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Proposed Design Modifications to Reduce Risk of Operating Rotary Field Mowers

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Abstract

The primary objective of this project was to reduce risk of injury associated with operating a rotary mower driven by a tractor power take-off (PTO) by developing and evaluating design improvements and determining their economic feasibility. Researchers have concluded that alteration of machinery design has a greater impact on the reduction of accidents than safety training. Implementation of an Operator Presence Sensing System (OPSS) and removal of the PTO are the two injury-reducing, engineering modifications evaluated by this research. Hydraulic power allows this to occur by providing dynamic braking, few moving parts (removal of the PTO), and controllable power. A hydraulic circuit was developed to power the mower and to enable an OPSS. Tractor hydraulics were simulated using a hydraulic training bench. Two mower configurations were tested: 6.55 cm^3 rev^{-1} (0.4 $in.^3$ rev^{-1}) displacement motor with a 0.748 kg blade and 47.5 cm^3 rev^{-1} (2.9 $in.^3$ rev^{-1}) displacement motor with a 9.4 kg blade. A PTO-driven rotary mower was not used to test the circuit due to spatial and safety limitations of the hydraulic training bench. Results from the first mower configuration verified the concepts behind the hydraulic circuit. The second configuration verified the OPSS and indicated the applicability of the circuit to a rotary mower.

Keywords. Agricultural safety, Rotary mowers, Mower safety, Deadman controls.

The use of PTO-driven rotary mowers can result in injury of consumers and in product liability for manufacturers. Rotary mowers comprise the following common machine hazards that contribute to a concern for user safety: power take-off PTO) shafts, shear-cutting points, and freewheeling parts (Murphy and Morrow, 1996).

Rotary mowers cause additional concern because they rarely inflict minor injuries, instead rotary mower accidents are generally associated with mangled appendages and traumatic fatalities (Wilkins, 1981).

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Data collected by Kentucky Fatality Assessment And Control Evaluation (FACE) between 1994-1997 indicated that the majority of reported fatalities associated with rotary mowers were a result of persons falling from a tractor, tractor turnover, unexpected movement of equipment, and being run over by equipment. During these types of accidents, an operator is at high risk of coming into contact with shear-cutting points and freewheeling parts. This presents a life-threatening situation for an operator due the severity of injuries inflicted by these components.

The multiple risks associated with operating rotary mowers also affect the manufacturing industry. According to Pawlak (1990), manufacturers of farm machinery face a serious financial burden because of product liability. Manufacturers need to implement high safety standards in order to foresee and avoid possible product liability. Lewis et al. (1998) concluded that meeting such standards could be accomplished by improving equipment design.

The intent of this project was to combine the knowledge of engineers and epidemiologists to evaluate the design, operation, maintenance, and work habits related to injuries and fatalities associated with PTO-driven rotary mowers. Investigation of these safety issues has led to the proposed implementation of an Operator Presence Sensing System (OPSS) and replacement of the mechanical PTO drive with a fluid drive to reduce the risk associated with operation of a rotary mower. In addition to eliminating hazardous PTO shafts, fluid drive systems can provide the quick reaction time required by the OPSS. A quick stopping time is essential to protect the operator from the rotating blades of the mower. This project examined the feasibility and the potential risk reduction associated with these two modifications. The ultimate project goal is to decrease the injury and fatality rate among people using rotary mowers. The specific objectives of the study were to: (1) design and fabricate a fluid drive system for a rotary mower with dynamic braking capability; (2) simulate an OPSS and experimentally determine the time required to stop a simulated mower blade; and (3) assess the operational potential of the proposed system for powering a commercial rotary mower and stopping the cutting blade within 1.5 s of receiving an OPSS signal.

