DEVELOPMENT OF A COMPUTER-CONTROLLED, HYDRAULIC, POWER TAKE-OFF (PTO) SYSTEM

R. S. Thomas, D. R. Buckmaster

ABSTRACT. *Nearly all tractor PTO arrangements used today consist of a rotating mechanical shaft with two or more universal joints. This arrangement continues to be an entanglement hazard. By driving the attachment with fluid power, this hazard could be eliminated. This project included the design, development, and testing of an experimental tractor based on a John Deere 4040. The entire power train was treated as a single computer-controlled system where both the PTO output and the wheels were driven by fluid power. To support direct laboratory and in-field comparison of hydraulic and mechanical scenarios, the original transmissions were left intact and functional. Results showed that at low power demand, the hydraulic system was more efficient than the mechanical equivalent; however, the mechanical version was more efficient over most of the load range and was capable of higher power output. The use of computer control helped to minimize fuel consumption by matching engine speed to loading conditions while maintaining operating speeds by varying hydraulic pump displacement. The hydraulic PTO system for the JD 4040 has the potential to match or increase fuel efficiency compared to its mechanical counterpart for loads up to about 40% of rated mechanical PTO output. Maximum power output in the hydraulic mode has the potential to be 75% of that achieved by the mechanical equivalent.*

Keywords. *Computer control, Fluid power, Hydrostatic transmission, PTO.*

The goal of this project was to examine the technical feasibility of replacing the current mechanical power take-off (PTO) design used on agricultural tractors. The approach taken was to treat the entire tractor drive and PTO as a single computer-controlled system. The benefits from such a design shift could produce dramatic improvements in both safety and versatility, and these improvements could be made today since hydraulic components large enough to drive any modern agricultural tractor are currently available. Such a design may contribute toward safer power transmission and help to eliminate the devastating injuries associated with PTO entanglement. Fluid power offers greater flexibility in machine design as well. Without the mechanical driveline, tighter turning radii are possible for field equipment. Fluid power has a high power-to-weight ratio and is an excellent match for electronic control. Since the use of fluid power lends itself so easily to electronic control, a number of other safety devices and control scenarios can be accommodated. Hydraulic wheel drive can easily and quickly be reversed or stopped. This rapid control coupled with tractor stability and position sensors could be used to interrupt or warn of imminent tractor rollover. The same quick response could be used in conjunction with the development of human presence sensors to shut down or reverse equipment upon detection. Ease of equipment reversal could also assist rescue teams in extricating people or be used for clearing clogs. Another possible advantage of fluid power with electronic control is ease of use. Further automating the control of a tractor may also reduce operator fatigue, thereby increasing both productivity and safety.

Nearly all tractor PTO arrangements used today incorporate a rotating mechanical shaft connected via two or more universal or constant-velocity joints. Commonly, PTO accidents involve the snagging of clothes, resulting in the victim being rapidly and violently drawn into (and around) the rotating shaft. PTO-related injuries are both common and severe. In 1998, 16 deaths were attributed to PTO entanglement (NSC, 2000). Extrapolating from National Safety Council (NSC, 2000) data, and data from Sell et al. (1985), one can estimate that 533 PTO-related injuries occur yearly, including 171 fractures, 139 amputations, and 16 deaths. PTO shaft shielding has been the industry answer to PTO safety, but inadequate maintenance is common and severely limits effectiveness. In a study of 2,513 tractors on New York farms, 45.1% had no master shield, and 3.4% of those present had defects (Chamberlain et al., 1998). Murphy et al. (1998) rated 49.4% of audited tractor master shields less than optimal. On PTO-driven machinery, 18.5% of driveline shaft shielding, 22.4% of machine master shielding, and 65.1% of driveline warning decals received less than optimal scores. Shielding has been used for decades, but it has not eliminated the safety hazard of PTOs. In a recommendation for change to ISO 5674, the Health and Safety Executive (1999) notes that while PTO safety is a perennial subject, accident numbers are still not falling significantly. One solution to the PTO entanglement hazard, and the focus of this research, is to eliminate this rotating mechanical tractor-implement connection completely.

