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Delayed Operation Characteristics of Power Shuttle According to Hydraulic Oil Temperature in the Hydraulic Circuit of Agricultural Tractor

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Abstract

Purpose: During the start-up period, the response time of a hydraulic system increases in the winter because of the increased oil viscosity caused by the cold weather. The problems of delayed tractor starting and excessive wear of the clutch disk occur for these reasons. Therefore, this study develops an analysis model using the commercial hydraulic analysis program AMESim to examine the characteristics of delays in power shuttle starting at different oil temperatures. **Methods:** In the experiment, a tractor was stationary on a flat surface with the engine running at a constant speed of 1,080 rpm. The forward lever was then pressed to activate the power shuttle at three different oil temperatures, and the pressure changes were measured. The pressure on the forward clutch control valve was measured by a pressure gauge installed on the hydraulic line supplied to the transmission from the main valve. An analysis model was also developed and verified with actual tests. **Results:** The trend of the simulated pressures of the power shuttle is similar to that of the measured pressures, and a constant modulation period was observed in both the simulation and test results. However, the difference found between the simulation and test results was the initial pressure required to overcome the initial force of the clutch spring. **Conclusions:** This study also examines the characteristics of the delayed startup of the power shuttle at different oil temperatures through simulations.

Keywords: Agricultural Tractor, AMESim, Hydraulic clutch, Hydraulic oil temperature, Power shuttle

Introduction

The power shuttle that is commonly used in agricultural tractors is called the forward–reverse power shift transmission because of the forward and reverse functions used in the transmission. The power shuttle can control the forward and reverse speeds of the tractor by using only a lever without a clutch pedal (Lee, 2009). This can be done by activating the clutch, which transfers the power of the engine to the transmission, with the wet multidisc friction plate based on the hydraulic changes in the modulation control valve.

The power shuttle, which demonstrates not only excellent work performance, as does the hydrostatic transmission (Kim et al., 2002), but also high manual transmission efficiency, is necessary because the forward–reverse power shift is used more frequently than the drive transmission and is often used with heavy machinery, such as loaders. In addition, the power shuttle can implement various dynamic characteristics by combining modulation control valves with controllers that do not have a foot pedal to operate the clutch. All of these advantages contribute to the increased use of power shuttles in agricultural tractors.

The main problems with the tractor transmission are shift shock, the malfunction of hydraulic devices at high temperatures, and the overheating of the forward–reverse clutch (Lee et al., 2005). However, the forward–reverse power shift is the most urgent problem because agricultural works frequently apply the forward–reverse power shift, which puts a larger load on the clutch; therefore, research should be conducted on the forward–reverse power shift

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(Lee et al., 2002).

Lee et al. (2003) developed and verified a forward–reverse power shift transmission and a computer simulation model of a hydraulic control system. They also examined the influences of the design parameters on the transmission quality using the simulation model. Kim et al. (2002) developed a system analysis technique for a power shift transmission using Easy5 software and examined the influences of hydraulic control variables and design parameters on the transmission quality and dynamic characteristics of the tractor.

The power shuttle is the core technology of the agricultural tractor using the hydraulic control system, and the performance of the tractor power shift is determined by the hydraulic controller. Therefore, optimal design parameters and verification are important for the control of the malfunction of the hydraulic control system (Kim, 2009).

To improve the performance of the automatic transmission, the hydraulic clutch within the automatic transmission should be smoothly connected at the correct time and released quickly. The pressure supplied to the clutch cylinder should be controlled to ensure the smooth connection of the hydraulic clutch, and the residual hydraulic pressure should be removed quickly when releasing the clutch (Kim et al., 1994). However, oil viscosity increases in the winter because of the cold weather, resulting in response delays in the hydraulic system during initial operation. These phenomena result in a delayed start and excessive wear to the clutch disk. In addition, inconvenience and dangerous situations in the slopes occur when operating tractors without an initial warm-up. This is because the hydraulic circuit should provide the proper hydraulic pressure to the hydraulic clutch at an appropriate time (Nam, 2002); however, cold temperatures slow the generation of the hydraulic pressure.

This study develops an analysis model using the commercial hydraulic analysis program AMESim and verifies the simulation model through verification tests. This study also examines the characteristics of the delayed startup of the power shuttle at different oil temperatures through simulations.