Background

Numerous studies are available indicating that PTO-driven rotary mowers are a common source of accidents on farms. A study of injuries of Kentucky farmers 55 years and older by Browning et al. (1998) showed that 18% of machine-related injuries involved rotary mowers. Piercy et al. (1984) investigated fatalities occurring between the years 1972-1982 on Kentucky Farms and discovered that the most common implement involved in machinery accidents was the rotary field mower. Gehlhausen (1995) cited a collection of newspaper articles regarding accidents associated with rotary mowers from 1984 to 1994. Three categories of accidents were identified: 36 involving people hit by trailing mowers after falling from tractors, 27 involving operators falling from tractors and being run over by the tractor, and 9 involving operators being hit by other equipment after falling from the tractor. Another study conducted by Buchele (1993) claimed that rotary mowers were the most dangerous farm machines, stating that approximately 22% of rotary mower accidents caused permanent injury.

The risks involved with operating rotary mowers are also a concern of manufacturers. Gehlhausen (1995) discussed the extent of manufacturers' product liability for accidents involving tractor-pulled rotary mowers. Liability for accidents using the PTO-driven mowers rests with both tractor and mower manufacturers.

Lewis et al. (1998) stated that the availability of affordable safety devices often increases manufacturers' liability when not implemented. They concluded that the best defense for manufacturers against liability expenses is improvement of equipment design because of a greater impact on the reduction of accidents than safety training. Murphy and Morrow (1996) proposed two design modifications to address safety concerns associated with operating farm implements such as rotary mowers: the installation of a rapid-response OPSS, and the elimination of the PTO driveline. An OPSS disengages power sources when an operator leaves a vehicle's seat (or other operating position). This safety device helps to prevent injury or fatality when an operator falls from the seat or the tractor overturns (Gehlhausen, 1995). One implementation of an OPSS required by the Consumer Product Safety Commission is on the walk-behind lawnmower. Generally located on the mower's handle bar, the system stops the mower when the user is not grasping the handle bar. The reduction in injuries documented after its implementation suggests that this type of safety device helped reduce accident rates (*Consumer Reports*, 1994). According to Murphy and Morrow (1996), some redesign is needed to improve present OPSS systems. The main problem is achieving the 0.5 s stopping time desired to effectively reduce walk-behind mower injuries (Buchele, 1993). Murphy and Morrow (1996) also pointed out the need for machine designers to make bypassing an OPSS system a difficult task for consumers in order to prevent the disabling of such systems.

Equipment manufacturers installed shielding to minimize injuries associated with people making contact with rotating PTO shafts (Bornzin, 1973). Sell et al. (1985) reviewed 72 PTO-related accidents and found four leading causes: PTO's engaged while equipment was stationery, inadequate shielding of PTO shafts, protruding push pins or bolts, and victims positioning themselves too near rotating PTO shafts. They reported that PTO shafts were the second leading cause of farm machinery accidents. Shearer et al. (1993) reported that PTO injuries are often associated with shielding being improperly installed, removed or not providing complete coverage.

Fluid power, as defined by Stewart and Storer (1968), "denotes the technology that deals with the transmission and control of energy by means of pressurized fluids". Fluid power was first used in the agricultural industry in the 1930s to allow operators to raise and lower implements from their seated position (Stepanek et al., 1995). Since the 1930s, fluid pressure and flow capacity available for auxiliary use on tractors have steadily increased from 2 MPa (300 PSI) and 12 L min^{-1} (3 gpm) to 20 MPa (3000 PSI) and 210 L min^{-1} (55 gpm).

An advantage of using fluid power, versus mechanical power, is the elimination of a complicated system of cams, gears, levers, etc. Since these mechanical parts are not used, there is also an avoidance of system overload, which causes catastrophic failure. In a fluid power system, a relief valve protects the system from these types of overloads. Fluid drives also facilitate greater control of continuous transmission speed. The disadvantages of fluid power systems are cost and inefficient energy transfer. Shearer et al. (1993) modeled a PTO system to have 95% efficiency (constant) and a fluid drive to have between 57-76% efficiency (variation dependent on power demand). One way to increase efficiency is to select component sizes correctly and to optimally control the power source (Lin and Buckmaster, 1994). Recent developments, such as reduced cost and increased life of hydraulic components, have led to increased utilization of fluid drives as an alternative to mechanical drives.

