fixed displacement pumps that operated a single function. As mobile fluid power systems evolved, they increased in complexity, and also in capability and efficiency, with the ability to control multiple functions simultaneously at variable pressure and flow. Modern pressure and flow compensated hydraulics have high power density and variable control for multiple functions. A modern implement may have many hydraulic functions with simultaneous fluid power demands. A planter, for example, may have hydraulic functions for raising/lowering, folding for transport, folding markers, row unit down-pressure, vacuum fan motors, and fluid pumps.

A study by the Oak Ridge National Laboratory and the National Fluid Power Association concluded that across all industry sectors, of which agriculture is the second largest segment comprising 21.2% of total fluid power component sales, fluid power system efficiencies range from less than 9% to as high as 60%, with an average efficiency of 22%. Considering cumulative losses (including the transfer of power from the engine to the hydraulic pump, pump efficiency, line losses, valve losses, etc.) typical mobile hydraulic systems have an overall efficiency of approximately 14% (Figure 16).
Figure 16 Energy losses in mobile load sensing (LS) hydraulic application.\textsuperscript{50}

Typical modern hydraulic systems for mobile equipment have one or more hydraulically controlled variable displacement pumps to meet the demand of combined flow for all actuators in the system and also the pressure demanded by the function requiring the highest pressure. While this satisfies requirements of the function with the highest pressure demand, pressure for other functions has to be reduced to a level required for each respective load. Reduction of pressure occurs when the fluid is throttled through control valves, which achieves the desired result of providing pressure and flow appropriate for each function. However, with the exception of the function operating at highest pressure, large energy losses occur due to the pressure drop for the remaining loads, as illustrated in Figure 17. Energy is lost as heat, which can drive additional energy consumption from a cooling fan if this heat is dissipated through the machine’s cooling system. Additionally, this hydraulic system configuration does not lend itself to effective energy recovery schemes, such as recovering kinetic energy from braking or recovering potential energy as actuators are lowered.\textsuperscript{51}
Research at Purdue University’s Maha Fluid Power Center includes work on high efficiency mobile hydraulics. Significant fuel savings have been demonstrated on a hydraulic test bed machine that was used to study throttle-less hydraulic actuation. Meter-less flow architecture allows flow from several smaller pumps to be paired with each individual actuator, thus eliminating the main control valve, which, as shown in Figure 16, is the location of greatest energy loss. The only control element is the pump displacement, and the unit automatically moves over-center to allow energy recovery. With this technology, hydraulic control valves are eliminated and control is instead achieved with displacement control or pump-controlled actuation. This technology is expected to provide fuel savings for multi-actuator machines used widely in construction, agricultural, and forestry industries with significant fuel savings already demonstrated on the test bed. Additional work is planned for a hydraulic hybrid configuration with a goal of 50% fuel savings.
savings with no degradation of performance. This hydraulic actuation technology has been demonstrated on a wheel loader where measurements showed 20% higher fuel efficiency. Independent side-by-side testing at a Caterpillar facility demonstrated 40% fuel savings over the standard machine.51

The Digital Displacement pump is another fluid power technology that shows promise for improving hydraulic system efficiency especially for part load conditions. A developer of this technology has claimed that replacing proportional valves and load sensing pumps in traditional fluid power systems with this digital displacement technology can provide double digit energy savings for off-road vehicles.52

Hydraulic transformers for use with common pressure rail hydraulic systems are also under development with the hopes of overcoming inefficiencies of the current state-of-the-art load sensing systems and metering valves. This concept eliminates the throttling losses of metering valves and is expected to have higher efficiency at part load than that of a load sensing pump. Market readiness of this concept and its efficiency are not known.53

PTO and hydraulic power systems are well established and effective for today’s applications, however, the search for more versatile and efficient power transfer continues. One alternative is electric power, which first debuted in 1954 on the Farmall Electrall tractor. The IH Farmall 450 had an integrated electric power generator that provided up to 10 kW of electric power for implements. Limited in part by availability of adequate electrical controls, the technology was not adopted at that time.54
A recent source suggests electric drives would be suitable and beneficial for 28 drives on modern agricultural machines, 45% of which currently use hydraulic power and 55% which are mechanically driven. Electric power enables variable speeds control so functions can operate independently of engine speed and use only the power and energy needed for the given function. This tends to be more efficient than power transmission via a PTO shaft or hydraulics. Tractor accessories that can be electrically powered include engine cooling fan, air brake compressor, air conditioner compressor, engine water pump, and hydraulic pumps. The alternator and starter can be eliminated in some cases if on-board electric power is available from a starter generator. ISOBUS, a communication protocol for controller networks on agricultural machinery, is a key enabler for advanced controls that take advantage of torque and speed control capabilities of these electric systems. ISOBUS can be for control of tractor mounted accessories or communication and control between tractors and implements.

According to ZF, the ZF TERRA+ starter generator (described on pages 27-28) can be used to power electric drives on implements (shown schematically in Figure 18) with improved efficiency and up to 10% more power available as compared to conventional hydraulic drives. ZF also offers a 50 kW PTO driven generator as a retrofit to provide electric power for implements. This unit can provide various voltage output configurations for independent control of two 25 kW electric drives.
John Deere has also designed for mobile electric power with two 7030E Premium Series tractors introduced in 2007 and the 6210RE in 2011; all of which are intended for the European market at this time. This is a more efficient alternative to PTO or hydraulic power transfer and can offer electric power for implements. The tractor can include outlets to power electric tools and other devices. Adoption of these tractors has been limited in part by the lack of availability of electrically powered implements. Another barrier to further implementation of electrically powered farm implements has been standardization of generator/implement voltage and electric connections for the tractor-implement interface. This is being evaluated by the Agricultural Industry Electronics Foundation (AEF) who is working to establish engineering standards for electrical components on farm equipment. Replacing hydraulics with electric systems could displace petroleum-based fluid used in fluid power systems; however, power density for typical hydraulic systems (W/kg and W/m^3) can be more than an order of magnitude greater than that of
electric systems, making it difficult to transition to larger and heavier electric components needed to deliver the same performance. “A comparison test with a John Deere 7530 Premium showed that the 7530 E-Premium produced fuel savings of up to 13.8 percent when pulling a power harrow and 9.3 percent when pulling a trailer. In a DLG-PowerMix test, the 7530 E-Premium saved up to 5 percent compared to conventional tractors, including John Deere 7530 Premium, Deutz-Fahr Agrotron M 640, Fendt 820 Vario TMS, Case IH Puma 180, and New Holland T7040.”

Fendt developed a concept tractor built from their 700 Series model that includes a 130 kW generator to power auxiliary functions on the tractor and provides electric power for implements. The Fendt concept uses DC voltage, while Deere is using AC power for implements. Standardization of electric power for implements will be important for getting this technology to market and having compatibility with third party implement designs.

A number of implement manufacturers have investigated electrification of implements to improve efficiency and productivity. Like electrified tractors, electrified implements have been primarily developed in the European market. A survey of Austrian manufacturers showed the following characteristics for electric drives:

### Advantages:
- Controllability
- Easy torque and speed measurement
- Possibility for fault finding
- Ease of distribution of power
- High efficiency
- Overload capability
- Low noise level

### Disadvantages:
- High mass
- Space requirements
- Cost efficient standard components currently not available
- Robustness
- Safety requirements
- Interface problems
- Storage technology
Fliegl Agrartechnik GmbH is touting electrically powered axles on trailers as a way to improve material handling/transport efficiency (as shown in Figure 19). Due to design complexities and lack of adequate controls to adjust trailer ground speed to match that of the tractor, mechanical PTO power transfer to drive trailer axles has not been practical. Although these obstacles can be overcome with hydraulic power, it is a low efficiency alternative. Electrically driven trailer axles are easier to implement and are becoming practical with the introduction of power generators on tractors. This manufacturer claims that an electric system would be 65%-75% efficient as compared to the 25%-55% efficiency of a hydraulic system. Powered axles on trailers can allow a smaller tractor to transport a greater amount of weight, enabling the reduction of tractor ballast, which means energy is not expended in carrying an unproductive mass of ballast.

Figure 19 Trailer with electrically powered traction axle.
Kinze’s 4900 planter is available with electrically driven seed and insecticide metering for high accuracy variable rate application of crop inputs. Other conventional drive options include hydraulic or ground drive configurations.\textsuperscript{61} The electric drive configuration is powered by a hydraulically driven generator mounted on the planter.\textsuperscript{62} Although no claims were found regarding reduction in fuel use or improved efficiency, this is an example of electrification of implements in the North American market, which currently does not offer tractors with onboard electric power. Kverneland Group (Norway) has also developed a variable rate precision seed drill where each planting unit is controlled independently with electric motors.\textsuperscript{63} Graham Equipment has developed an Electric Planter Drive kit to retrofit existing planters.\textsuperscript{64}

Amazone has developed electric drives for precision seeding and a fully electric chemical applicator (Figure 20). Each major function on the UX eSpray trailed sprayer is powered independently with electric motors. Independent, fully variable control allows each function to be operated at optimum speed and load for the given conditions without being tied to tractor engine speed.\textsuperscript{65}
Kuhn/Rauch has developed an electrically powered twin-disk fertilizer spreader that enables variable control of disc speed for more uniform fertilizer application. A study of the Rauch “Axis” spreader showed that an electric drive configuration was most efficient and had the best tractor/implement fuel economy when compared to two hydraulic drive configurations and one mechanical drive configuration on this same model spreader. One source reports Claas is investigating the use of electric drives for self-propelled forage harvesters and combine harvesters. Patent filings by other parties further indicate investigation of agricultural machinery electrification. A patent for electrically powered
hay baler claims energy savings by optimizing machine functions for crop conditions (Figure 21). A patent exists for a combine harvester with integrated electric motors powering grain threshing and conveying functions, as well as drive wheels, but makes no claims on efficiency gains (Figure 22).

