

## Determination of PID Coefficients for the Ascending and Descending System Using Proportional Valve of a Rice Transplanter

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### Abstract

**Purpose:** This study was conducted to develop a linear Proportional-Integral-Derivative (PID) control algorithm for the ascending and descending system of a rice transplanter and to analyze its response characteristics. **Methods:** A hydraulic model using a single-acting actuator, proportional valve and a PID control algorithm were developed for the ascending and descending system. The PID coefficients are tuned using the Ziegler-Nichols (Z-N) method and the characteristics of unit step response are analyzed to select the PID coefficients at various pump speeds. **Results:** Results showed that the performance of the PID controller was superior in any condition. It was found that the highest settling time and maximum overshoot were less than 0.210 s and 5%, respectively at all pump speed. It was determined that the steady state errors were 0% in all the cases. The lowest overshoot and settling time were calculated to be nearly 2.56% and 0.205 s, respectively at the pump rated speed (2650 rpm). **Conclusions:** The results indicated that the developed PID control algorithm would be feasible for the ascending and descending system of a rice transplanter. Finally, it would be helpful to plant the seedlings uniformly and improve the performance of the rice transplanter.

**Keywords:** Hydraulic actuator, PID controller, Proportional valve, Rice transplanter, Simulation

### Introduction

A rice transplanter is an indispensable machine for rice farming. In Korea, rice farming is completely mechanized and the mechanization of rice transplanter is approximately 99.9% (Yu et al., 2015). At present, several problems such as low sustainability, brittleness and so on have been identified in the existing transplanter, and demand for mechanized rice transplanters in domestic and overseas markets is decreasing because of the technological gap with developed countries such as

Japan, USA and so on (Siddique et al., 2018). The most critical issue is that only the ascending descending system of existing rice transplanter is mechanized which is associated with the tilting of the planted seedlings. Given that an on/off valve, which does not able to perform precisely, was used to develop the existing model. It is necessary to use highly precise technology, such as proportional valve to replace the on/off valve, to enhance the comprehensive performance of rice transplanter.

Several articles have been published on transplanter applying hydraulic actuators and proportional valve. Hydraulic actuators are widely used in autonomous industry for their tolerance, density, and sizing, and hydraulic actuators can effectively control velocity using

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a proportional valve (Backe, 1993; Burrows, 1994). In addition, hydraulic actuators have various benefits such as low production cost, quick response, and a simple control algorithm (Song and Lee, 2015; Shin et al., 2015; Shin et al., 2013). Hydraulic proportional valves improved the performance and efficiency of the hydraulic servo or actuating system. Moreover, proportional valves are capable of efficiently operating the actuator during the field operation and provide optimal power control depending on workload. Zhu et al. (2012) reported that the hydraulic control system of a high-speed transplanter improved the stability and efficiency of transplanter. The simulation results displayed the real-time state graphs which revealed that the performance of a transplanter using hydraulic control system was stable, flexible, and comfortable in the field. However, the planting device needed to be operated manually. Wei et al. (2015) also applied a proportional valve to the manual control system of rice transplanter for autonomic navigation only. Lian et al. (2009) modified the transplanting control system, transmission, navigation, and variable operation of the Kubota rice transplanter with an electronic control system using a proportional valve instead of the manual control system. The results showed that the electronic control system of transplanter performed similar to a manual operating system.

In existing literature, it is found that the ascending and descending system using proportional valve of a rice transplanter has not been studied in detail. Some researchers have stated the advantages and reasons of the application of the hydraulic proportional valves and actuators in various agricultural machineries. The proportional valves improved the flexibility, increased the comfort of drivers and also saved energy (Zardin et al., 2018; Ahn et al. 2015; Borghi et al., 2014). The function of proportional valves is to supply a constant or required flow rate, and make manual or automatic adjustments utilizing the sensors, controllers and actuators (Lisowski and Filo, 2016; Bravo et al., 2017). This study also indicated that hydraulic companies and scientific research organizations are highly interested in proportional valves because they enable a smooth adjustable flow rate capacity. Therefore, the proportional valve is applied in the ascending and descending system of rice transplanter to control the flow rate as well as actuator displacement.