Researchers have demonstrated that mechanical PTO drives on implements can be replaced with fluid drives. Shearer et al. (1993) replaced a mechanical PTO with a fluid drive for a round baler. The hydraulic circuit developed for this fluid drive utilized a pressure relief valve to protect the system from catastrophic failure and a check valve to prevent oil cavitation when the pump was disengaged. Morgan (1992) described a Massey-Ferguson 1020H (Hydrostatic) tractor that used a fluid drive to power a rotary mower. He described this tractor and mower combination as a White Hydraulics, Inc. research vehicle and noted that additional research was required in order to gain better knowledge of the system and enable commercial marketing of the design. He suggested that the addition of larger displacement pumps on small tractors is critical for wider utilization of fluid drives for implements. Without larger pumps on tractors, additional pumps have to be retrofitted in order to operate some implements. The addition of larger pumps by tractor manufacturers would be less expensive than retrofitting, and this would help decrease the cost for users who are interested in powering implements using fluid drives. This study assumes the tractor has the necessary hydraulic capabilities (pressure and flow) available.

Materials and Methods

Hydraulic Circuit

A hydraulic circuit was designed and fabricated to implement an OPSS on a rotary mower (see fig. 1). Manipulation of solenoid valves in the circuit allowed for stopping the mower blade. A Parker DW3 solenoid directional control valve provided fluid control from the pump, through the system, and back to the tank. A Parker DS201 solenoid bypass valve stayed in the open position unless it was energized. These two valves switched to the closed position when they received a signal simulating that an operator had left the operating location. The closing of these two valves diverted fluid to a Parker RD103 sequencing valve, which opened when the pressure exceeded 2.07 MPa (300 psi). This allowed fluid to flow through the sequencing valve, through an adjustable Parker N-series orifice (or needle) valve, and through a Parker CV103 check valve to the motor. The check valve allowed fluid to flow back to the motor inlet during dynamic braking to prevent cavitation. The pressurized fluid quickly dissipated its energy in the form of heat as it passed through the orifice and traveled along the braking subcircuit. The braking subcircuit consisted of the sequencing valve, the adjustable orifice, the pressure relief valve, the

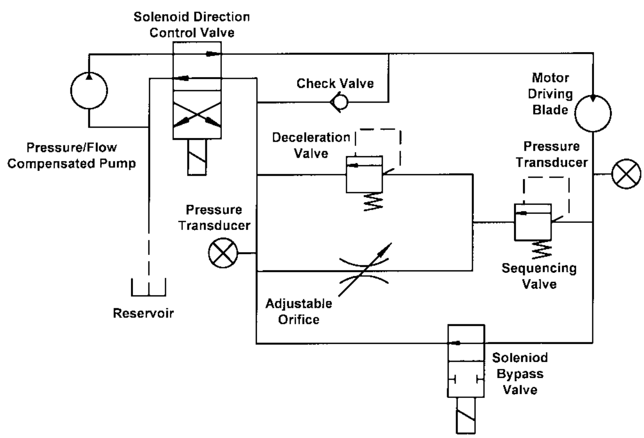

Figure 1–Schematic of the experimental hydraulic circuit.

check valve, and the motor. If pressure exceeded 17.9 MPa (2,600 psi), the Parker RD103 pressure relief valve opened to protect the components of the hydraulic circuit. Pressurized fluid was provided by a pressure- and flow-compensated variable displacement pump. Flow paths were constructed with 1.27 cm hose and 1.9 cm steel tubing. The hydraulic circuit was sized for a flow rate of 45.4 L min^{-1} (12 gpm) with a maximum operating pressure of 20.7 MPa (3000 psi). This number was determined using current applications of hydraulics with rotary field mowers. Figure 2 shows the experimental driving/braking circuit mounted on a laboratory hydraulic test bench.