**Figure 21** John Deere patent for round baler with electrically driven rollers

**Figure 22** John Deere patent for electrically driven threshing cylinder
Electrification of implements has the potential for energy savings through more efficient power transfer and savings from the precision agriculture perspective through more efficient and effective application of seed and chemicals. Parallel development of electrified tractors and implements, along with standardization of electric power protocols and hardware, will be key to further development of electrification of mobile agricultural equipment.67

**Tires and Tire Pressure**

Since their first use on agricultural tractors in the 1930’s, pneumatic tires have demonstrated significant benefits over steel wheels. A 1934 evaluation of steel wheels vs. pneumatic tires on a WC Allis-Chalmers tractor showed a 45% fuel economy advantage for the rubber tires. Since that time pneumatic tires have become the dominant traction device for agricultural tractors and implements. Early bias ply tires have been replaced in many applications with more efficient radial tires, with one source indicating radial tires have a 6% to 14% advantage over bias ply tires in traction, fuel efficiency, and reduced wheel slippage.71

Improvements in tire performance continue to be realized as tire technology evolves. Michelin is indicating up to 25% fuel saving for their new Ultaflex technology agricultural tires that operate at lower than standard pressures72 (although the baseline for this comparison is not clear). Firestone is indicating as much as 4% fuel savings for tractor tires with their AD2 Advanced Deflection Design compared to standard equivalent-size radial tires.73
While tractors are primarily designed to operate off-road, ground conditions vary significantly. A tractor set up for optimum performance for tillage in sandy soil may have reduced efficiency when pulling a chemical applicator in heavy clay soils. Two variables on the tractor that affect this performance are tire pressure and ballast. Both parameters have been studied and show there is no single setting for all conditions. For example, proper ballast is achieved when the tractor has approximately 10% wheel slip when working on soil. Optimum ballast configuration changes as soil conditions and implement loads change. Performance related to tire pressure is similar, optimum tire pressure for one set of conditions is inefficient for others.

A field demonstration showed an 8% average fuel savings for a four-wheel-drive tractor when operating with recommended tire pressures versus over inflated tires. A mechanical front-wheel-drive tractor showed an 11% improvement in field capacity and 26% improvement in fuel efficiency using tire pressure recommendations as compared to over inflated tires. Although this field demonstration was not a scientific study, it clearly indicates potential fuel savings with proper tire pressure. An on-farm study with self-reported data from four four-wheel-drive tractors doing fall tillage showed 5.3% to 10.3% improvement in fuel efficiency for properly inflated tires versus over inflated tires.74

Effects of tire inflation pressure were studied for a tractor towing a trailer on paved roads. The test was repeated at several load levels with three different tire inflation pressures, each of which was within the recommended inflation pressure range indicated in the tractor operator’s manual. Results clearly showed benefits of having the correct tire
pressure for the operating conditions, with up to 11.4% improvement in fuel economy (miles per gallon in this case) for the highest inflation pressure as compared to the lowest inflation pressure. This study included only transport operations on paved roads and therefore did not have data showing likely benefits of low inflation pressures for off-road farm operations.

A central tire inflation system (CTIS) marketed by Spicer is claimed to improve fuel efficiency by up to 3.3% and increase tire life by up to 10%. This system includes an on-board air compressor and pneumatic control unit to allow the operator to increase tire pressure for travel on hard surface roads and decrease tire pressure for work in softer soils.

Fendt is offering a factory installed tire pressure regulating system on their 900 Vario series of tractors with claims of up to a 10% fuel savings. The system is also capable of adjusting tire pressure on the tractor as well as towed implements. Fendt indicates the system is capable of increasing tire pressure by one bar (14.5 psi) in ten minutes and reducing by one bar (14.5 psi) in two minutes via command from an electronic operator interface in the cab.

Patents for central tire inflation systems owned by major industry OEMs (original equipment manufacturers) indicate interest in these systems for series production. Existing patents include systems not only for tires on tractors and prime movers but also for implements where inflation pressure may be managed based on predetermined soil maps and GPS position in real time or closed loop control based on soil characteristics or
dynamic axle loads measured in real-time.\textsuperscript{78} Patents include combine harvesters where gross machine mass, and therefore tire loads, change during operation as harvested material is accumulated in an on-board storage bin or with different headers.\textsuperscript{79} This principle could also apply to chemical applicators where gross machine weight changes as the payload is dispersed.

Literature reviewed regarding central tire inflation systems suggested that, in addition to reduced fuel use and longer tire life, other benefits include increased crop production as a result of reduced soil compaction, increased traction/reduced slippage, improved operator comfort, and improved stability when driving on hard surface roads.

A patent filing exists for a concept involving dynamically changing a vehicle’s footprint during operation (described as a virtual-foot, or v-foot). Similar to the central tire inflation systems, claimed benefits include improving fuel efficiency by reducing rolling resistance and reduced soil compaction by increasing ground contact area, but also improved stability. However, in this case, the premise is to change the shape of the ground contact patch by one of several means, including changing the shape, stiffness, volume, or other properties of wheels, tracks, or tires by using polymers, magneto-rheological or electro-rheological materials, changing gas pressure, changing a circular wheel into a generally triangular track, or changing width of ground contacting elements. The virtual-foot concept can include closed loop control, based on geographic zones with desired vehicle characteristics based on topography or soil characteristics.\textsuperscript{80} Although improved fuel efficiency is claimed, this claim is not quantified.
Galileo Wheel Ltd. has developed a tire with an “accordion-shaped” folding sidewall (Figure 23) claimed to allow change in foot print area while maintaining consistent ground pressure distribution and fuel savings through reduced rolling resistance and reduced slippage. At low tire pressures the sidewall folds, as opposed to a standard tire where the sidewall deforms and causes increased rolling resistance and generation of heat.  

![Galileo wheel with accordion-shape sidewall design.](image)

Figure 23 Galileo wheel with accordion-shape sidewall design.  

There are clear benefits to having proper tire inflation pressures, but it is not clear if there is a net reduction in energy use when energy to power an air compressor is included. Data comparing energy consumed by tire pressure management devices (e.g. air compressors and associated hardware) versus energy saved by having optimally inflated tires was not found.
1.3 Implement Operations—Secondary Fuel Use

Although tractors are seen as the primary energy users because that is where fuel is consumed, implements drive the energy required by the tractor. While it is certainly justified to expend effort and resources on improving tractor efficiency, a tractor’s primary purpose is to provide power to implements where work is done on the soil or crop. By improving implement efficiency a direct reduction in energy use is achieved. Tillage and harvesting operations have been shown to typically require the most energy (Figures 3 and 24). Although some research has focused on implement efficiency, reports related to engine, powertrain, and overall tractor efficiency are far more common.

Reports on tillage implements equipped with sweeps and rolling coulters indicated the configuration of ground engaging components could be arranged to reduce draft requirements by up to 45% during field tests;\textsuperscript{82} note that this does not involve design changes to the ground engaging tools, but simply changing their location and relative position on the implement frame. While it is not clear if this dramatic reduction in draft is achievable on a commercial scale or in all soil types, it does demonstrate potential for energy reduction requirements through optimization.
Tractor and implement designs have evolved over time becoming more productive and reliable, but performance of the tractor-implement system has largely been determined by the skill and experience of the operator. With modern electronic controls being applied to this equipment, tractor-implement-automation is becoming feasible. Tractor-implement-automation via controller area network (CAN), using ISOBUS communication protocols, can allow an implement to dynamically request changes in tractor parameters. This has been demonstrated on the John Deere 6030 Premium series tractors with a John Deere 864 round baler and with a Pöttinger Jumbo Loader Wagon, where the implements are allowed to request control of tractor parameters including travel speed, acceleration rate, and hydraulic and PTO controls. In the case of tractor-implement-automation applied to the
round baler, as the rate of material entering the baler changed, as detected by real-time sensors monitoring the volume of hay in the incoming windrow or by approximated PTO torque determined by the tractor, the baler requested changes in travel speed and the acceleration rate of the tractor to provide uniform flow rate into the baler. When the baler sensed the chamber was full, a request was sent to the tractor to stop, and the bale was automatically tied. Once the tying operation was complete, hydraulic controls were requested to activate and eject the finished bale. Benefits included increased productivity, avoidance of plugging the baler, reduced idle time, and more consistent bales. At this time, only implements “certified” by the tractor manufacturer can request control of select tractor parameters because some proprietary messages are used that are not specified in ISOBUS Class 3. ISOBUS Class 3 is the highest level of ISOBUS capability that allows the tractor electronic control unit (ECU) to receive and execute commands that are sent from implement ECUs. In the case of the John Deere 6R series tractors, tractor-implement-automation appears to be available in the European market but is not yet available in North America.83

With no ability for an implement to control tractor hydraulics directly, some implement designs include auxiliary hydraulic valves installed on the implement itself where they can be controlled locally. In this case, the tractor hydraulics may be active continuously so fluid power is immediately available to the auxiliary valves to provide local control of implement functions. Maintaining continuous activation of the tractor hydraulics and throttling oil through a second set of valves can result in significant energy losses and heating of the oil. Additional parasitic losses may result if the tractor’s cooling fan is
required to run faster to cool the hydraulic oil. Better integration of tractor and implement controls may reduce these inefficiencies.

Improvements in combine efficiency include optimizing material flow through the threshing unit, reducing overall machine weight, and including advanced controls that adjust machine parameters to maximize operating efficiency. Patents granted to major OEMs show that consideration has been given to the electrification of combine harvesters, but this has not yet shown up in the marketplace.

1.4 Conclusions

Gradual increases in fuel efficiency have been measured since 1958 (Figures 6-8). Equipment manufacturers have responded to customer needs by investing in improved efficiency, driven in part by high fuel prices, but an improved efficiency measurement practice is needed. Equipment manufacturers are pursuing efficiency improvements from single components to the farming-system level. Many design concepts show promise at their respective levels and in a specific context, but effects at the machine and farming system level may not be captured with current industry-standard test procedures designed for verifying manufacturer’s claims (e.g. OECD Code 2). During a September 2013 meeting of the U.S. OECD coordinating committee, OEM representatives reinforced the need for evolution of test procedures, with an example of evaluating tractors with CVT/IVT powertrains. Current tests evaluate performance at full-load and steady-state operating points, thus capturing the operating envelope of the tractor; however, current
practice does not represent in-field operations where loads often vary significantly with little sustained operation at full load. Current test procedures may show good efficiency for a traditional powertrain at full load, but may show low efficiency for a CVT/IVT powertrain under that same condition (section 1.2 Tractor mechanics: Powertrain). The inverse is likely true for part load conditions, which are likely more representative of in-field operations. Test procedures more representative of field operations, comprised of simultaneous and varying draft, hydraulic, and PTO loads, are needed to better measure tractor efficiency and offer customers and manufacturers data needed to choose the most efficient equipment.