In addition to eliminating entanglement injuries, hydraulic power as a replacement offers control flexibility, accom-

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The authors are **Roderick S. Thomas,** Engineering Associate, and **Dennis R. Buckmaster, ASABE Member Engineer,** Associate Professor, Department of Agricultural and Biological Engineering, The Pennsylvania State University, University Park, Pennsylvania. **Corresponding author:** Roderick S. Thomas, 249 Agricultural Engineering Bldg., The Pennsylvania State University, University Park, PA 16802; phone: 814-865-2252; fax: 814-863-1031; e-mail: rst5@psu.edu.

modating safety features such as interlocks, "dead man" controls, and equipment reversal to aid in extricating people (Morgan, 1992). Noting that the alteration of machinery design has a greater impact on the reduction of accidents than safety training, White et al. (2000), applied hydraulic power transmission in conjunction with an operator presence sensing system (OPSS) to a rotary mower. The research demonstrated the potential of such a system to detect a hazardous situation and stop the blade of a rotary mower quickly enough to avoid injury. Motivated by the potential to reduce product liability associated with PTO driveline injuries, Shearer et al. (1993) converted a large round baler from mechanical to fluid power drive. Shearer suggested that the potential for reducing PTO-related accidents is substantial and should be a major consideration for adoption of hydraulic drives for agricultural machinery.

Although fluid power has been suggested as a safer alternative to the traditional PTO for transmitting tractor power, it is not completely free from danger. The principle risk to an operator or mechanic working with high-pressure fluids is known as injection injury. This type of injury is caused by body contact with a stream of high-pressure fluid and results in the fluid being forced into the body under pressure. Such injuries can even be sustained by contacting a fine, nearly invisible stream from a leaking hose or fitting in a high-pressure hydraulic system. Although considered a serious injury, no fatalities due to injection injury were reported in the literature. Even though high-pressure hydraulic systems are common in agriculture, no agricultural safety survey reviews could be found that specifically addressed the issue of injection injury. The mining industry relies very heavily on fluid power (particularly using water-based fluids to reduce fire hazard associated with oils). As in the agricultural literature, no mining safety summaries could be found that directly addressed injection injury. While underreporting may be a factor, this lack of emphasis on the problem may also suggest that incidents resulting in significant injury are rare in these industries (Thomas and Buckmaster, 2003).

Low efficiency is often cited as a primary disadvantage of fluid power drives. One method for increasing the efficiency is through the use of an optimized engine-fluid power drive system (Lin and Buckmaster, 1996). Lin and Buckmaster (1996) developed a steady-state mathematical model to simulate a variable-displacement pump, fixed-displacement motor (VPFM) system powered by a diesel engine. The simulation indicated that, for an assumed load cycle, a hydraulic system with 88.8% average overall efficiency would be needed to match the fuel efficiency of a 95% efficient mechanical shaft.

Since poorer efficiency is one of the principle drawbacks to the application of fluid power, evaluating the power efficiency of the hydraulic system in comparison with current designs is key to proving feasibility. To support direct laboratory and in-field comparison of hydraulic and mechanical scenarios, a versatile engine/drive testing platform based on a John Deere 4040 tractor was developed. This tractor is typical of agricultural tractors in the 30 to 75 kW (40 to 100 hp) range being used on farms today and is suitable for operation of most PTO-driven equipment. Although typical, the John Deere 4040 cannot be considered state-of the-art because of its age. Today's best diesel engines are more efficient over a wider load range, are better governed, and have flatter torque curves. Since the John Deere 4040 is older, there is likely more efficiency to be gained through optimal engine speed control. There have also been advancements in the design of transmissions for agricultural tractors, such as the stepless gearboxes offered by Fendt, Claas, Steyr, John Deere, and others. Although the configurations differ, they all combine gears and hydraulics to produce hybrid transmissions with infinitely variable ratios. These transmissions improve on their mechanical counterparts by providing exactly the right ratio for a given task. Infinite ratios have been available with hydrostatic transmission for many years, but the new stepless transmissions are more efficient since only a portion of the power is supplied to the wheels using fluid power. The stepless transmissions offer improved performance in locomotion, but they do not yet address the issue of fixed PTO ratios. Modern tractors equipped with infinitely variable transmissions must still run with high engine speeds to provide standard PTO speeds for proper implement function.