Materials and Methods

Test model

The tractor used in this study is shown in Figure 1, and its specifications are given in Table 1.

Figure 1. View of the tractor.

Table 1. Specifications of the tractor

Item		Specification
Engine rated power / Rated speed		55 kW / 2,400 rpm
F/R operation		Power shuttle
T/M Gear	Forward	24
	Reverse	24
Maximum speed		30 km/h

Hydraulic power shuttle circuit

As shown in Figure 2, the hydraulic power shuttle circuit includes a hydraulic pump, a relief valve, a forward clutch control valve, a forward–reverse clutch, and an accumulator.

The pump was connected to the engine, and it ran when the engine was started. Therefore, the high pressure predetermined by the relief valve was maintained from the pump to the forward clutch control valve. When a current was applied to the forward clutch control valve with a state of pressure, b solenoid was magnetized and combined with the forward clutch.

Comprehensive power shuttle model

Figure 3 shows the simulation model of the circuit shown in Figure 2 developed using the commercial hydraulic analysis program AMESim. The main parameters were based on the data provided by the tractor manufacturer and the actual values measured during a bench test.

In this study, the gear pump was modeled assuming that a constant oil flow was provided because the engine rotated with a constant speed. An analysis model was developed assuming that there were no characteristic changes to the forward clutch control valve when the oil

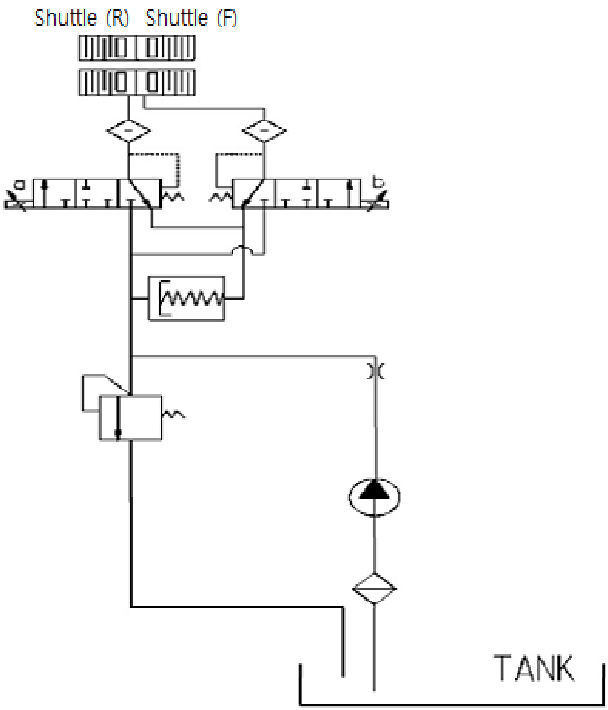


Figure 2. Hydraulic power shuttle circuit.

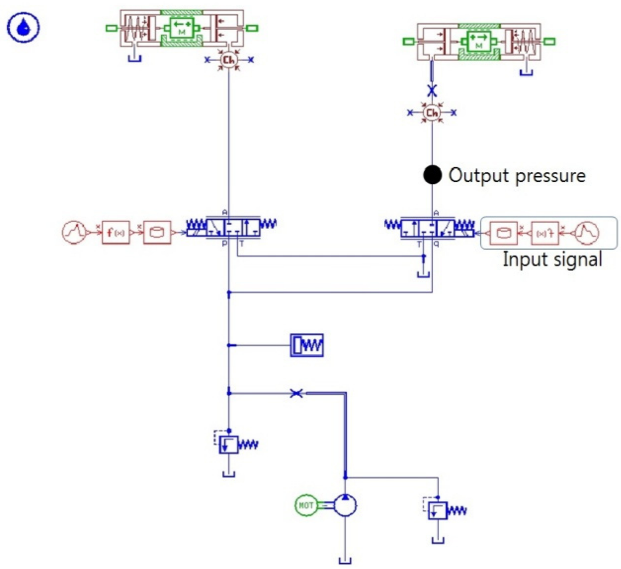


Figure 3. AMESim analysis model of power shuttle.

temperature was varied. This study considered only the hydraulic aspects of the hydraulic clutch and not the mechanical ones; therefore, the clutch was considered to be applied only under hydraulic loads.