Tractor fuel use is primarily dependent on the operation of implements, such as tillage and planting equipment. A direct reduction in energy use is achieved by improving implement efficiency or substituting implement types. Tillage and harvesting operations typically require the most energy (Figures 3 and 24). Yet, reports related to engine, powertrain, and tractor level efficiency are far more common than those regarding efficiency of implements and harvesters. Potential to increase efficiency exists for 1) selection and optimization of ground engaging tools to reduce draft, 2) improved power transfer to implements, and 3) in-field equipment utilization (Section 1.3). Little information has been found on harvester innovations related to petroleum reduction. More research is needed on diesel use associated with practical changes in implements and harvest equipment.
CHAPTER 2: EVALUATION OF THE ACCURACY OF MACHINE REPORTED CAN DATA FOR ENGINE TORQUE AND SPEED

2 Introduction

To create advanced test procedures that are more representative of field operations, data from actual field operations are needed to determine magnitude, duration, and combination of loads to apply with test equipment. The most direct way to measure in-field loads is by installing analog instruments on tractors and implements. For example, a load cell and radar to measure drawbar pull and ground speed, torque and speed sensors to measure PTO power, and flow meters and pressure sensors to measure hydraulic power. Some disadvantages associated with using analog sensors for in-field measurements include:

- Analog sensors and data acquisition systems are expensive and time consuming to install, especially if they need to be replicated on many tractors and implements.
- Analog sensors need to withstand shock, vibration, dust, moisture, and sometimes harsh temperatures of the off-road equipment operating environment.
- Analog sensors can be a hindrance for the tractor operator.
- Equipment configurations may need to be altered to accommodate sensors.
- Travel to the equipment is required for sensor installation and maintenance of the data acquisition system.
- Analog data may have to be merged with data from other sources including tractor and implement Controller Area Networks (CAN) and Global Navigation Satellite Systems (GNSS).
Modern off-road equipment use onboard sensors and computers which broadcast sensor information via CAN bus based on SAE J1939 and ISO11783 protocols. This information can be captured and used in lieu of analog sensor measurements in some cases. Hardware and software tools that interface with the machine CAN bus to interpret and record machine information are commercially available. If information from the CAN bus can be used in lieu of analog measurements there may be a significant savings in cost, complexity, and effort required to collect tractor and implement data in the field.

Although there are advantages to using CAN data instead of analog measurements a primary concern is accuracy of the CAN parameters. Many CAN parameters are based on on-machine sensors, for which traceable calibration is not available, or parameters of interest may be based on software tables or calculations and are not actually measured directly. Before pursuing large scale field data collection based on CAN data some effort is required to verify the accuracy of parameters of interest. The objective of research described in this chapter was to evaluate accuracy of net engine torque determined from J1939 CAN messages.

### 2.1 Engine torque measurement

An experiment was performed to determine how closely net engine torque based on J1939 CAN parameters correlated with torque measured with a dynamometer. Hardware used for this experiment included a four cylinder John Deere industrial diesel engine fitted with a TwinDisc SP111HP3 clutch and connected to a Dynamatic 1519DG eddy current dynamometer as shown in Figure 25. The Tier 3 John Deere model 4045HG485 engine
had 4.5 L displacement with nominal power of 115 kW at 2400 rpm. The dynamometer was equipped with an Interface 1110-JW load cell and Sensoronix EP10234 speed sensor and was controlled with a DynoOne EC1001C Eddy Current Dynamometer Controller. The load cell had dual outputs to provide a load signal to the dynamometer controller and a load signal to the data acquisition system.

![Image of engine and dynamometer](image)

**Figure 25** Engine and dynamometer configuration

Dynamometer torque was calibrated in increments of 203.37 Nm with certified weights. As calibration weights were incrementally applied, load cell voltage was recorded as an average over a 20 second duration at constant load. This process was repeated as the weights were incrementally removed. Load cell voltage for corresponding loading and
unloading points were averaged together to account for hysteresis from loading direction.

A calibration table was made from the applied load and corresponding load cell voltage.

The calibration table was applied to the load cell signal in the data acquisition software and weights were applied again to verify the calibration. Results are shown in Table 1 along with percent error for each load point. The largest error was 0.19% and the average error was 0.11%.

**Table 1** Summary of dynamometer torque calibration results

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<th>Applied Load (Nm)</th>
<th>DAQ Measurements</th>
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<tr>
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<td>Loading (Nm)</td>
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</tr>
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Values from Table 1 are shown graphically in Figure 26. A trend line was fit to the average scaled values. The coefficient of determination, or $R^2$ value, of 0.999 was an indication of strong correlation between applied load and measured load.
Figure 26 Cross plot of measured vs. applied dynamometer torque load after calibration.

The data acquisition system, shown in Figure 27 consisted of a National Instruments cDAQ-9174 equipped with the following modules:

- NI 9862 1-Port High-Speed/FD NI-XNET CAN C Series Module
- NI 9219 Universal Analog Input, 24-Bit, 100S/s/ch, 4 ch Module
- NI 9401 5V/TTL, Bidirectional Digital I/O, 8 Ch Module
A custom LabVIEW program was created to view and record data during the experiment. The graphic user interface and block diagram for this program are shown in APPENDIX A. The data acquisition system had three inputs: J1939 CAN, dynamometer torque (voltage signal), and dynamometer speed (digital signal). The LabVIEW software was configured to use the same sample clock for all signals to ensure synchronization of the collected data. Data were collected in waveform format at a rate of 20 Hz and were streamed to a LabVIEW Technical Data Management Streaming (.tdms) log file with no additional signal conditioning.

Figure 27 National Instruments cDAQ-9174 with modules
It should be noted that J1939 parameter groups are broadcast at varying rates and some are updated based on engine crank angle, or when a state change occurs, rather than at a specific time interval. To ensure time synchronization of CAN signals with analog signals, the same analog sample clock was used to start the CAN data task and to resample CAN data at the same rate as the analog signals. A CAN database file (.dbc file) was used with NI-XNET to decode the J1939 messages in real time before writing the parameters to the log file.

2.2 J1939 CAN messages for torque estimation

As described in SAE J1939-71 and SAE J1939DA (Digital Annex), static messages available in PGN 65251 EC1 – Engine Configuration 1 provide information about the general shape of the engine torque curve and engine reference torque. Engine configuration data retrieved from the CAN bus for the John Deere 4.5L engine used in this study are shown in Table 2. Torque messages on the CAN bus were reported as a percent of engine reference torque which is the 100% reference value for all defined engine torque parameters. For the engine used in this study, engine reference torque is 700 Nm.
Table 2 PGN 65251 EC1 - Engine Configuration 1 messages from John Deere 4.5L engine.

<table>
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<th>parameter name</th>
<th>SPN</th>
<th>Units</th>
<th>Reported value</th>
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<tbody>
<tr>
<td>EngSpeedAtIdlePoint1</td>
<td>188</td>
<td>rpm</td>
<td>800</td>
</tr>
<tr>
<td>EngSpeedAtPoint2</td>
<td>528</td>
<td>rpm</td>
<td>2470</td>
</tr>
<tr>
<td>EngSpeedAtPoint3</td>
<td>529</td>
<td>rpm</td>
<td>500</td>
</tr>
<tr>
<td>EngSpeedAtPoint4</td>
<td>530</td>
<td>rpm</td>
<td>1156.625</td>
</tr>
<tr>
<td>EngSpeedAtPoint5</td>
<td>531</td>
<td>rpm</td>
<td>1813.25</td>
</tr>
<tr>
<td>EngSpeedAtHighIdlePoint6</td>
<td>532</td>
<td>rpm</td>
<td>2470</td>
</tr>
<tr>
<td>EngPercentTorqueAtIdlePoint1</td>
<td>539</td>
<td>%</td>
<td>77</td>
</tr>
<tr>
<td>EngPercentTorqueAtPoint2</td>
<td>540</td>
<td>%</td>
<td>77</td>
</tr>
<tr>
<td>EngPercentTorqueAtPoint3</td>
<td>541</td>
<td>%</td>
<td>77</td>
</tr>
<tr>
<td>EngPercentTorqueAtPoint4</td>
<td>542</td>
<td>%</td>
<td>90</td>
</tr>
<tr>
<td>EngPercentTorqueAtPoint5</td>
<td>543</td>
<td>%</td>
<td>89</td>
</tr>
<tr>
<td>EngReferenceTorque</td>
<td>544</td>
<td>Nm</td>
<td>700</td>
</tr>
</tbody>
</table>

Torque values reported as percent of engine reference-torque were converted to engineering units using Equation 1. The stationary engine torque map as described in PGN 65251 EC1 – Engine Configuration 1 for the John Deere 4.5 L engine used in this study is shown in Figure 28.

Equation 1

\[
torque \ (Nm) = \frac{\text{percent torque} \ (\%)}{100} * \text{EngReferenceTorque} \ (Nm)
\]
Engine friction torque values that correspond to the stationary engine torque values given in PGN 65251 EC1 are defined in PGN 64743 EC3 – Engine Configuration 3 but the EC3 friction torque values were not reported for this engine. SAE J1939DA defines a set of messages in PGN 64912 - Advertised Engine Torque Curve that may have been helpful in understanding the torque characteristics of this engine but these messages were also not reported for this engine.

SAE J1939 defines Net Engine Brake Torque (Power) as:

The measured torque (or power output) of a "fully equipped" engine. A fully equipped engine is an engine equipped with accessories necessary to perform its intended service. This includes, but is not restricted to, the basic engine, including fuel, oil, and cooling pumps, plus intake air system, exhaust system, cooling system,
alternator, and starter, emissions, and noise control. Accessories which are not necessary for the operation of the engine, but may be engine mounted, are not considered part of a fully equipped engine. These items include, but are not restricted to, power steering pump systems, vacuum pumps, and compressor systems for air conditioning, brakes, and suspensions.86

Net engine brake torque is equal to the torque developed in the cylinders minus torque required to overcome engine friction and torque required to drive engine accessories. These loads that are necessary for operation of the engine but at the same time do not contribute to useful work are often described as parasitic loads. Engine parasitic loads can include:

- Engine friction
- Air intake restrictions
- Exhaust system restrictions
- Fuel pump
- Oil pump
- Coolant pump
- Alternator
- Fan
- Other accessories such as: air compressor, air conditioning compressor, etc.

SAE J1939 defines several messages that can be used to calculate net engine brake torque. These messages and their abbreviated definitions from SAE J1939DA are given below:

**Actual Engine Percent Torque (SPN 513)**

The calculated output torque of the engine. The data is transmitted in indicated torque as a percent of reference engine torque. The engine percent torque value will not be less than zero and it includes the torque developed in the cylinders required to overcome friction.
Actual Engine Percent Torque – Fractional (SPN 4154)

The parameter is used in combination with SPN 513 Actual Engine Torque. The resulting actual engine torque will be calculated by adding these two parameters.