Optimized computer control of a diesel engine requires a mathematical representation of its operation. Jahns et al. (1990) introduced functional techniques for describing engine performance in terms of fuel consumption, speed, and torque. The technique developed describes diesel engine performance using a polynomial containing nine coefficients (determined by measuring nine operating points). Jahns et al. (1990) reported that performance could be determined within an accuracy range of 0.2% to 2.5%. The resulting engine performance map can be further analyzed to produce an optimum performance equation that relates any given load point to a corresponding engine speed of best fuel efficiency.

Design of Experimental Tractor

The experimental tractor (fig. 1) was designed around a 67 kW (90 hp) John Deere model 4040 agricultural tractor. The tractor's original power train was left intact, including the mechanical PTO stub and planetary "power shift" transmission. In this way, a direct comparison between the hydraulic and mechanical systems could be made, and identical loading conditions could be provided by alternately switching between modes. Hydraulic power was delivered to the wheels and PTO using two separate, variable-speed hydrostatic transmissions controlled by a central computer. This computer was used to monitor power demand and adjust the engine speed and hydrostatic transmissions to meet desired operating speeds while keeping the engine near its optimal operating point. When treating the entire tractor as a system, some of the hydraulic system's inherent inefficiencies can be negated by more efficient engine performance (Thomas, 2001).

Hydraulic System Overview

Major components of the hydraulic drive system included an input gearbox, two hydrostatic transmission circuits (one to drive the wheels and one to drive the PTO), a reservoir, a cooler, and an electronic control system. The hydraulic pumps were driven by the original PTO through an additional gearbox.

The tractor control system consisted of input controls, a computer, and sensors including radar for ground speed, inductive zero-speed gear tooth sensors for shaft speeds,

Figure 1. Experimental tractor.