Omega thin film voltage output pressure sensors (PX213 series) measured pressure downstream from the motor when the solenoid valves were closed (during OPSS triggering, see figs. 1 and 2). Fluid pressure was measured during dynamic braking to identify conditions for which damage to circuit components might occur. A Turck inductive sensor motion detector (see fig. 2) mounted near the blade detected blade rotational speed and the time required to stop the blade once the OPSS was triggered. A Computerboards DAS-08 board collected information from the pressure transducers and the motion detector. The pressure transducers required analog-to-digital conversions, while the motion detector used a counter-timer. During the experimentation, pressure readings and blade rotational speed were monitored. Pressure transducer and motion detector readings were taken at intervals of 0.01 s and 0.1 s, respectively. A Visual Basic program collected input data from the pressure transducers and motion detector and stored the data in a file. This program was developed from files provided in the Universal Library.

Two toggle switches were used to simulate an OPSS. The switches shifted the positions of the solenoid directional control valves by connecting/disconnecting

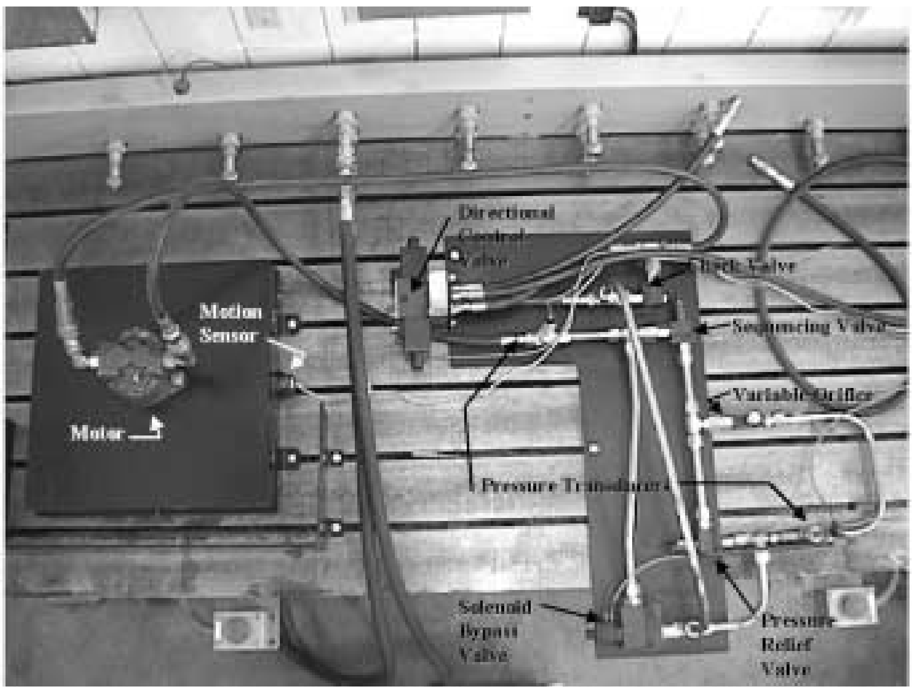

Figure 2–Photograph of experimental hydraulic circuit on the laboratory test bench.

12Vdc. Each test began with both valves open. After the blade reached desired speed, the two toggle switches were simultaneously activated to close the valves, thus simulating an OPSS to initiate dynamic braking and stoppage of the blade.

Blade/Motor Configurations

A small blade and motor (configuration 1) were used to test the hydraulic braking circuit in the laboratory. The blade dimensions were 52.71 cm × 5.72 cm × 0.32 cm and the blade mass was 0.755 kg (see fig. 3). The volumetric displacement of the hydraulic motor was 6.55 cm^3 and, with oil flow rate adjusted to 21.2 L min^{-1}, the blade speed was approximately 2900 rpm. The mass moment of inertia of the blade and motor assembly of configuration 1 was computed and is shown in table 1.