Nominal Engine Friction - Percent Torque (SPN 514)

The calculated torque that indicates the amount of torque required by the basic engine itself added by the loss torque of accessories. It contains the frictional and thermodynamic loss of the engine itself, pumping torque loss, and the losses of fuel, oil and cooling pumps. The realization can be done by a map dependent on engine speed and engine temperature and an offset value for additional loss torques. See SPN 2978 for an indicator that describes the possible inclusion of engine parasitic losses such as cooling fan, etc. in this parameter value.

Estimated Engine Parasitic Losses – Percent Torque (SPN 2978)

The calculated torque that indicates the estimated amount of torque loss due to engine parasitics, such as cooling fan, air compressor, air conditioning, etc. It is expressed as a percent of Engine Reference Torque.

The engine used for this study reported Actual Engine Percent Torque (Fractional) (SPN 4154) as a constant value of 1.875% which is outside its defined range of 0-0.875%. This parameter has a resolution of 0.125%/bit and an offset of zero. When the reported value of
1.875% is divided by 0.125%/bit, the result is 15\textsubscript{10} or 1111\textsubscript{2}. Per SAE J1939DA, values of 1000\textsubscript{2}-1111\textsubscript{2} indicate this message is not available and so it will not be considered in subsequent torque calculations in this study.

Estimated Engine Parasitic Losses Percent Torque (SPN 2978) was reported as a constant value of 130% which is outside of the defined data range of -125 to 125% for this parameter. The parameter has a resolution of 1%/bit and an offset of -125%. When the -125% offset is applied to the 130% reported value the result is 255\textsubscript{10} or FF\textsubscript{16}. SAE J1939 specifies that undefined bytes should be sent as FF\textsubscript{16} thus indicating this message is not defined for this engine so it will not be used in subsequent torque calculations.\textsuperscript{86, 88}

When the reported value for Estimated Engine Parasitic Losses Percent Torque is equal to FB\textsubscript{16} it indicates that all parasitic losses calculated by the engine are included in the engine’s Nominal Friction Percent Torque (SPN 514).\textsuperscript{87} As described in the previous paragraph, the reported value for Estimated Engine Parasitic Losses Percent Torque is not defined for this engine and does not show that the parasitic load for the fan is included in the Nominal Engine Percent Torque data values.
A schematic of the engine and dynamometer used in this study is shown in Figure 29. As described above, gross torque generated in the cylinders is characterized by Actual Engine Percent Torque (SPN 513). Parasitic loads, with exception of the cooling fan, are included in the definition of Nominal Engine Friction Percent Torque (SPN 514). Engine net brake torque is calculated from CAN parameters by subtracting Nominal Friction Torque from Actual Engine Torque and multiplying by Engine Reference Torque as shown in Equation 2.

**Equation 2**

\[
\text{Net Torque (Nm)} = \frac{(\text{ActualEng}\%Tq - \text{NominalEngFriction}\%Tq)}{100} \times \text{EngRefTorque}
\]
2.3 Fan torque estimation

Information presented on the CAN bus indicates fan load is not accounted for in Nominal Engine Friction Percent Torque or Estimated Engine Parasitic Losses parameters so fan load will be accounted for separately. The engine sales distributor (Industrial Irrigation, Hastings, NE) provided fan power at various speeds as shown in Table 3. This fan load data does not correspond with the engine speeds of interest in this study so this data was used to create a model of fan torque as a function of fan speed.

**Table 3** Fan power at various speeds provided by engine distributor. Torque was calculated from the power and speed values.

<table>
<thead>
<tr>
<th>speed (rpm)</th>
<th>power (kW)</th>
<th>torque (Nm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1476</td>
<td>1.5</td>
<td>9.7</td>
</tr>
<tr>
<td>1674</td>
<td>2.3</td>
<td>13.1</td>
</tr>
<tr>
<td>1836</td>
<td>3.0</td>
<td>15.6</td>
</tr>
<tr>
<td>2016</td>
<td>3.9</td>
<td>18.5</td>
</tr>
<tr>
<td>1640</td>
<td>2.1</td>
<td>12.2</td>
</tr>
<tr>
<td>1860</td>
<td>3.1</td>
<td>15.9</td>
</tr>
<tr>
<td>2040</td>
<td>4.1</td>
<td>19.2</td>
</tr>
<tr>
<td>2240</td>
<td>5.4</td>
<td>23.0</td>
</tr>
<tr>
<td>1804</td>
<td>2.8</td>
<td>14.8</td>
</tr>
<tr>
<td>2046</td>
<td>4.1</td>
<td>19.1</td>
</tr>
<tr>
<td>2244</td>
<td>5.4</td>
<td>23.0</td>
</tr>
<tr>
<td>2464</td>
<td>7.2</td>
<td>27.9</td>
</tr>
<tr>
<td>1968</td>
<td>3.7</td>
<td>18.0</td>
</tr>
<tr>
<td>2232</td>
<td>5.3</td>
<td>22.7</td>
</tr>
<tr>
<td>2448</td>
<td>7.0</td>
<td>27.3</td>
</tr>
</tbody>
</table>

According to fan laws, fan power is proportional to the fan speed cubed.\(^8^9\) Because fan power is the product of torque and speed, fan torque is proportional to fan speed squared. Regression analysis using the available fan data provided a best fit equation to be used for
estimating fan loads for engine speeds of interest. A quadratic function of the form shown in Equation 3 was used to model fan torque as a function of speed.

**Equation 3** \[ y = mx^2 \]

This model provided logical values for the physical system where zero torque occurs at zero speed and as speed increases, fan torque increases at an increasing rate. Equation 4 shows the resulting model where fan torque is in Nm and fan speed is in rpm. The coefficient of determination, or \( R^2 \) value, of 0.999 is an indication that this model for fan torque is a good fit for the original data. Additional regression statistics are shown in Table 4.

**Equation 4** \[ \text{fan torque} = 4.57989E-06 \times \text{fan speed}^2 \]

**Table 4 Fan torque regression statistics for Microsoft Excel LINEST() regression.**

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>constant base coefficient</td>
<td>m = 4.57989E-06</td>
</tr>
<tr>
<td>coefficient of determination</td>
<td>( r^2 ) = 0.999938</td>
</tr>
<tr>
<td>standard error values for the coefficients</td>
<td>( se_n ) = 9.6037E-09</td>
</tr>
<tr>
<td>The F statistic, or the F-observed value</td>
<td>F = 227422</td>
</tr>
<tr>
<td>regression sum of squares</td>
<td>( ss_{reg} ) = 5626</td>
</tr>
<tr>
<td>standard error for the y estimate</td>
<td>( se_y ) = 0.157</td>
</tr>
<tr>
<td>degrees of freedom</td>
<td>df = 14</td>
</tr>
<tr>
<td>residual sum of squares</td>
<td>( ss_{resid} ) = 0.346</td>
</tr>
</tbody>
</table>

Figure 30 shows fan torque calculated from information provided by the engine sales distributor along with values predicted with Equation 4.
Figure 30 Plot of actual fan torque and fan torque estimated with regression model.

The drive ratio between the engine crank shaft and the fan was calculated from measured fan and engine speeds as shown in Equation 5. Fan torque was multiplied by this ratio to reflect fan load back to the engine.

**Equation 5** \( \frac{960 \text{ rpm fan speed}}{800 \text{ rpm engine speed}} = 1.20 \)

2.4 Data collection and analysis

For this study it was important to synchronize J1939 CAN data with analog data to enable valid comparison between reported and measured parameters. An indication of the level of synchronization is shown by comparing J1939 reported engine speed and measured
dynamometer speed. If the signals were not well synchronized large differences would be expected during transitions in engine speed. Figure 31 shows an over-plot of J1939 EngSpeed (SPN 190) and dynamometer speed. Included in Figure 31 is a plot of absolute difference, or error, in these signals along with statistics related to the error. The plot shows consistent error of approximately ±5 rpm across all speeds and mean error of 0.05 rpm with no perceivable change in magnitude during speed transitions.

<table>
<thead>
<tr>
<th>EngSpdError statistics</th>
<th>rpm</th>
</tr>
</thead>
<tbody>
<tr>
<td>min</td>
<td>-5.6</td>
</tr>
<tr>
<td>max</td>
<td>4.9</td>
</tr>
<tr>
<td>mean</td>
<td>0.05</td>
</tr>
<tr>
<td>standard deviation</td>
<td>1.2</td>
</tr>
</tbody>
</table>

Figure 31 Graph showing synchronization of J1939 EngSpeed (SPN 190) and analog dynamometer speed

A cross-plot and linear regression of dynamometer speed and J1939 EngSpeed is shown in Figure 32. A slope close to one with an intercept of zero, along with an $R^2$ value of 0.999, are strong evidence of correlation of these speed signals.
Early in this project a significant amount of noise was seen in the analog torque signal, especially at speeds above 1700 rpm. Inspection of mechanical components revealed some imbalance in the driveshaft between the clutch and dynamometer. A bearing and yoke assembly was replaced and the driveshaft was dynamically balanced. Repair and balancing of the driveshaft helped reduce vibration but did not eliminate noise in the torque signal. The noise is not believed to be electrical in nature because no changes in signal quality were seen after trying different shielded instrument cables, load cells, amplifiers, and data acquisition modules. For the duration of this study no solution was found to reduce the noise in the torque signal. It is believed to be caused by dynamic (mechanical) interaction between the engine, clutch, and dynamometer and interaction between engine and dynamometer control systems.

**Figure 32** Cross-plot and linear regression of dynamometer speed and J1939 EngSpeed
Figure 33 shows data from an engine speed sweep where speed was reduced from 2470 rpm, high idle engine speed, down to 1200 rpm. The noise in the torque signal was very apparent at engine speeds greater than 1700 rpm. From 1700 rpm down to 1300 rpm the torque signal was stable but at 1200 rpm the signal became unstable. Due to these findings only data from 1300 rpm to 1700 rpm are used in this study. This is unfortunate because data of interest for field data collection will include a broader range of engine speeds.

Data were collected at steady state points while decreasing and then increasing engine speed in 100 rpm increments with the engine at full throttle. The engine was allowed to settle and remain at steady state for two to three minutes at each operating point. During this process, the maximum torque capability of the engine was found at each respective
engine speed. At each of the same 100 rpm engine speed increments, partial load (i.e. torque) measurements were also made. For these measurements the throttle was reduced until engine speed was at the desired value and then the dynamometer controller was used to apply varying amounts of torque. Torque was increased in 136 Nm increments and then decreased at the same intervals while being held at steady state for two to three minutes at each load increment. All data were recorded with no additional filtering or signal conditioning.