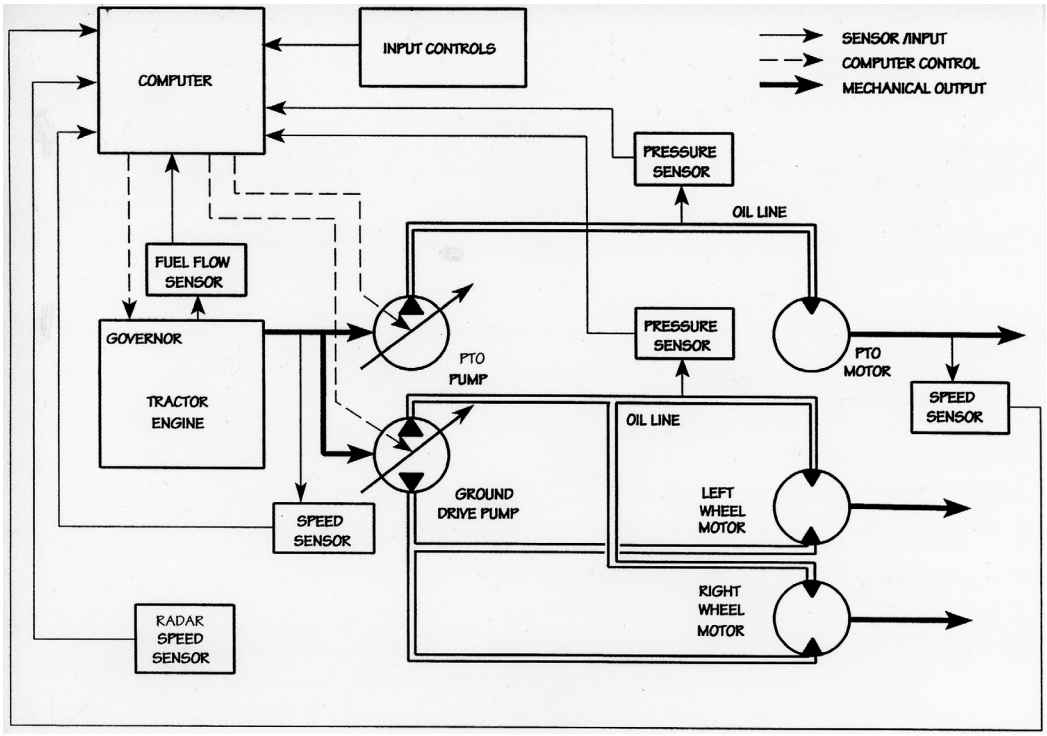

Figure 2. Functional block diagram of the hydraulic drive and electronic tractor control systems (filtration, charge circuit, loop flush, cooling, and reservoir are not shown).

hydraulic pressure transducers, and a fuel flow sensor (fig. 2). Travel speed was input via a joystick, while direction was via a rocker switch, and PTO speed was input using a rotating knob.

PTO Drive System

The hydraulic PTO was driven by an open-loop hydrostatic transmission consisting of an Ifield V60 bent-axis axial piston pump with a displacement of 57.6 cm^3 rev^{-1} (3.51 $in.^3$ rev^{-1}) and a HTL MHA500 C radial piston motor with a displacement of 508 cm^3 rev^{-1} (31.03 $in.^3$ rev^{-1}). System pressure was limited to 34,500 kPa (5,000 psi) by a fast-acting relief valve near the output of the pump. Cooling flow through the case was provided by the wheel drive closed-loop charge pump via a flow control valve set to 13.6 L min^{-1} (3 gpm).

The PTO pump was coupled directly to the auxiliary drive pad at the rear of the wheel drive pump. With a final drive ratio of 1:3.2 provided by open spur gears, a PTO output of 540 rpm was achieved with the hydraulic drive motor turning at 169 rpm. This speed was nearly ideal in terms of overall efficiency for the motor. Considering volumetric inefficiencies and transmission ratios, a minimum calculated engine speed of 1,625 rpm was needed to achieve an output speed of 540 rpm.

PTO drive system pressure was limited to 34,500 kPa (5,000 psi). The maximum theoretical output torque was 2,790 N m (2,058 ft lb) at the motor shaft. Translated through the 1:3.2 final drive, and using the manufacturer's "nominal running torque" values, maximum design output was 786 N m (578 ft lb) and 44.5 kW (59 hp). This level of output was chosen based on the expected available input torque and the predicted efficiency of the hydrostatic transmission itself. Power output higher than this was not deemed possible while driving the pumps via the PTO and speed increaser gearbox.

Computer System

Control, indication, and data logging were handled by a Pentium 150 MHz computer. Input/output was interfaced by two National Instruments DAQCard-1200 PCMCIA cards. National Instruments LabView 5.0 running under Microsoft Windows 95 performed most of the data collection, conversion, control, and indication functions. Since the counter/timer chips used in the DaqCard-1200 are not well suited to monitoring the number and frequency of pulse output sensors used on the tractor, measurement and conversion of frequency output signals was handled by an auxiliary Basic Stamp IIsx microcontroller.

One of the key functions of the computer was to measure the power demand, through pressure and speed data, and determine the proper engine speed to minimize fuel consumption. The determination was based on the optimal performance line developed from empirical data and the technique of Jahns et al. (1990). The following optimal performance equation estimates optimum engine speed as a function of power demand:

$$N_e = 2585(1.055 - 3.458 - {}^{P_{kW}}/^{75.88}) \ (r^2 = 0.9996)$$

$$N_e = 2585(1.055 - 3.458 - {}^{P_{hp}}/^{56.58}) \ (r^2 = 0.9996)$$

where

N_e = optimal engine speed (rpm)
P_{kW} = engine power demand (kW)
P_{hp} = engine power demand (hp)