Tables 2–6 show the specifications of the power shuttle with the hydraulic circuit. The cracking pressure of the relief valve is 1.96 MPa.

Table 2. Properties of hydraulic oil

Item	Unit	Temperature (°C)		
		25	40	60
Kinematic viscosity	cSt	103.8	53.5	26.3
Absolute viscosity	cP	88.2	45.5	22.3

Table 3. Specifications of hydraulic pump

Item	Specifications
Displacement	7.2 cc/rev
Rotational velocity	500–3000 rpm
Typical speed	2500 rpm
Shaft speed	1040 rpm
Direction of rotation (Viewed from shaft end)	CCW
Relief pressure setting	15.69 MPa

Table 4. Specifications of forward clutch control valve

Item	Specifications
Company / Country	Daesung NACHI / Korea
Type	3-way, 3-position
Flow rate range	7–21 L/min
Current range	0–1 A
Pressure control range	0–1.6 MPa
Carrier frequency	127 Hz
Dead band	0–0.2 A

Table 5. Specifications of hydraulic clutch

Item	Specifications	
Disk type	Wet multi-plate clutch, Waffle type	
Number of disks	Forward	6
	Reverse	6
Disk material	JFP-207C	
Disk gap	0.25 mm	
Flow rate contact area of piston	20%	
Movement distance of piston	8 mm	
Strength at clutch coupling	9,334 N	
Friction torque	546 N·m	
Driving torque	64 N·m	

Table 6. Specifications of accumulator

Item	Specifications
Model / Company / Country	TT-PS-AC / Daesung NACHI / Korea
Piston diameter	28 mm
Piston stroke	16.5 mm
Spring rating	8.2 N/mm
Spring load when fully discharged	329.6 N

Verification test

In this study, the tractor was stationary on a flat surface with the engine running at a constant speed of 1,080 rpm. The forward lever was then pressed with three different oil temperatures (25, 40, and 60°C) to activate the power shuttle, and the pressure changes were measured. These three temperatures were in the normal preheating temperature range and were selected to verify the simulation. A hydraulic supply unit, which controls the oil temperature, was used in the system to maintain a consistent oil temperature. The system sent the oil, which was supplied to the main valve from the steering valve, to the hydraulic supply unit and then resent oil of the same amount and under the same pressure to the main valve from the hydraulic supply unit. This verification testing device was a circular system in that the oil supplied to the transmission from the forward clutch control valve retuned to the hydraulic supply unit.

The pressure on the forward clutch control valve was measured by a pressure gauge (PA 21 SR/KELLER) installed on the hydraulic line supplied to the transmission from the main valve.

Results and Discussion

Simulation analysis

Figure 4 (a) shows the simulation results of the power shuttle analysis model. The pressure was observed to increase at a constant rate for all three oil temperatures at the modulation section, and the tractor starting was delayed by approximately 0.58 s at an oil temperature of 25°C when the pressure for starting the tractor was 0.2 MPa. At temperatures of 40 and 60°C, the delays were 0.56 and 0.54 s, respectively. From these results, it can be stated that the starting delay increased with decreasing oil temperature.