Raw data were analyzed using DIAdem software. A twenty point, symmetric moving average was used to smooth the dynamometer torque signal shown in Figure 33. Torque was calculated from J1939 CAN parameters as described in Equation 2. An average of each signal was taken for each load point described above. Before comparison was made between dynamometer torque and J1939 calculated net torque the load from the cooling fan had to be accounted for. Average engine speed at each respective load point, along with the fan drive ratio, was used with Equation 4 to estimate fan torque. This fan torque estimate was then reflected back to the engine and subtracted from J1939 calculated torque to get the final estimate of net engine torque. A summary of calculated net engine torque and dynamometer torque is provided in the table in APPENDIX B and is also shown graphically in Figure 34. The difference between J1939 indicated torque, reported in EC1 – Engine Configuration, and calculated net engine torque may be due to lack of Nominal Engine Friction parameters defined in PGN 64743 EC3 – Engine Configuration 3. Nominal Engine Friction parameters reported in EC3 Engine Configuration would have
been subtracted from the corresponding J1939 Indicated Torque parameters bringing the Indicated Torque curve closer to calculated net engine torque and dynamometer torque.

Figure 34 Torque vs. engine speed plot showing J1939 EC1 - Engine Configuration Indicated Torque, calculated net engine torque, and dynamometer torque.

The engine used in this study was not installed in a chassis and so there were no powertrain, implement, or other external devices that can interact with the engine controls. For data collection performed at the machine level there may be other torque modes initiated by cruise control, torque limiting, braking system, transmission control, and others modes depending on engine and machine configuration. Engine Torque Modes (SPN 899) can be monitored to better understand the torque control status of the engine. A no-load speed sweep for the engine used in this study showed changes in Engine Torque Mode (SPN 899) as the controlling feature changed with engine status. Three engine
torque modes can be seen with this simple speed sweep as shown in Table 5 and Figure 35.

**Table 5** Engine Torque Modes for no-load speed sweep on John Deere 4.5L engine

<table>
<thead>
<tr>
<th>Engine Torque Mode (SPN 899)</th>
<th>Description</th>
<th>Engine speed (rpm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>Low idle governor/no request (default mode)</td>
<td>800 (low idle)</td>
</tr>
<tr>
<td>1</td>
<td>Accelerator pedal/operator selection</td>
<td>&gt; 800 and &lt; 2470</td>
</tr>
<tr>
<td>9</td>
<td>High speed governor</td>
<td>2470 (high idle)</td>
</tr>
</tbody>
</table>

**Figure 35** Engine Torque Modes reported during no-load speed sweep on John Deere 4.5L engine
2.5 Results

Descriptive statistics for overall error between calculated net engine torque and
dynamometer torque are shown in Table 6. The range of error was 44.7 Nm for all data
and, on average, calculated net engine torque was 15.3 Nm higher than dynamometer
torque with a standard deviation of 12.0 Nm.

Table 6 Descriptive statistics for torque error.

<table>
<thead>
<tr>
<th></th>
<th>all data</th>
<th>decreasing load</th>
<th>increasing load</th>
</tr>
</thead>
<tbody>
<tr>
<td>min error</td>
<td>-9.9</td>
<td>-9.9</td>
<td>-5.0</td>
</tr>
<tr>
<td>max error</td>
<td>34.8</td>
<td>31.2</td>
<td>34.8</td>
</tr>
<tr>
<td>mean error</td>
<td>15.3</td>
<td>14.7</td>
<td>16.0</td>
</tr>
<tr>
<td>error standard deviation</td>
<td>12.0</td>
<td>12.5</td>
<td>11.7</td>
</tr>
</tbody>
</table>

During data collection it was noted that the torque level did not seem to return to the same
value when approaching a load point from a different loading direction. A statistical test of
the mean torque error was completed to compare increasing and decreasing load
directions. This analysis showed that the assumption of equal variance was valid and no
statistical difference in the mean error based on loading direction was found. Table 7
shows test statistics for torque error comparing decreasing and increasing load direction
for all data. Test statistics for torque error comparing loading direction evaluated by speed
are shown in Table 8, and test statistics for torque error comparing loading direction
evaluated by torque are shown in Table 9. In each case the p-value was greater than the
alpha level of 0.05. Therefore, one would fail to reject the null hypothesis that mean
increasing torque error is equal to mean decreasing torque error and conclude there is no
difference in the data with regard to loading direction. Accordingly, all data will be used in subsequent analysis and loading direction will not be considered.

**Table 7** Test statistics for torque error comparing decreasing and increasing load direction for all data.

<table>
<thead>
<tr>
<th>direction</th>
<th>Method</th>
<th>Mean</th>
<th>95% CL Mean</th>
<th>Std Dev</th>
<th>95% CL Std Dev</th>
</tr>
</thead>
<tbody>
<tr>
<td>decrease</td>
<td></td>
<td>14.6701</td>
<td>8.8148</td>
<td>20.5253</td>
<td>12.5109</td>
</tr>
<tr>
<td>increase</td>
<td></td>
<td>15.9568</td>
<td>10.4918</td>
<td>21.4218</td>
<td>11.6771</td>
</tr>
<tr>
<td>Diff (1-2)</td>
<td>Pooled</td>
<td>-1.2868</td>
<td>-9.0335</td>
<td>6.4600</td>
<td>12.1011</td>
</tr>
<tr>
<td>Diff (1-2)</td>
<td>Satterthwaite</td>
<td>-1.2868</td>
<td>-9.0347</td>
<td>6.4612</td>
<td></td>
</tr>
</tbody>
</table>

| Method     | Variances  | DF   | t Value | Pr > |t| |
|------------|------------|------|---------|------|---|
| Pooled     | Equal      | 38   | -0.34   | 0.7385 |
| Satterthwaite | Unequal  | 37.821 | -0.34 | 0.7385 |

**Equality of Variances**

<table>
<thead>
<tr>
<th>Method</th>
<th>Num DF</th>
<th>Den DF</th>
<th>F Value</th>
<th>Pr &gt; F</th>
</tr>
</thead>
<tbody>
<tr>
<td>Folded F</td>
<td>19</td>
<td>19</td>
<td>1.15</td>
<td>0.7668</td>
</tr>
</tbody>
</table>

**Table 8** Test statistics for differences between directions evaluated by speed. Simple Effect Comparisons of direction by speed level Least Squares Means. Adjustment for Multiple Comparisons: Holm-Tukey.

<table>
<thead>
<tr>
<th>Simple Effect Level</th>
<th>direction</th>
<th>direction</th>
<th>Estimate</th>
<th>Standard Error</th>
<th>DF</th>
<th>t Value</th>
<th>Pr &gt;</th>
<th>t</th>
<th>Adj P</th>
</tr>
</thead>
<tbody>
<tr>
<td>speed_1300</td>
<td>decrease</td>
<td>increase</td>
<td>-3.1841</td>
<td>8.4306</td>
<td>30</td>
<td>-0.38</td>
<td>0.7083</td>
<td>0.7083</td>
<td></td>
</tr>
<tr>
<td>speed_1400</td>
<td>decrease</td>
<td>increase</td>
<td>-4.0607</td>
<td>8.4306</td>
<td>30</td>
<td>-0.48</td>
<td>0.6335</td>
<td>0.6335</td>
<td></td>
</tr>
<tr>
<td>speed_1500</td>
<td>decrease</td>
<td>increase</td>
<td>-0.6465</td>
<td>8.4306</td>
<td>30</td>
<td>-0.08</td>
<td>0.9394</td>
<td>0.9394</td>
<td></td>
</tr>
<tr>
<td>speed_1600</td>
<td>decrease</td>
<td>increase</td>
<td>0.6466</td>
<td>8.4306</td>
<td>30</td>
<td>0.08</td>
<td>0.9394</td>
<td>0.9394</td>
<td></td>
</tr>
<tr>
<td>speed_1700</td>
<td>decrease</td>
<td>increase</td>
<td>0.8109</td>
<td>8.4306</td>
<td>30</td>
<td>0.10</td>
<td>0.9240</td>
<td>0.9240</td>
<td></td>
</tr>
</tbody>
</table>
Table 9 Test statistics for differences between directions evaluated by torque. Simple Effect Comparisons of direction by torque loading level Least Squares Means. Adjustment for Multiple Comparisons: Holm-Tukey.

| Simple Effect Level | direction | direction | Estimate | Standard Error | DF | t Value | Pr > |t| | Adj P |
|---------------------|-----------|-----------|----------|----------------|----|---------|-------|---|-------|
| Loading 136 Nm      | decrease  | increase  | 3.4210   | 5.6513         | 32 | 0.61    | 0.5492 | 0.5492 |
| Loading 271 Nm      | decrease  | increase  | -3.4631  | 5.6513         | 32 | -0.61   | 0.5443 | 0.5443 |
| Loading 407 Nm      | decrease  | increase  | -2.3765  | 5.6513         | 32 | -0.42   | 0.6769 | 0.6769 |
| Loading Full Throttle | decrease | increase | -2.7285  | 5.6513         | 32 | -0.48   | 0.6325 | 0.6325 |

Analysis of error between calculated net engine torque and dynamometer torque showed that the torque error was different from zero and calculated torque was statistically higher than measured torque when evaluated overall and by speed and load. In each case the p-values were less than the alpha level of 0.05 and therefore one would reject the null hypothesis that the error is equal to zero and conclude that the calculated torque is statistically different than the measured torque. Table 10 shows test statistics for overall error between calculated and measured torque. Test statistics for error between calculated and measured torque evaluated by speed is shown in Table 11, and test statistics for error between calculated and measured torque evaluated by torque is shown in Table 12.
Table 10 Test statistics for error between calculated and measured torque for all loads.