However, proportional valves as well as hydraulic

actuators have some unwanted characteristics such as nonlinearity, hysteresis, and asymmetric grain flow. The behavior of nonlinearity can change the system load and operating conditions (Zhang, 1999). According to Anderson (1988), the nonlinearity can impact the performance of the hydraulic system, accuracy of the control, and the stability of the hydraulic system. Therefore, a control algorithm is important to control those nonlinear properties of proportional valves or hydraulic actuating system.

There are several types of controllers to control the servo system such as fuzzy logic controller (Milecki and Rybarczyk, 2015), nonlinear control system using Lypunov function (Sohl and Bobrow, 1999), nonlinear robust control system (Guan and Pan, 2008), and PID controller (Rozali et al., 2010; Kim et al., 2016). Among them, proportional, integral, and derivative (PID) controller is the most common, and it is frequently used to control the hydraulic system in autonomous industries (Rozali et al., 2010). A PID controller is designed using a Laplace transformer (Shang, 2004). Simplicity, flexibility, and adaptability are the main advantages of the PID control algorithm (Jiangxue, 2017). In addition, the tuning coefficients of PID controllers are comparatively easier to determine using the Ziegler-Nichols methods. Therefore, the objectives of this study are i) to develop a PID controller for the ascending and descending system designed using the hydraulic proportional valve of a rice transplanter, ii) to analyze the response characteristics of the PID controller, and iii) to determine and optimize the PID coefficients.

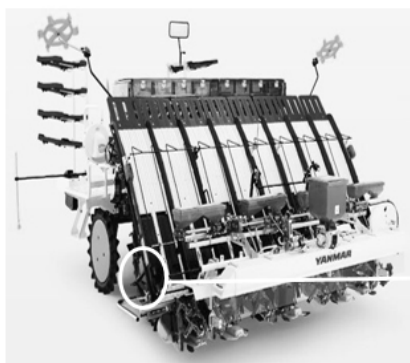
## Materials and Methods

### Rice transplanter

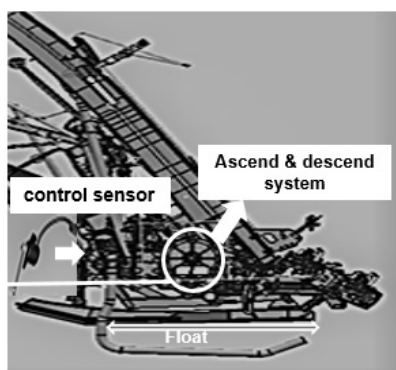
Generally, rice transplanter has five important parts that are hydraulically interrelated with each other, namely,

**Table 1.** Specifications of rice transplanter

Item	Specifications
Length × Width × Height (mm)	3300 × 2095 × 1620
Weight (kg)	770
Engine Rated power (kW)	15.6 @3200
Fuel capacity (L)	37
Total displacement (L)	0.903
Planting distance (cm)	30



(a) Rice transplanter (Yanmar YR6D)



(b) Ascending and descending system

**Figure 1.** Ascending and descending system of rice transplanter.

the engine, hydrostatic transmission, power assisted steering, horizontal control system, and ascending and descending system. The specifications of a rice transplanter is listed in Table 1.

In this article, the ascending and descending system of a rice transplanter has been studied. The ascending and descending system has a float sensor, which controls the planting depth maintaining the height between soil and float. Transplanter performance depends on the actuator movement of the ascending and descending system. The

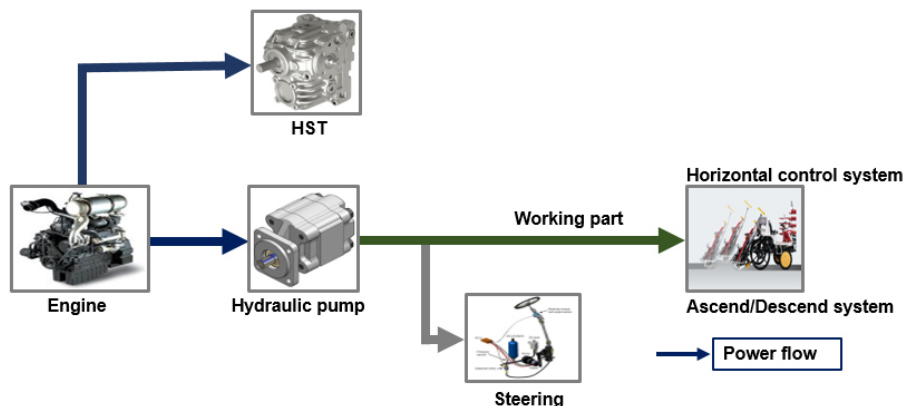
actuator displacement has two effects on the planted seedlings: the transplanter float pushes, and presses the soil. As a result, the planted seedlings tilt and finally, effect on the planting uniformly as well as transplanter performance. Therefore, it is a vital issue to control the actuator displacement of the ascending and descending system. The ascending and descending system of rice transplanter is shown in Figure 1.