The blade was positioned in an enclosed steel box to protect operators and observers. This box, along with the hydraulic circuit, was mounted on the test bench

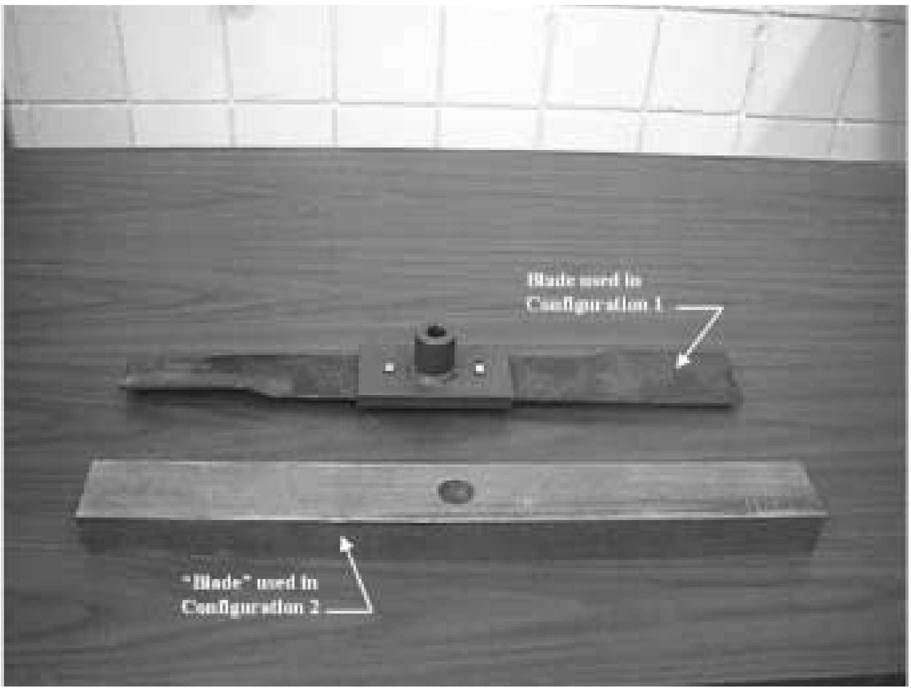

Figure 3–Photograph of blades attached to motor shafts and used to test the ability of the experimental circuit to stop blade rotation.

Table 1. Mass moments of inertia of the blade and various motor components used in testing configuration 1

Component	Moment of Inertia (kg m^2)	Percent of Total
Blade	0.0177	86.3
Hub	0.00272	13.2
Shafts and gears	0.00009	0.39
Oil	0.0000025	0.012
Total	0.0205	100

as shown in figure 2. The cracking pressure of the pressure relief valve was set at 17.9 MPa.

Eight orifice settings were investigated using configuration 1. Different settings of the orifice were described in terms of revolutions of the orifice control knob starting with the orifice completely open. The various settings were 3, 3.5, 4, 4.5, 5, 5.5, and 6 revolutions. The orifice was also tested completely closed. Each orifice setting was tested five times to determine an average of stopping times and maximum braking pressure. The motion sensor and pressure transducers monitored the stopping times and downstream fluid pressure, respectively.

A commercial mower equipped with a 1.52 m blade operating at 730 rpm develops an approximate kinetic energy of 12.6 kN m. The small blade and motor assembly of configuration 1 produced a maximum kinetic energy of approximately 0.43 kN m. Thus, in order to test the circuit under conditions more comparable to that of a field mower, a larger motor and blade combination was utilized.

The second blade/motor configuration (configuration 2) was assembled to develop greater kinetic energy and, therefore, provide a more realistic evaluation of the experimental circuit. In configuration 2, a steel bar, 53.3 cm long, 5.72 cm wide, and 3.18 cm thick with a mass of 9.40 kg was used to simulate a blade (see fig. 3). The bar was mounted on a 47.5 cm^3 rev^{-1} (2.9 $in.^3$ rev^{-1}) displacement motor and was connected to the experimental drive/braking circuit. This was the most massive “blade” that could be tested using safety shielding that was fabricated and mounted on the hydraulic test bench. The mass moment of inertia of the blade and motor assembly of configuration 2 was computed as 0.237 kg m^2.