<table>
<thead>
<tr>
<th>N</th>
<th>Mean</th>
<th>Std Dev</th>
<th>Std Err</th>
<th>Minimum</th>
<th>Maximum</th>
</tr>
</thead>
<tbody>
<tr>
<td>40</td>
<td>15.3134</td>
<td>11.9628</td>
<td>1.8915</td>
<td>-9.8953</td>
<td>34.7707</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Mean 95% CL Mean</th>
<th>Std Dev 95% CL Std Dev</th>
</tr>
</thead>
</table>

| DF  t Value Pr > |t| |
|-----------------|---------|
| 39 8.10 <.0001 |

Table 11 Least Squares Means test statistics for error between calculated and measured torque evaluated by speed.

| Speed | Estimate | Standard Error | DF  t Value | Pr > |t|
|-------|----------|----------------|-------------|-------|
| 1300  | 17.4466  | 1.1182         | 20          | 15.60 | <.0001 |
| 1400  | 5.1507   | 1.1182         | 20          | 4.61  | 0.0002 |
| 1500  | 22.2433  | 1.1182         | 20          | 19.89 | <.0001 |
| 1600  | 16.6044  | 1.1182         | 20          | 14.85 | <.0001 |
| 1700  | 15.1221  | 1.1182         | 20          | 13.52 | <.0001 |

Table 12 Least Squares Means test statistics for error between calculated and measured torque evaluated by torque.

| Loading | Estimate | Standard Error | DF  t Value | Pr > |t|
|---------|----------|----------------|-------------|-------|
| 136 Nm  | 15.6295  | 1.0002         | 20          | 15.63 | <.0001 |
| 271 Nm  | 28.3741  | 1.0002         | 20          | 28.37 | <.0001 |
| 407 Nm  | 4.5462   | 1.0002         | 20          | 4.55  | 0.0002 |
| Full Throttle | 12.7039 | 1.0002 | 20 | 12.70 | <.0001 |
All speed and load combinations were evaluated to look for trends corresponding to changes in speed and load. Table 16 in APPENDIX C shows test statistics for differences between loadings when speed was held constant. Table 17 in APPENDIX C shows test statistics for differences between speeds when loading was held constant. P-values for some speed and load combinations were less than the 0.05 alpha level indicating a statistically significant difference but for other speed and load combinations the p-value was greater than the 0.05 alpha level indicating no statistically significant difference between the respective loads at a given speed or speeds at a given load. No consistent trends in torque error were found based on speed or load. Torque error plotted versus engine speed (Figure 36) and versus dynamometer torque (Figure 37) show no apparent patterns or trends.

**Figure 36** Torque error versus engine speed.
Figure 37 Torque error versus dynamometer torque.

A linear regression of measured torque on calculated torque showed a strong relationship where an $R^2$ value of 0.99 confirmed that measured torque explained 99% of variation in calculated torque. The regression equation shown in Figure 38 was quite close to a slope of one but with an offset of 23 Nm, indicating calculated torque is consistently higher than measured torque.

Figure 38 Cross-plot showing correlation of calculated net torque and measured torque.
2.6 Sources of error

In this study there were several potential sources of error. One such source had to do with the resolution of J1939 torque parameters that are reported as a percent of engine reference torque. Engine reference torque for this engine was 700 Nm which results in a resolution of 7 Nm for the two parameters used in the net engine torque calculation, Actual Engine Percent Torque and Nominal Engine Friction Percent Torque. If the high resolution Actual Engine Percent Torque - Fractional parameter (SPN 4154) had been available, torque resolution would have been 0.125% of Engine Reference Torque, or 0.875 Nm for this engine. This improvement in resolution would have allowed for a much better evaluation of torque error.

Another potential source of error comes from the fact that the J1939 reported torque parameters are not directly measured but instead originate from calculations or tables developed by the engine manufacturer. Conditions under which these parameters were developed are not known to the user of the J1939 data and may be different than conditions present when the user is collecting data. Differences could include oil viscosity and intake and exhaust restrictions. Also, the engine used for development of these parameters may have had different friction losses due to break-in and/or engine tolerances or different cooling system.

It is believed that the mathematical model for the fan load was accurate based on the information available but other characteristics of the cooling package and air flow in the
local environment could affect these results. A detailed study of the cooling package and fan could be done but was not practical for this study.

Although the dynamometer was calibrated with certified weights, some hysteresis was found in the torque signal that may contribute to the overall torque error in this study. Average dynamometer torque signal error was 0.11% and maximum error was 0.19%.

In this study all torque measurements were attempted under steady state load conditions to exclude any dynamic effects. Torque during load transitions was excluded from the data analysis but mechanical vibration or other sources of signal noise may have contributed to overall error.

2.7 Conclusions

SAE J1939 defines parameters that can be used to estimate net engine torque. Not all parameters were reported for the engine that was used in this study so additional work was done to model the parasitic load from the cooling fan to ensure all loads were accounted for. On average, J1939 calculated net engine torque was 15.3 Nm higher than dynamometer torque with an overall range in torque error of 44.7 Nm. Evaluation of torque error hysteresis based on loading direction showed no statistical difference in direction at an alpha level of 0.05 when evaluated by speed, load, and overall, indicating loading direction can be reasonably ignored in analysis. A linear regression of measured torque on calculated torque showed a strong relationship indicated by a high $R^2$ value. The error between calculated torque and measured torque was statistically greater than zero at
an alpha level of 0.05 when evaluated by speed, load, and overall. There was significant interaction between speed and load with regard to torque error, however, no consistent trends were found.

Potential sources of error included low resolution of J1939 torque signals, mathematical modeling of the fan, possible error in analog signals, signal noise, and the fact that this study used a different engine and different environmental conditions than those used to develop of the J1939 torque parameters.

Comparison of J1939 EngSpeed (SPN 190) and dynamometer speed showed that J1939 EngSpeed is a good predictor of actual engine speed. J1939 EngSpeed was used to verify synchronization of J1939 parameters and analog signals.

2.8 Suggestions for future work

The source of noise in the dynamometer torque signal should be identified before conducting further tests. Steps as simple as installing a torsional damper in the driveline and refining the dynamometer PI controller may improve measurement quality. Reducing torque signal noise may allow for a more comprehensive evaluation across the entire engine operating range.

Conclusions of this study are based on results obtained from a single engine. Additional testing is recommended with different engine models from the same, and different manufacturers, to provide a more thorough evaluation. Testing an engine that reports
Actual Engine Percent Torque – Fractional (SPN 4154) and Estimated Engine Parasitic Losses – Percent Torque (SPN 2978) would help evaluate the significance of the J1939 torque resolution and ensure engine parasitic loads are accounted for by the engine manufacturer. In addition to evaluating more engines, the power of the statistical analysis could be improved by increasing the number of observations at each point of interest.

With further testing, if it can be proven that the error between calculated net engine torque and dynamometer torque is consistent then it may be practical to simply apply an offset to the calculated torque to get closer estimation of actual torque.
CHAPTER 3: TOOLS FOR CAN DATA COLLECTION

3 Introduction

Many hardware and software tools are commercially available for monitoring and logging J1939 messages from a tractor’s CAN bus and implement bus (ISOBUS 11783). Some tools that have been used during this study are listed in Table 13.

Table 13 Summary of CAN tools used during this study.

<table>
<thead>
<tr>
<th>hardware</th>
<th>software</th>
<th>result</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vector CANcaseXL log</td>
<td>CANalyzer</td>
<td>viewed decoded parameters and logged raw messages to a Vector ASCII file (.asc)</td>
</tr>
<tr>
<td>National Instruments 9862 NI-XNET High-Speed CAN Module</td>
<td>LabVIEW</td>
<td>decoded messages and resampled parameters in real time before logging data to a National Instruments Technical Data Management Streaming file (.tdms)</td>
</tr>
<tr>
<td>Kvaser Leaf Light v2</td>
<td>CanKing</td>
<td>viewed raw CAN messages and logged them to a text file (.txt)</td>
</tr>
<tr>
<td>Kvaser Leaf Light v2</td>
<td>Matlab</td>
<td>viewed decoded CAN messages and logged decoded parameters to a Matlab data file (.mat)</td>
</tr>
</tbody>
</table>

All of these tools were successful at interfacing with the CAN bus and each has advantages for specific use cases. Considerations included:

- portability (i.e. powered via USB vs. external power supply, physical size, etc.)
- number of CAN channels supported
- ease of use (i.e. plug & play vs. having to create your own software to communicate with the device)
- cost
- software licensing
- ability to:
  - withstand harsh environmental conditions
  - use database file (.dbc) to configure specific parameters of interest and decode messages in real time
  - customize software for specific analysis
  - simultaneously read signals from analog sensors and merge this with CAN data
  - create a log file format compatible with other post processing software
3.1 Application development for J1939 field data collection

Matlab software and Kvaser Leaf Light v2 hardware (Figure 39) were used to investigate the feasibility of field data collection with real-time data analysis. This software and hardware were chosen because of availability, cost, ability to use database files to decode data in real time, and ability to create custom software and user interface. A project was undertaken to create a graphic user interface and executable file to collect CAN data, convert it to engineering units, and log it to a file while completing some basic analysis of select J1939 CAN parameters. CAN bus connectivity with Matlab was facilitated with use of Matlab’s Vehicle Network Toolbox (VNT).\(^9^0\) VNT includes several functions for configuring CAN bus communication using Vector and Kvaser hardware and using database files (.dbc) for decoding CAN messages.\(^9^1\) An advantage of using Matlab for this project was the ability to use native Matlab functions for custom analysis and visualization.

Figure 39 Kvaser Leaf Light v2 USB-to-CAN interface\(^9^2\)
Devices and software that decode raw J1939 messages do so with industry-standard database files (.dbc). Most software used to monitor CAN messages include a database editor that allows messages and parameters to be added or edited in database files. For this study a large database was available but to reduce complexity of the project the database file was configured to include only a few messages of interest which are listed in Table 14. Of the seventy three parameters included in the database not all were supported by the John Deere 6135 tractor which was used for this project.

Table 14 Messages included in database file for J1939 data collection application.

<table>
<thead>
<tr>
<th>message name</th>
<th>PGN</th>
<th>acronym</th>
<th>number of parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine Configuration 1</td>
<td>65251</td>
<td>EC1</td>
<td>27</td>
</tr>
<tr>
<td>Electronic Engine Controller 1</td>
<td>61444</td>
<td>EEC1</td>
<td>8</td>
</tr>
<tr>
<td>Electronic Engine Controller 3</td>
<td>65247</td>
<td>EEC3</td>
<td>9</td>
</tr>
<tr>
<td>Electronic Engine Controller 4</td>
<td>65214</td>
<td>EEC4</td>
<td>5</td>
</tr>
<tr>
<td>Engine Torque History</td>
<td>65168</td>
<td>ETH</td>
<td>19</td>
</tr>
<tr>
<td>Fuel Economy (Liquid)</td>
<td>65266</td>
<td>LFE1</td>
<td>5</td>
</tr>
</tbody>
</table>

Matlab Graphic User Interface Development Environment (GUIDE) was used to create the graphic user interface (GUI) shown in Figure 40. This GUI allowed a user to select a J1939 database file and specify bus speed before starting the CAN channel. When the CAN channel was started, streaming data was decoded and several parameters were shown on a time history plot. Real time analysis was done to generate a cumulative bivariate histogram with EngSpeed and ActualEngPercentTorque parameters. The histogram was shown visually on a 2D contour plot and a 3D bar graph. When logging was complete the user was able to stop the CAN channel and save the decoded messages. Collected data was processed to convert the parameter group structure to a Matlab
structure-array where each message included just a time stamp vector and its group of decoded parameters. The program sorted messages to exclude those with empty parameters before saving the decoded message structure as a Matlab formatted file (.mat). Matlab deploytool Application Compiler was used to create a standalone application that was installed from an executable file without a Matlab license.