Results and Discussion

The curve representing the performance of the PTO hydraulic system includes an efficiency correction factor. This factor is an estimated adjustment for power loss through the mechanical transmission between the engine and hydraulic pumps. To maintain the ability to compare hydraulic and original mechanical modes, it was necessary to leave the as-built mechanical PTO intact and use it to drive the hydraulic pumps. A gearbox was added between the PTO and pump input to facilitate driving the pumps at approximately engine speed. In effect, the engine speed was geared down and then back up to the original speed. If a production version were developed, the pumps could be driven at engine speed directly. These mechanical transmission losses would be avoided and were considered in analysis. Deere and Company (1999) suggests using a PTO drive efficiency rating of 87% for calculating power loss from the engine to the PTO. This means that only 87% of the net engine power is available at the PTO shaft. This figure is consistent with *ASAE Standards* (2003), which indicates that 87% to 90% of net engine power is typically available at the PTO. Remanufacturing specifications used by John Deere for this engine stipulate setting the fuel pump for a gross engine output of 82 kW (110 hp). Allowing 6 hp for the fan, and a PTO output of 90 hp, yields an estimated efficiency of 86.5%, which is consistent with the above efficiency estimates. Manufacturer's data for the added speed-increasing gearbox suggest an efficiency of 96% for this application. Using 87% for the tractor transmission and 96% for the added gearbox yields an estimated drive efficiency of 83.5% (16.5% loss). The performance curve was generated by reducing the specific fuel consumption by 16.5%. To estimate the increased output, the hydraulic mode curve was extended horizontally to the right by 16.5% for points beyond the measured maximum torque.

Control Systems

The control loops were subjected to a rapid command change under no-load conditions, and the resulting response was recorded. The maximum acceptable deviation from the target speed during steady-state operation is defined as the tolerance band. A design tolerance band of ±100 rpm was selected for the engine speed loop. Engine speed response was tested using an abrupt command change from 1578 rpm to 2279 rpm. The rise time, defined as the time interval between 10% and 90% of the commanded amplitude change, was 2.7 s with a peak value of 2352 rpm. Overshoot was 73 rpm and within the design tolerance band. Settling time, the time it takes for the response to reach and stay within its tolerance band, was 8.1 s. The system response was a cumulative result of the entire system including the engine, governor, actuator, sensor, electronics, and computer with their associated errors and lag times. Although stable enough for testing, the time lag and overshoot were perceptible to the operator and were considered less than ideal. Fortunately, there was adequate torque reserve when operating on the optimal speed curve, so engine stall was not a problem.

A command input step from 456 to 547 rpm was used to test hydraulic PTO response. Rise time was 1.4 s. Speed peaked at 565 rpm for an overshoot of 18 rpm. Settling time was 9.4 s using a tolerance band of ±10 rpm. The response of the PTO loop was marginal at best. Control of reasonably steady-state loads was not a problem, but control of varying

loads was slow and inaccurate. Although the rise time of 1.4 s was acceptable, there was a delay of 2.7 s after the change of target speed where no response was measurable.

Performance of the control loop was limited by three major factors. First, and most significant, was the pump design. As with all positive-displacement pumps, output pressure is relatively low with light loading and increases with a corresponding increase in load. Since the design of this particular pump utilizes output pressure to supply the displacement control servo, the speed of pump response and the sensitivity to electronic control correspondingly vary with speed and load. The second major factor limiting performance was sluggish speed measurement. Shaft speed data from the microcontroller were only available to the computer approximately once per second. This means that the control program was acting on data that were already about 1 s behind actual conditions. Finally, the computer control program itself contributed to system lag. The program loop time was approximately 250 ms when no speed data were being transmitted from the microcontroller and 450 ms with speed data transfer. Loop gains had to be severely limited to accommodate the pump response variability as well as data and loop delays. The slow response was compounded, to the point of potential instability, when all three loops (including ground drive) were running simultaneously with varying loads.