### Power flow of rice transplanter

Engine power is divided into the hydrostatic transmission (HST) and hydraulic pump. Then power from the hydraulic pump is transmitted to the horizontal control system, and ascending and descending system through the power assisted steering mechanism. The principle of the power-assisted steering mechanism is that the steering force is geared up through the hydraulic circuit and helps turn the machine easily. Actually, the rotational speed and torque generated by the engine act on the hydraulic pump, and the flow rate generated by the pump supplies power to the hydraulic motor, resulting in driving the working institutions. The horizontal control system maintains a steady machine balance, and supplies plants to the planting institution continuously. Consequently, the ascending and descending system is associated with planting at the desired depth. A schematic diagram of the rice transplanter power flow is shown in Figure 2.

### Block diagram of the ascending and descending system

The ascending and descending system of the rice transplanter is developed based on the power flow of rice transplanter hydraulic system simulated by LMS AMESim (ver.: 16, Imagine S. A. company, France). The ascending



**Figure 2.** Schematic diagram of rice transplanter power flow.

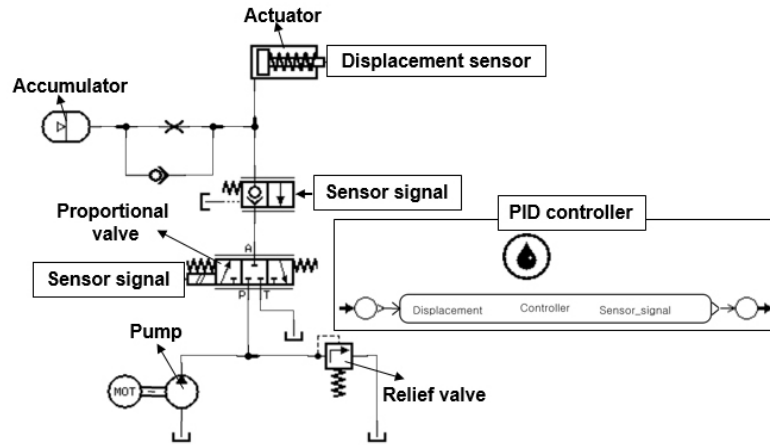


Figure 3. Block diagram of the ascending descending system of rice transplanter.

and descending of rice transplanter is shown in Figure 3.

The hydraulic proportional valve controls the flow rate supplied to the actuator, and the hydraulic actuator performs the ascending and descending by extending and retracting. The accumulator reserves the excessive fluid flow and supplies to the actuator in an emergency. The parameters of the hydraulic components were considered with respect to the actual rice transplanter. Specifications of the ascending and descending system of rice transplanter are listed in Table 2 (Siddique et al., 2018; Yoshida and Miyakawa, 2013; Amirante et al., 2013; Song and Lee, 2007; Zhang, 1999; Esposito, 1980).

External load, nonlinear factors, asymmetric gain, and friction greatly affect the displacement tracking of the

ascending and descending system of the rice transplanter.