![Matlab graphic user interface for J1939 data collection application. A time history plot with four J1939 signals is shown at the top. The contour plot and 3D bar graph at the bottom are visualizations of a bivariate histogram of EngSpeed and ActualEngPercentTorque.](image)

**Figure 40** Matlab graphic user interface for J1939 data collection application. A time history plot with four J1939 signals is shown at the top. The contour plot and 3D bar graph at the bottom are visualizations of a bivariate histogram of EngSpeed and ActualEngPercentTorque.
It is worth noting that net engine torque was not calculated for the histogram analysis described above. The intent was to generate a speed-torque histogram which required only relative values and so using the reported percent torque provided a satisfactory result for this proof-of-concept. Additional project-specific analysis can be added as needed to this GUI application. It is also important to note that because messages are broadcast at different rates, resampling of parameters from different messages may be required for specific analysis.

J1939 CAN data is reported in hexadecimal format and is not directly human readable. Some data analysis software includes functions for importing raw J1939 data and converting it to decimal format for post processing. Matlab and DIAdem are two such software packages that were explored during this project. Matlab VNT includes a native function, `canMessageImport()`, that is intended to import and decode CAN messages but it was discovered that this function did not work for J1939 CAN messages. Matlab Technical Support was helpful in providing a workaround to enable the `canMessageImport()` function to work with J1939 CAN messages and indicated this functionality would be included in future releases of VNT. Once converted to decimal format Matlab was very effective for J1939 data analysis and visualization. See APPENDIX D for additional details.

DIAdem software is able to import and decode J1939 data generated by a variety of CAN hardware and software tools including Vector ASCII files (.asc) that were used in this study. DIAdem offered an option to import Kvaser files with .log format but not the .txt
format available from CanKing. National Instruments Technical Support for DIAdem provided a method to import Kvaser .txt files into DIAdem that they intend to include in future releases.

### 3.2 Conclusions and suggestions for future work

This project concluded with successful creation of a working proof-of-concept for a custom J1939 CAN data collection and analysis tool. However, this proof-of-concept program is not sufficient for widespread deployment or long-duration data logging. Optimization of the software should be investigated to speed up loop time with perhaps use of sub functions and/or different architecture such as producer-consumer parallel processing loops. Other opportunities may include improving data management to avoid generating a very large data array before saving it to a log file during prolonged operation of the program. Other features and functionality could include:

- Add functionality to the GUI to allow the user to determine which messages in the selected database are available on the CAN bus. Messages with empty parameters could be detected and excluded from the process by filtering PGN identifiers. The expected benefit would be faster loop time and reduced log file sizes.
- Allow the user to select hardware type and CAN channel from the GUI.
- Include additional error handling; for example, handle errors if the application is started before connection to CAN bus.
- Allow the user to dynamically select parameters for visualization in the time history plot(s).
- Consider using the Matlab animatedline() function instead of plot() function for visualization.

Additional Matlab Vehicle Network Toolbox functions and examples from this study are given in APPENDIX D.
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APPENDIX A - LabVIEW FRONT PANEL AND BLOCK DIAGRAM
The Sample Clock is used as the source clock to synchronize all of the channels and the XNET Session. Therefore, the slave tasks (CI tasks and XNET Session) are started first because they depend on the sample clock of the master task (AI). The master task is started after the slave tasks because this actually starts the AI sample clock. Since the CI uses the AI sample clock, the CI tasks will also start at the same time.
APPENDIX B - ENGINE TORQUE DATA

Table 15 John Deere 4.5L engine torque data.

<table>
<thead>
<tr>
<th>engine speed (rpm)</th>
<th>calculated torque (Nm)</th>
<th>fan torque (Nm)</th>
<th>calculated net engine torque (Nm)</th>
<th>dynamometer torque (Nm)</th>
<th>torque error (%</th>
<th>loading direction</th>
</tr>
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<td>J1939</td>
<td></td>
<td></td>
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</tr>
</tbody>
</table>
APPENDIX C - TEST STATISTICS FOR TORQUE ERROR BASED ON SPEED AND LOAD INTERACTION.

Table 16 Test statistics for differences between loadings when speed is held constant.

| Simple Effect Level | loading_perc | loading_perc | Estimate | Standard Error | DF | t Value | Pr>|t| | Adj P |
|---------------------|--------------|--------------|----------|----------------|----|---------|------|-------|
| speed_level 1300    | 136          | 271          | -25.1780 | 3.1628         | 20 | -7.96  | <.0001 | <.0001|
| speed_level 1300    | 136          | 407          | -9.3442  | 3.1628         | 20 | -2.95  | 0.0078 | 0.0362|
| speed_level 1300    | 136          | Full Throttle| -11.1872 | 3.1628         | 20 | -3.54  | 0.0021 | 0.0103|
| speed_level 1300    | 271          | 407          | 15.8338  | 3.1628         | 20 | 5.01   | <.0001 | 0.0004|
| speed_level 1300    | 271          | Full Throttle| 13.9908  | 3.1628         | 20 | 4.42   | 0.0003 | 0.0014|
| speed_level 1300    | 407          | Full Throttle| -1.8430  | 3.1628         | 20 | -0.58  | 0.5666 | 0.9361|
| speed_level 1400    | 136          | 271          | -26.4903 | 3.1628         | 20 | -8.38  | <.0001 | <.0001|
| speed_level 1400    | 136          | 407          | 5.1099   | 3.1628         | 20 | 1.62   | 0.1218 | 0.3928|
| speed_level 1400    | 136          | 100          | -8.6642  | 3.1628         | 20 | -2.74  | 0.0126 | 0.0564|
| speed_level 1400    | 271          | 407          | 31.6002  | 3.1628         | 20 | 9.99   | <.0001 | <.0001|
| speed_level 1400    | 271          | Full Throttle| 17.8261  | 3.1628         | 20 | 5.64   | <.0001 | <.0001|
| speed_level 1400    | 407          | Full Throttle| -13.7741 | 3.1628         | 20 | -4.35  | 0.0003 | 0.0016|
| speed_level 1500    | 136          | 271          | -4.4697  | 3.1628         | 20 | -1.41  | 0.1730 | 0.5060|
| speed_level 1500    | 136          | 407          | 21.2608  | 3.1628         | 20 | 6.72   | <.0001 | <.0001|
| Simple Effect Level | loading_perc | loading_perc | Estimate | Standard Error | DF | t Value | Pr>|t| | Adj P |
|---------------------|--------------|--------------|----------|----------------|----|---------|------|------|
| speed_level 1500 136 | 100          | 6.7872       | 3.1628   | 20             | 2.15 | 0.0443  | 0.1730 |
| speed_level 1500 271 | Full Throttle | 11.2569      | 3.1628   | 20             | 3.56 | 0.0020  | 0.0098 |
| speed_level 1500 407 | Full Throttle | -14.4736     | 3.1628   | 20             | -4.58 | 0.0002  | 0.0010 |
| speed_level 1600 136 | 271          | -5.6029      | 3.1628   | 20             | -1.77 | 0.0917  | 0.3155 |
| speed_level 1600 136 | 407          | 17.0140      | 3.1628   | 20             | 5.38 | <.0001  | 0.0002 |
| speed_level 1600 136 | Full Throttle | 18.4678      | 3.1628   | 20             | 5.84 | <.0001  | <.0001 |
| speed_level 1600 271 | 407          | 22.6169      | 3.1628   | 20             | 7.15 | <.0001  | <.0001 |
| speed_level 1600 271 | Full Throttle | 24.0707      | 3.1628   | 20             | 7.61 | <.0001  | <.0001 |
| speed_level 1600 407 | Full Throttle | 1.4538       | 3.1628   | 20             | 0.46 | 0.6507  | 0.9669 |
| speed_level 1700 136 | 271          | -1.9820      | 3.1628   | 20             | -0.63 | 0.5380  | 0.9223 |
| speed_level 1700 136 | 407          | 21.3761      | 3.1628   | 20             | 6.76 | <.0001  | <.0001 |
| speed_level 1700 136 | Full Throttle | 9.2243       | 3.1628   | 20             | 2.92 | 0.0085  | 0.0392 |
| speed_level 1700 271 | 407          | 23.3581      | 3.1628   | 20             | 7.39 | <.0001  | <.0001 |
| speed_level 1700 271 | Full Throttle | 11.2063      | 3.1628   | 20             | 3.54 | 0.0020  | 0.0101 |
| speed_level 1700 407 | Full Throttle | -12.1518     | 3.1628   | 20             | -3.84 | 0.0010  | 0.0052 |
Table 17 Test statistics for differences between speeds when load is held constant.