The maximum fluid delivery rate of the hydraulic test bench was approximately 106 L min^{-1} (28 gpm) thus, the maximum rotational speed achievable in configuration 2 was approximately 2,000 rpm. The resulting kinetic energy (KE_2) developed in configuration 2 was then computed as 5.2 kN m. This is approximately 41% of the kinetic energy calculated for the commercial mower blade. Table 2 is a compilation of the parameters and attributes of the two blade and motor configurations.

Seven experiments were administered using configuration 2. For six of the experiments, the orifice was modified to create new system dynamics. Orifice settings of 1, 2, 3, 4, 5, and 6 turns were tested with the pressure relief valve cracking pressure set at 17.9 MPa. A seventh test was conducted at an orifice setting of six turns and at an increased pressure relief valve cracking pressure of 20 MPa. Each test was replicated three times to determine an average blade stopping time and fluid pressure during dynamic braking. Data acquisition equipment collected these measurements.

Table 2. Parameters associated with an experimental braking circuit tested using two blade and motor configurations

	Configuration 1	Configuration 2
Moment of inertia (blade and motor) (kg m^2)	0.0205	0.237
Maximum speed (rpm)	2,900	2,000
Maximum kinetic energy (kN m)	0.43	5.2
Motor displacement (cm^3 rev^{-1})	6.55	47.5

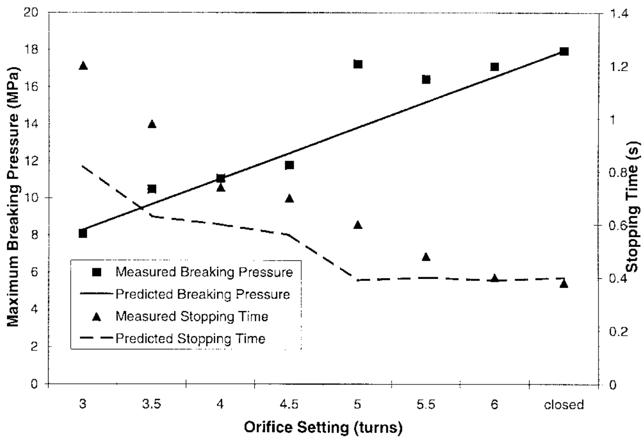

Figure 4–Measured and predicted average maximum braking pressure with measured and predicted time to stop blade rotation versus orifice setting in testing configuration 1.

Results and Discussion

Configuration 1

Average measured maximum pressures and blade stopping times versus orifice settings in configuration 1 are presented in figure 4. The maximum pressure measured when the orifice was closed corresponds to the cracking pressure of the pressure relief valve (17.9 MPa) as expected. Maximum measured pressure was approximately proportional to orifice setting from orifice openings of three to four and one-half turns. At five turns the maximum pressure increased to the approximate cracking pressure of the pressure relief valve. The overall variation of maximum pressure with orifice opening was nearly linear, with the notable exception of the substantial increase at the five-turn orifice setting.

Average measured blade stopping times decrease as orifice opening increases as expected. The predicted stopping times (t_s) shown in figure 4 were determined as follows.