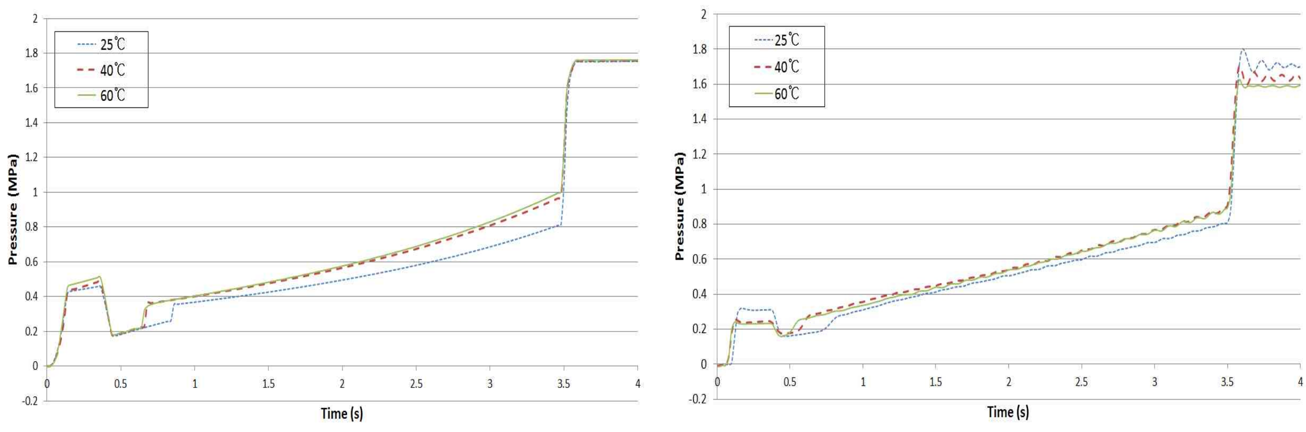
The tractor did not slip on the ramp when the pressure reached the final modulation pressure value. The final pressures were 0.81, 0.97, and 1.00 MPa at temperatures of 25, 40, and 60°C, respectively. When the oil temperature decreased, the density and viscosity of the oil increased; this resulted in the reduced pressure (Lee, 2009).

Test analysis

The oil temperature changed from 25 to 60°C when the power shuttle transmission was tested at room temperature. Five measurements at each temperature investigated in this study (25, 40, and 60°C) were conducted to observe the pressure changes when the forward lever was operated.

Figure 4(b) shows the changes in the pressure of the forward clutch at different temperatures when the forward–reverse lever was switched to forward, maintaining the tractor in an idle state. The forward clutch pressure, which is needed to overcome the initial force of the return spring, decreased with increasing temperature (0.3, 0.24, and 0.23 MPa at 25, 40, and 0 60°C, respectively).

The pressure was observed to increase at a constant rate for all three temperatures in the modulation section, and it can be concluded that the starting of the tractor was delayed based on the fact that the modulation section moved to the right as the temperature decreased. The delay times at 25, 40, and 60°C were 0.74, 0.58, and 0.52 s, respectively. This is attributable to the longer response



(a) (b)

Figure 4. (a) Simulated and (b) measured pressures of power shuttle at different oil temperatures (25, 40, and 60°C).

time of the hydraulic system delaying the starting as the temperature decreased. The pressures increased suddenly to 1.71, 1.65, and 1.59 MPa at 25, 40, and 60°C, respectively, around the end of the modulation section, and these results show that the pressure needed for clutch engagement increases with decreasing temperature.

Comparison of simulation and test results

The trend of the simulated pressures of the power shuttle is similar to the measured pressures, and a constant modulation period was observed in both the simulation and test results. However, the difference between the simulation and test results is the initial pressure required to overcome the initial force of the clutch spring. At 25°C, the simulation results showed 0.44 MPa, whereas the test results showed 0.3 MPa. At 40°C, the simulation and test results showed 0.45 and 0.23 MPa, respectively, and at 60°C, the simulation and test results showed 0.48 and 0.23 MPa, respectively. The differences were 31.8%, 48%, and 52% at 25, 40, and 60°C, respectively, and the different flow rates supplied to the clutch caused these results.

When the pressure for starting the tractor was 0.2 MPa, the delay times varied. At 25°C, the simulation results showed 0.58 s, and the test results showed 0.74 s. At 40°C, the simulation result showed 0.56 s, and the test results showed 0.58 s. At 60°C, the simulation result showed 0.54 s, and the test results showed 0.52 s.

Influence of oil temperature

Figure 5 shows the differences between the delay times at each temperature in the simulation using AMESim.

The starting pressure did not occur at a temperature of −30°C, because the flow was not formed as a result of the low viscosity. Additionally, the starting was delayed by 1.54 s at −20°C. At low temperatures, starting the tractor and reaching the final modulation pressure showed increased delay times.