## Design of controller for ascending and descending system

### Actuator model

In this study, a single acting hydraulic cylinder is used. Its velocity can be determined by the flow rate supplied to the actuator. A proportional valve is applied to regulate the flow rate, and the extending motion of the actuator is defined by flow community theory where the leakage and friction are omitted (Zhang, 2009; Zhang, 1999; Esposito, 1980). The flow rate supplied to the actuator is as follows.

$$Q_a = Ay' + \frac{V_1}{\beta} P' \quad (1)$$

where  $Q_a$  is the flow rate to the actuator ( $m^3/s$ ),  $A$  is the cylinder area at the head end ( $m^2$ ),  $P$  is the cylinder pressure at the head end ( $N/m^2$ ),  $V_1$  is the volume of the cylinder ( $m^3$ ),  $y$  is the displacement of the actuator (m),  $y'$  and  $P'$  are the 1<sup>st</sup> derivative form of actuator displacement ( $y$ ) and cylinder pressure, respectively, and  $\beta$  is the fluid bulk modulus ( $N/m^2$ ).

The single acting actuator retracting motion is accomplished by using gravity or spring in the rod end.

$$PA - K_{spring} X_{act} = my'' + F \quad (2)$$

where  $K_{spring}$  is the spring rate ( $N/m$ ),  $X_{act}$  is the length of stroke (m),  $m$  is the mass of the actuator (kg),  $y''$  is the 2<sup>nd</sup> derivative of  $y$ , and  $F$  is the external force acting on the cylinder (N).

The supplied flow rate to the head end of the actuator can be estimated by the orifice equation as described below.

Table 2. Specifications of the ascending and descending system

Item	Specifications	
Actuator	Piston diameter	55 mm
	Rod diameter	15 mm
	Spring preload	600 N
	Spring rate	40,000 N/m
	Pressure in rod chamber	2 bar
Accumulator	Gas precharge pressure	40 bar
	Accumulator volume	0.16 L
Proportional valve	Valve rated current	200 mA
	Valve natural frequency	50 Hz
	Valve damping ratio	1.8
	Valve gain	200
Pump	displacement	4.5 cc/rev
	speed	2650 rpm
Relief valve cracking pressure	242 bar	
Density	850 kg/m <sup>3</sup>	
Bulk modulus	17,000 bar	

$$Q_o = C_d A_o \sqrt{\frac{2}{\rho} (P_s - P)} \quad (3)$$

where  $Q_o$  is the flow rate to the head end of the cylinder ( $m^3/s$ ),  $C_d$  is the orifice coefficient,  $A_o$  is the valve orifice area ( $m^2$ ), and  $P_s$  is the supply pressure of pump ( $N/m^2$ ).

The orifice area can be calculated by the following equation.

$$A_o = C_g u \quad (4)$$

where  $C_g$  is the gain coefficient of the solenoid valve, and  $u$  is the control input voltage signal.

Therefore, the following equation is the flow rate to the head end of the actuator.

$$Q_o = C_f u \sqrt{(P_s - P)} \quad (5)$$

Here,

$$C_f = C_d C_g \sqrt{\frac{2}{\rho}} \quad (6)$$

where  $C_f$  is the flow gain coefficient, and  $\rho$  is the fluid density ( $kg/m^3$ ).

From equation (1) and (5), it is stated that

$$C_f u \sqrt{(P_s - P)} = A v + \frac{v_1}{\beta} P' \quad (7)$$

Here,  $v$  is the velocity of the actuator, and neglecting spring back pressure for extending motion of actuator.

$$P A = m v' + F \quad (8)$$

Here,  $v'$  is the 1<sup>st</sup> derivative of  $v$ , and equation (7) and (8) provide the nonlinear state of the model. To get linearized formation, these equations are needed to make the differential equation using constant coefficient. The differential equations for linear actuator are below.

$$K_{input} \delta u - K_{pressure} \delta P = A \delta v + \gamma \delta P' \quad (9)$$

$$\delta P A_1 = m (\delta v)' \quad (10)$$

The equations (9) and (10) can be applied to design the control algorithm for linear actuator.

Here,

$$K_{input} = \left( \frac{\partial Q_s}{\partial u} \right) u = u_o \quad (11)$$

$$K_{pressure} = \left( \frac{\partial Q_s}{\partial P} \right) P = P_o \quad (12)$$

$$\gamma = \frac{v_1}{\beta} \quad (13)$$

where  $K_{input}$  is the input signal,  $K_{pressure}$  is the gain factor of cylinder chamber pressure, and  $\delta P$ ,  $\delta Q_s$ , and  $\delta v$  are the increment of cylinder pressure, pump flow supply, and actuating velocity, respectively.  $\gamma$  is the dumping factor.