PTO Dynamometer

PTO testing was performed using an AW model NEB 400 dynamometer. Specific fuel consumption vs. power is displayed in figure 3. The data indicate that the efficiency of the hydraulic system was competitive with the mechanical system up to approximately 27 kW (36 hp) load. This is attributed to the ability of the system to adjust engine speed according to load while still maintaining output speed. The mechanical system was more efficient beyond this point and reached a maximum of 58 kW (78 hp) at 540 rpm compared to 44 kW (59 hp) for the hydraulic system. In terms of specific fuel consumption, the mechanical mode curve shows about 25% lower fuel consumption than the hydraulic mode at high loads, while the hydraulic mode was more efficient than the mechanical mode at 10 kW by about 18%. The tractor used in this research achieved only 58 kW (78 hp) at the PTO compared to the manufacturers specification of 67.1 kW (90 hp) and the Nebraska Tractor Test maximum of 67.8 kW (90.9 hp) (Nebraska Tractor Test Laboratory, 1980). Since the fuel consumption data collected for this project were in reasonable agreement with the Nebraska Tractor Test data, it was concluded that the reduced maximum power output was due to reduced maximum fuel flow and not from a problem that would affect the validity of the test results. No adjustments were made to the fuel system.

Liquid Manure Storage Agitation Pump

The liquid manure storage agitation pump used was a single-speed, 540 rpm input, model 12FT.PP manufactured by Imperial Calumet Industries. The pump operated consistently in both hydraulic and mechanical modes; however, the load was heavy for the hydraulic mode. Although operation was smooth, the engine could only maintain an average speed of 1633 rpm at full throttle (table 1). The engine speed was depressed below the operating point specified by the optimal performance equation due to the heavy load. Specific fuel consumption for the mechanical run was 0.49 kg kW-h^{-1} (0.81 lb hp-h^{-1}). The efficiency-corrected specific fuel consumption for the hydraulic mode averaged 0.48 kg kW-h^{-1} (0.79 lb hp-h^{-1}), representing a 2.0% improvement in fuel efficiency over the mechanical mode. The difference between specific fuel consumption means was significant at the 0.01 level using a two-sample t-test with unequal variances.

The fuel efficiency results were more favorable for the hydraulic mode than would have been predicted based on earlier testing. These results, however, must be viewed with caution since the engine speed was lower than intended.

Forage Harvester

A New Holland model 782 forage harvester fitted with a two-row corn head was used to test combined wheel drive and PTO loading in the field. The harvester was satisfactorily operated in randomized strips of corn at 3.2 km h^{-1} (2 mph). Both modes worked satisfactorily, with the exception of plugging. Periodic plugging of the harvester was a problem in both mechanical and hydraulic modes, but was worse in the

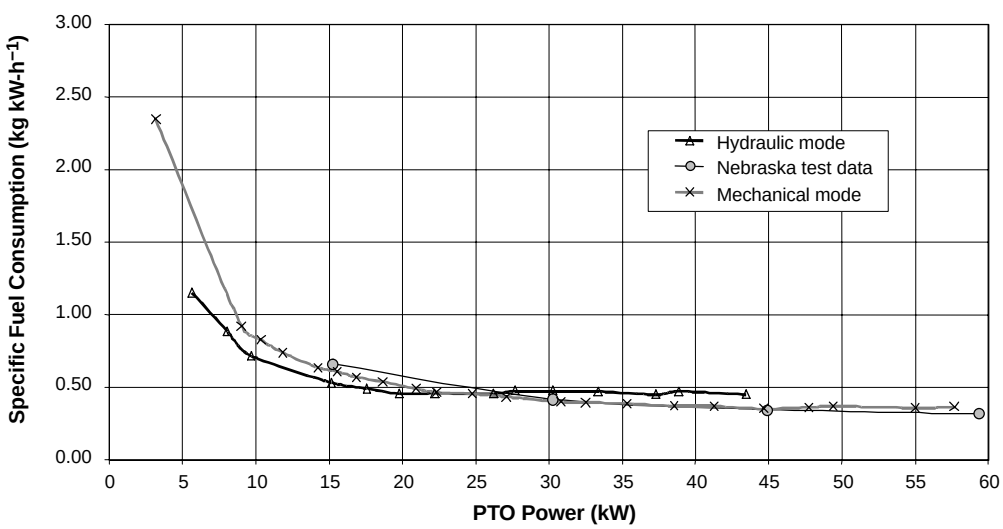

Figure 3. Specific fuel consumption vs. PTO power for the mechanical and fluid drives.

Table 1. Summary of mechanical and hydraulic mode performance with three implements.