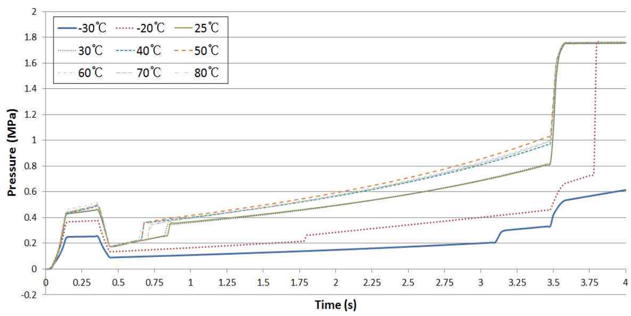


Figure 5. Simulated pressure of power shuttle at different oil temperatures (−30, −20, 25, 30, 40, 50, 60, 70, and 80°C).

Table 7. Simulated pressure of power shuttle at temperatures of −30, −20, 25, 30, 40, 50, 60, 70, and 80°C.

Time (s)	Temperature (°C)								
	−30	−20	25	30	40	50	60	70	80
0	0	0	0	0	0	0	0	0	0
0.25	0.25	0.37	0.43	0.44	0.45	0.46	0.48	0.46	0.46
0.5	0.09	0.13	0.18	0.18	0.18	0.18	0.19	0.18	0.18
0.75	0.09	0.15	0.23	0.23	0.36	0.37	0.36	0.34	0.34
1	0.10	0.16	0.36	0.36	0.4	0.41	0.4	0.39	0.39
1.25	0.11	0.18	0.39	0.38	0.43	0.45	0.44	0.43	0.43
1.5	0.12	0.19	0.42	0.41	0.47	0.49	0.48	0.47	0.47
1.75	0.13	0.21	0.45	0.45	0.51	0.54	0.52	0.51	0.51
2	0.14	0.28	0.49	0.49	0.56	0.59	0.57	0.57	0.56
2.25	0.16	0.31	0.53	0.53	0.61	0.64	0.63	0.62	0.62
2.5	0.17	0.34	0.58	0.58	0.67	0.7	0.69	0.68	0.68
2.75	0.18	0.37	0.63	0.63	0.74	0.78	0.75	0.75	0.74
3	0.19	0.40	0.68	0.69	0.80	0.85	0.83	0.82	0.81
3.25	0.31	0.43	0.75	0.75	0.89	0.94	0.91	0.90	0.90
3.5	0.41	0.51	1.05	1.06	1.24	1.31	1.25	1.24	1.24
3.75	0.56	0.72	1.75	1.75	1.75	1.75	1.76	1.76	1.76
4	0.61	1.75	1.75	1.75	1.75	1.75	1.76	1.76	1.76

According to the results of Cho et al. (2003), the characteristics of the pressure at temperatures above 70°C had similar tendencies; however, the results of this study demonstrate that pressure characteristics and starting delay times were not significantly different above 60°C. Problems observed at high temperatures were the result of the decreasing pressure (Table 7).

Conclusions

This study developed a model for power shuttle analysis using AMESim. In addition, simulation and test results were compared to verify the developed analysis model. Using the developed model, the characteristics of the delay time when starting the power shuttle at various oil temperatures were analyzed. The results of this study are as follows:

(1) From the results of the simulation analysis, different starting delay times were observed at three different oil temperatures: 0.58, 0.56, and 0.54 s at 25, 40, and 60°C, respectively. The difference was minor, but the delay increased with decreasing oil temperature.

(2) From the results of the test analysis, the tractor showed different delay times at each temperature: 0.74, 0.58, and 0.52 s at 25, 40, and 60°C, respectively. This is attributable to the longer response time of the hydraulic system causing the starting to be increasingly delayed with decreasing temperature.

(3) The trend of the simulated pressures of the power shuttle is similar to the measured pressures, and a constant modulation period was observed in both the simulation and test results. However, the difference found when comparing the simulation and test results was the initial pressure required to overcome the initial force of the clutch spring. The differences were 31.8%, 48%, and 52% at 25, 40, and 60°C, respectively, and the different flow rates supplied to the clutch caused these results.

(4) The oil temperature influenced the start delay. The pressure did not form at –30°C, and starting was delayed by 1.54 and 0.58 s at –20 and 70°C, respectively. Similar delay times were observed at 60 and 70°C, which indicates that there is no significant difference above 60°C.

Thus, further studies on developing a modulation with optimum starting capabilities at low oil temperatures are needed.

Conflict of Interest

The authors have no conflicting financial or other interests.

Acknowledgments

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