| Simple Effect Level (Nm) | speed_level | speed_level | Estimate | Standard Error | DF | t Value | Pr>|t| | Adj P |
|--------------------------|-------------|-------------|----------|----------------|----|---------|-----|-------|
| loading_136 1300         | 1400        | 8.3797      | 3.1628   | 20             | 2.65| 0.0154  | 0.0986 |
| loading_136 1300         | 1500        | -22.1186    | 3.1628   | 20             | -6.99| <.0001  | <.0001 |
| loading_136 1300         | 1600        | -18.0549    | 3.1628   | 20             | -5.71| <.0001  | 0.0001 |
| loading_136 1300         | 1700        | -16.2574    | 3.1628   | 20             | -5.14| <.0001  | 0.0004 |
| loading_136 1400         | 1500        | -30.4983    | 3.1628   | 20             | -9.64| <.0001  | <.0001 |
| loading_136 1400         | 1600        | -26.4346    | 3.1628   | 20             | -8.36| <.0001  | <.0001 |
| loading_136 1400         | 1700        | -24.6372    | 3.1628   | 20             | -7.79| <.0001  | <.0001 |
| loading_136 1500         | 1600        | 4.0637      | 3.1628   | 20             | 1.28 | 0.2135  | 0.7031 |
| loading_136 1500         | 1700        | 5.8611      | 3.1628   | 20             | 1.85 | 0.0787  | 0.3729 |
| loading_136 1600         | 1700        | 1.7974      | 3.1628   | 20             | 0.57 | 0.5762  | 0.9782 |
| loading_271 1300         | 1400        | 7.0674      | 3.1628   | 20             | 2.23 | 0.0370  | 0.2077 |
| loading_271 1300         | 1500        | -1.4103     | 3.1628   | 20             | -0.45 | 0.6605  | 0.9912 |
| loading_271 1300         | 1600        | 1.5202      | 3.1628   | 20             | 0.48 | 0.6360  | 0.9883 |
| loading_271 1300         | 1700        | 6.9385      | 3.1628   | 20             | 2.19 | 0.0402  | 0.2223 |
| loading_271 1400         | 1500        | -8.4777     | 3.1628   | 20             | -2.68 | 0.0144  | 0.0929 |
| loading_271 1400         | 1600        | -5.5472     | 3.1628   | 20             | -1.75 | 0.0948  | 0.4259 |
| loading_271 1400         | 1700        | -0.1289     | 3.1628   | 20             | -0.04 | 0.9679  | 1.0000 |
| loading_271 1500         | 1600        | 2.9305      | 3.1628   | 20             | 0.93 | 0.3652  | 0.8833 |
| loading_271 1500         | 1700        | 8.3488      | 3.1628   | 20             | 2.64 | 0.0157  | 0.1005 |
| loading_271 1600         | 1700        | 5.4183      | 3.1628   | 20             | 1.71 | 0.1022  | 0.4487 |
| loading_407 1300         | 1400        | 22.8338     | 3.1628   | 20             | 7.22 | <.0001  | <.0001 |
| loading_407 1300         | 1500        | 8.4865      | 3.1628   | 20             | 2.68 | 0.0143  | 0.0924 |
| loading_407 1300         | 1600        | 8.3034      | 3.1628   | 20             | 2.63 | 0.0162  | 0.1032 |
| loading_407 1300         | 1700        | 14.4628     | 3.1628   | 20             | 4.57 | 0.0002  | 0.0015 |
| loading_407 1400         | 1500        | -14.3473    | 3.1628   | 20             | -4.54 | 0.0002  | 0.0017 |
| loading_407 1400         | 1600        | -14.5304    | 3.1628   | 20             | -4.59 | 0.0002  | 0.0015 |
| loading_407 1400         | 1700        | -8.3710     | 3.1628   | 20             | -2.65 | 0.0155  | 0.0991 |
### Simple Effect Comparisons of speed_lev*loading Least Squares Means By loading_perc
Adjustment for Multiple Comparisons: Holm-Tukey

| Simple Effect Level (Nm) | speed_level | speed_level | Estimate | Standard Error | DF | t Value | Pr>|t| | Adj P |
|--------------------------|-------------|-------------|----------|----------------|----|---------|-----|------|
| loading_407              | 1500        | 1600        | -0.1831  | 3.1628         | 20 | -0.06   | 0.9544 | 1.0000 |
| loading_407              | 1500        | 1700        | 5.9764   | 3.1628         | 20 | 1.89    | 0.0734 | 0.3543 |
| loading_407              | 1600        | 1700        | 6.1595   | 3.1628         | 20 | 1.95    | 0.0657 | 0.3260 |
| loading_Full Throttle    | 1300        | 1400        | 10.9027  | 3.1628         | 20 | 3.45    | 0.0025 | 0.0191 |
| loading_Full Throttle    | 1300        | 1500        | -4.1441  | 3.1628         | 20 | -1.31   | 0.2050 | 0.6882 |
| loading_Full Throttle    | 1300        | 1600        | 11.6001  | 3.1628         | 20 | 3.67    | 0.0015 | 0.0118 |
| loading_Full Throttle    | 1300        | 1700        | 4.1540   | 3.1628         | 20 | 1.31    | 0.2039 | 0.6864 |
| loading_Full Throttle    | 1400        | 1500        | -15.0469 | 3.1628         | 20 | -4.76   | 0.0001 | 0.0010 |
| loading_Full Throttle    | 1400        | 1600        | 0.6974   | 3.1628         | 20 | 0.22    | 0.8277 | 0.9994 |
| loading_Full Throttle    | 1400        | 1700        | -6.7487  | 3.1628         | 20 | -2.13   | 0.0454 | 0.2450 |
| loading_Full Throttle    | 1500        | 1600        | 15.7443  | 3.1628         | 20 | 4.98    | <.0001 | 0.0006 |
| loading_Full Throttle    | 1500        | 1700        | 8.2982   | 3.1628         | 20 | 2.62    | 0.0163 | 0.1036 |
| loading_Full Throttle    | 1600        | 1700        | -7.4461  | 3.1628         | 20 | -2.35   | 0.0289 | 0.1693 |
APPENDIX D - MATLAB VEHICLE NETWORK TOOLBOX BASIC FUNCTIONS AND EXAMPLES

Matlab Vehicle Network Toolbox (VNT) provides functions that can be used to create a custom scripts for CAN bus communication as well as a GUI for accessing and logging CAN data. This GUI, called Vehicle CAN Bus Monitor, can be accessed from the Matlab Command Window with the command canTool or from the Matlab APPS menu. The Matlab VNT web page includes a comprehensive list of VNT functions along with many examples. The sections below describe some basic functions used in this study.

Matlab functions for basic J1939 communication

The short script provided below includes basic functions required to connect to CAN hardware and receive messages from the J1939 CAN bus. When the script is executed J1939 parameter groups are stored in the pgRx variable in the Matlab Workspace. This data structure can be saved as a Matlab data file (.mat) for future analysis. Continuous logging can be accomplished by using a loop function.

```matlab
% Load a database file configured for messages and parameters of interest.
  db = canDatabase('j1939DataBaseFile.dbc');

% Create a CAN channel for the hardware and CAN channel that will be used.
% Example channel setup for Vector:
  rxCh = j1939Channel(db,'Vector','CANcaseXL 1',1)
% Example channel setup for Kvaser:
  rxCh = j1939Channel(db,'Kvaser','Leaf Light v2 1',1)
```
% Configure CAN bus speed: generally 250k or 500k for off-road machinery with J1939.
    configBusSpeed(rxCh,500000);

% Now that the CAN channel has been configured, start the CAN channel.
    start(rxCh)

% Receive messages from the CAN channel and store them in a variable.
    pgRx = receive(rxCh,Inf)
    pgRx(1)

% When done receiving messages stop the CAN channel to release resources.
    stop(rxCh)

Import raw J1939 CAN data from Vector ASCII (.asc) files or Kvaser text (.txt) files

Matlab Vehicle Network Toolbox includes native function canMessageImport() that is intended to import CAN data in hexadecimal format from Vector ASCII (.asc) files or Kvaser text (.txt) files. These data files would be logged using Vector or Kvaser software. The canMessageImport() function uses a database file (.dbc) to decode hexadecimal CAN messages with the output being scaled engineering units. During this study it was discovered that this function did not work for J1939 messages. Matlab Technical Support confirmed this and provided j1939import () function as a work-around. The basic commands are provided below to show an example of how this function can be used, however, this work-around requires an additional Matlab file to be placed in a program directory. Additional details are not provided here as Matlab Technical Support indicated they intend to include this in future releases of Vehicle Network Toolbox. Figure 41
shows a plot of engine speed from CAN data that was decoded with the j1939import() function.

```matlab
% Load a database file configured for messages and parameters of interest.
    db = canDatabase('j1939DataBaseFile.dbc');

% Import Vector ASCII (.asc) file or Kvaser text (.txt) file.
% Example j1939 import for Vector file:
    pgs = j1939Import('VectorDataFile.asc', 'Vector', db);
% Example j1939 import for Kvaser file:
    pgs = j1939Import('KvaserDataFile.txt', 'Kvaser', db);

% Extract all signals from parameter group.
    pgs(1)
    pgs(1).Signals

% Get group of signals for Electronic Engine Configuration 1 (EEC1) parameter group
    pgs_EEC1 = pgs(strncmp('EEC1', {pgs.Name}));
    EEC1_Signals = [pgs_EEC1.Signals];

% Plot EngSpeed vs. EEC1 Timestamp
    plot([pgs_EEC1.Timestamp], [EEC1_Signals.EngSpeed]);
    hold on;

% Get group of signals for Electronic Engine Configuration 3 (EEC3) parameter group
    pgs_EEC3 = pgs(strncmp('EEC3', {pgs.Name}));
    EEC3_Signals = [pgs_EEC3.Signals];

% Plot EnginesDesiredOperatingSpeed vs. EEC3 Timestamp
    plot([pgs_EEC3.Timestamp], ...
         [EEC3_Signals.EnginesDesiredOperatingSpeed]);
    legend('EngSpeed','EnginesDesiredOperatingSpeed');
```
Sorting and visualizing CAN data with Vehicle Network Toolbox

The j1939import() function allows access to the decoded parameters but the data is still in a Parameter Group data structure. The Matlab commands below can be used to create a Matlab structure array with just the messages and their associated timestamps in a more concise and easy to use data structure.

```matlab
% Initialize data array.
msgStructure = []

% Get the number of messages in the database.
dbMessages = db.Messages;
numMsgs = length(dbMessages);
```
% Use for-loop to iterate through each message to extract the signals.
for i = 1:numMsgs

% Get Parameter Group name.
structName = char(dbMessages(i));

% Extract data for each respective message.
msgData = extractAll(rxMsg,structName);

% Test to see if the signals contain data.
if isempty([msgData.Signals])==true

% If the signals are empty then exclude them from the data structure.
else

% If the signals are not empty then place the message timestamp and message signals in the data array.
msgStructure.(structName).Timestamp = [msgData.Timestamp];
msgStructure.(structName).Signals = [msgData.Signals];
end
end

Matlab functions described above were used to create a simple GUI to extract, sort, and visualize J1939 parameters stored in a log file. The GUI shown in Figure 42 allowed the user to select a database file configured with messages and parameters of interest which was used to extract and decode parameters from a user selected log file. The user selected from a list of available parameter groups and then selected specific signals to plot for visualization.
Figure 42 Matlab GUI for decoding, sorting, and visualizing CAN parameters.