The torque required to decelerate the blade and motor is given by:

$$T_{inertia} = I\alpha = I \frac{\overline{\omega}_{max}}{t_s} \tag{1}$$

where the moment of inertia of the motor and blade (I) is 0.0205 kg m^2 and uniform angular deceleration ($\overline{\omega}_{max}$) was assumed from 2,900 rpm to rest. This torque was assumed to equal the torque applied to the motor by the pressurized oil, or:

$$T_{motor} = \zeta_t V_{d,m}\, p \tag{2}$$

where ς_t is the motor torque efficiency, $V_{d,m}$ is the motor displacement volume, and p is pressure created by the braking circuit. Equations 1 and 2 were solved for t_s by assuming that the measured maximum braking pressure was sustained throughout the breaking time interval, t_s. This should result in underprediction of stopping time as breaking pressure would be expected to decrease from a maximum value to zero during t_s. However, for the most restricted valve settings (5.5 to 6.5 turns), figure 4 shows that predicted and measured stopping times were approximately equal. As the opening of the adjustable orifice increased, predicted stopping times fell increasingly below those measured, indicating that the ratio of average to maximum breaking pressure decreased as the orifice opening increased. If we represent braking pressure versus time as:

$$p = p_{max}\left[1 - \left(\frac{t}{t_s}\right)^n\right] \tag{3}$$

then the measured stopping time at the orifice setting of three turns is approximately equivalent to an average braking pressure computed with n = 2 in equation 3. For orifice settings ≤ 5.5 turns, braking pressure variation corresponded to 2 ≤ n ≤ 5 in equation 3.

Configuration 2

Figure 5 presents average maximum measured braking pressure for configuration 2 along with predicted maximum braking pressure. Again, predicted maximum braking pressure was computed as decreasing proportionally with orifice opening from the cracking pressure setting of the pressure relief valve (17.9 MPa). However, in this configuration the maximum braking pressure reached the cracking pressure of the pressure relief valve at the three-turn orifice setting and was greater than cracking pressure above that setting. The assumption of linear pressure variation substantially under-predicted maximum braking pressure.

The prediction of stopping times using equations 1 and 2 agreed well with measured results. This indicates that braking pressure remained approximately equal to p_{max} during most of the braking time, t_s. Also, the maximum stopping time at the

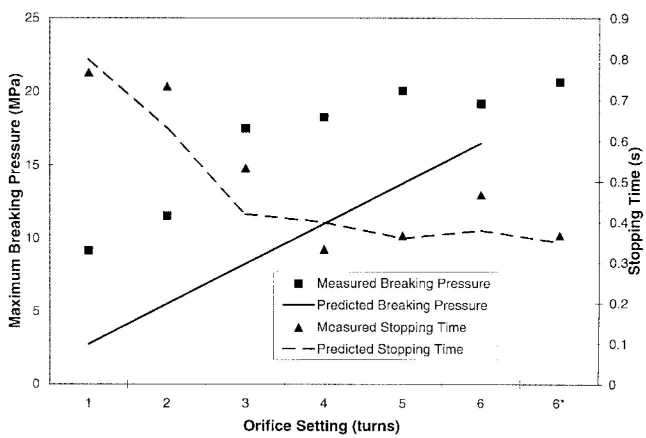

Figure 5–Measured and predicted average maximum braking pressure with measured and predicted time to stop blade rotation versus orifice setting in testing configuration 2.

most open valve setting was substantially less than the required or design stopping time of 1.5 s.

The observation at setting 6* corresponds to an orifice setting of six turns with the pressure relief valve cracking pressure increased to 20 MPa. The measured stopping time for this test was predicted well using the solution of equations 1 and 2. Thus, the decrease of braking pressure during t_s corresponds to a relative large value of n in equation 3.

Equations 1 and 2 can also be used to predict the time required to stop the blade on a commercial mower with I = 4.3 kg m^2 and a rotational speed of 730 rpm. If we again assume a pressure relief valve cracking pressure of 20 MPa and a hydraulic motor with displacement volume of 110 cm^3 rev^{-1}, the predicted stopping time is 1.05 s.

This result, in view of the maximum measured braking pressure versus orifice opening shown in figure 5, suggests that a braking circuit may perform the function of stopping a mower blade in ≤ 1.5 s if an adjustable pressure relief valve was used as a decelerating valve without a sequencing valve and orifice. This possibility will be addressed in future testing of the braking circuit on a commercial rotary mower.