	Agitation Pump		Harvester		Roller Mill	
	Mechanical	Hydraulic	Mechanical	Hydraulic	Mechanical	Hydraulic
Engine speed (rpm)	2250	1630	2240	2110	2220	1635
PTO speed (rpm)	553	537	549	549	545	558
Ground speed (km h^{-1})	--	--	3.07	3.16	--	--
Specific fuel consumption (kg kW-h^{-1})	0.49	0.48	0.41	0.37	1.70	1.24

hydraulic mode of operation. Presumably this was due to the lower maximum power available at the PTO. When a potential plugging situation arose, the mechanical mode had more reserve power available to clear the material. This problem would be reduced if the hydraulic pumps were driven directly by the engine, thereby eliminating the losses through the mechanical transmission. It would require an engine on the order of 30% larger with a corresponding increase in hydrostatic transmission capacity to perform all of the same tasks as the mechanical version.

As shown in table 1, mean mechanical mode specific fuel consumption was 0.41 kg kW-h^{-1} (0.67 lb hp-h^{-1}). Specific fuel consumption for the efficiency-corrected hydraulic mode averaged 0.37 kg kW-h^{-1} (0.61 lb hp-h^{-1}). Fuel consumption was 9.8% lower in the hydraulic mode than in the mechanical mode. Average travel speed was 3% higher in the hydraulic mode. The difference in specific fuel consumption was significant at the 0.01 level using a two-sample t-test with unequal variances.

Roller Mill

A 540 rpm, model K, PTO-driven roller mill from Roskamp Manufacturing, Inc., was used to process dried kernel corn. The corn was unloaded from a gravity-feed wagon into an auger, which kept the input hopper of the roller mill steady and full. Operation was problem-free in both modes of operation.

The roller mill was an example of a light load for this tractor and had a very steady torque requirement. Operation of this equipment represents the situation where the hydraulic mode has the greatest advantage over the mechanical mode. Hydraulic mode specific fuel consumption was lower than in the mechanical mode even before correcting for drive transmission losses. As shown in table 1, mean specific fuel consumption for the mechanical system was 1.70 kg kW-h^{-1} (2.79 lb hp-h^{-1}), while the efficiency-corrected specific fuel consumption for the hydraulic mode was 1.24 kg kW-h^{-1} (2.04 lb hp-h^{-1}). The efficiency improvement of the hydraulic mode compared to the mechanical mode was 27%. The average PTO speed in hydraulic mode was 558 rpm, compared to 545 rpm in the mechanical mode. The difference in specific fuel consumption was significant at the 0.01 level using a two-sample t-test with unequal variances.

Summary and Conclusions

At low power demand, the hydraulic system, combined with optimal engine control, was more efficient than the mechanical equivalent; however, the mechanical version was more efficient over most of the load range and was capable of higher power output. The following conclusions can be drawn from this research:

- A hydraulic PTO system for the JD 4040 could match or improve fuel efficiency compared to its mechanical counterpart for loads up to about 40% of the rated mechanical PTO output.
- Maximum PTO output in the hydraulic mode has the potential to be 75% of that achieved by the mechanical equivalent.
- A 30% more powerful engine would be required for a hydraulically driven tractor to perform all the same tasks as the mechanical equivalent.

Recommendations

Adoption of the innovations explored in this research would be a huge shift for the industry. This being the case, all doubts about the value of safety, the feasibility of fluid power, and the economics of such a shift must be addressed first. For example:

- A profile of typical yearly loading for farm tractors could not be found in current literature. The most current data found were from 1961. Good loading data would allow engineers to optimize engine and drive design, and allow a proper and thorough economic analysis.
- A full-scale prototype tractor with directly driven hydraulic pumps is a logical next step for research in this area. This second prototype may include hybrid fluid/mechanical drives for both locomotion and PTO drives for even higher efficiency.
- Implements and attachments optimized for fluid power need to be developed, tested, and evaluated.

A complete and thorough economic evaluation needs to be completed. This evaluation should include cost advantages associated with mass production, lifetime operating costs, and the value of improved safety considering both the tractor and implement sets.

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