Summary and Conclusions

An experimental circuit to stop a rotating blade driven by a hydraulic motor within 1.5 s was fabricated and tested. Preliminary tests utilizing a small blade and motor demonstrated that the circuit performed as designed. Subsequent tests were conducted with a larger motor and blade that developed 41% of the kinetic energy calculated for a commercial mower motor/blade configuration. Results confirmed that the experimental circuit met operational requirements, stopping the blade in 0.33 to 0.77 s, depending on orifice and pressure relief valve cracking pressure settings.

Using a method of calculating blade stopping time that was verified by experimental results, we can predict that the experimental circuit will meet the design requirement of stopping a commercial mower blade in 1.5 s without exceeding allowable limits of fluid pressure (typically, 17 MPa on a tractor fluid drive system). Finally, the experimental results indicated that the sequencing valve and adjustable orifice used in our experimental braking circuit may not be necessary to stop a rotary mower blade in the specified time of 1.5 s.

Additional experiments will be conducted with a commercial rotary mower modified to be driven by fluid power. The experimental braking circuit will be tested with and without the sequencing valve and adjustable orifice shown in figures 1 and 2. One or more infrared sensors will be mounted near the tractor seat as an OPSS.

References

Bornzin, J. H. 1973. Toward a safer PTO. *Agricultural Engineering* 54(7): 22-24.

Browning, S. R., H. Truszczynska, D. Reed, and R. H. McKnight. 1998. Agricultural injuries among older Kentucky farmers: The farm family health and hazard surveillance study. *Am J Ind Med* 33: 341-353.

Buchele, W. F. 1993. Agricultural safety, design first, then educate! ASAE Paper No. 93-8001. St. Joseph, Mich.: ASAE.

Consumer Reports. 1994. *1995 Buyers Guide: Lawn Mowers* 59(13): 227-229.

Gehlhausen, J. 1995. Hell on wheels: Liability for tractor-pulled mower injuries. *Trial* 31: 64-71.

Lewis, M. Q., N. L. Sprince, L. F. Burmeister, P. S. Whitten, J. C. Torner, and C. Zwerling. 1998. Work-related injuries among Iowa farm operators: An analysis of the Iowa Farm Family Health and Hazard Surveillance Project. *Am J Ind Med* 33: 510-517.

Lin, T., and D. R. Buckmaster. 1994. Evaluation of an optimized engine-fluid power drive system in place of a mechanical PTO. ASAE Paper No. 94-1529. St. Joseph, Mich.: ASAE.

Morgan, G. H. 1992. Eliminate agricultural tractor PTO driveline accidents. ASAE Paper No. 92-1604. St. Joseph, Mich.: ASAE.

Murphy, D. J., and T. Morrow. 1996. Sensing human presence. *Resource* 3(11): 7-8.

Pawlak, R. R. 1990. Product liability litigation: A burden to the farm industry. *Agricultural Engineering* 71: 16-18.

Piercy, L. R., and L. Stallones. 1984. Fatal accidents on Kentucky farms. ASAE Paper No. 84-5508. St. Joseph, Mich.: ASAE.

Sell, W. E., W. P. Campbell, and W. E. Field. 1985. Summary of on-farm PTO related accidents. ASAE Paper No. 85-5513. St. Joseph, Mich.: ASAE.

Shearer, S. A., F. Dong, P. T. Jones, and L. D. Swetnam. 1993. Fluid-power drive for round balers. *Applied Engineering in Agriculture* 9(1): 21-27.

Stepanek, P. J., A. P. Wilcox, and S. A. Shearer. 1995. Proposed standard for hydraulic coupler capacity on agricultural tractors. *Applied Engineering in Agriculture* 11(1): 41-50.

Stewart, H. L., and J. M. Storer. 1968. *Fluid Power*. Indianapolis, Ind.: Howard W. Sams & Co., Inc.

Wilkins, F. 1981. Product liability design defects in farm implements–A practical view from the plaintiff's side. *J Missouri Bar* 37: 253-258.