The dynamic model for the hydraulic system is given as below.

$$\frac{\gamma m}{A} \delta P' = K_{input} \delta u - A \delta x' - K_{pressure} \delta P \quad (14)$$

where  $\delta P'$  is the increment of pressure changing rate in the cylinder,  $x'$  is the velocity of the linear actuator, and  $\delta x'$  is the increment of cylinder velocity.

### PID controller

Single acting actuator and proportional valve were the main concern. A controller was designed on the basis of transfer function of feedback control system. This type of controller stabilizes the system using a simple PID algorithm.

$$G(s) = \left( K_p + \frac{K_i}{s} + K_d s \right) \quad (15)$$

where  $K_p$ ,  $K_i$ , and  $K_d$  are the coefficients of proportional, integral, and derivative, respectively. According to Isidori, (1989), the feedback linearization method is applied in equation (14) and the governing equation for single acting actuator is as follows.

$$\frac{V(s)}{u(s)} = \frac{K_{input}}{A} \frac{\omega_n^2}{(s^2 + 2\zeta\omega_n s + \omega_n^2)} \quad (16)$$

where  $\zeta$  is the valve damping ratio,  $\omega_n$  is the valve natural frequency of proportional valve which is denoted by Hz. According to Amirante et al., (2013), the range of natural frequency of a commercial proportional valve is 20 Hz to 85 Hz for the stabilization of the system.  $K_{input}$  is the input signal, and  $A$  is the cylinder area at the head end ( $m^2$ ). The values of the coefficients that are used to

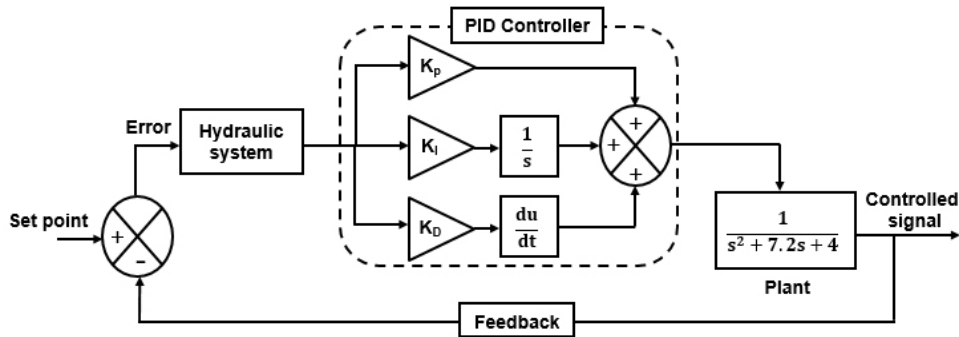


Figure 4. Schematic block diagram of PID controller for ascending and descending system.

determine  $K_{input}$  are listed in Table 2.

Rice transplanters require faster rise time for the ascending and descending system. Yoshida and Miyakawa, (2013) reported that overdamping occurs when damping ratio is higher than 1, and overdamping assists in having a faster rise time. Equation (16) indicates a second-order system, which is used as a plant. It can control the velocity, i.e., flow rate as well as actuator displacement. The schematic block diagram of control algorithm is shown in Figure 4, in which a hydraulic system (Figure 3) developed using AMESim is installed by MATLAB/Simulink cosim interference.

### PID coefficients simulation

The control algorithm for ascending and descending system of a rice transplanter was developed using MATLAB/Simulink interferences (ver.: R2015a, MathWorks, USA). According to Choi et al. (2010), PID coefficients are carefully tuned, and the primary coefficients are selected by the Ziegler-Nichols (Z-N) method.

$$K_p = 1.2 \frac{T}{L} \quad (17)$$

$$K_i = \frac{K_p}{2L} \quad (18)$$

$$K_d = K_p \times 0.5L \quad (19)$$

where  $L$  is the delay time (s), and  $T$  is the time constant (s)

The Z-N method which is an open loop method, proposes procedures to determine the coefficients  $K_p$ ,  $K_i$ , and  $K_d$  using the equation (17) - (19). This method is characterized two constants: one is the delay time ( $L$ ) and another is the time constant ( $T$ ). Those constants are calculated based on the tangent line drawn at inflection

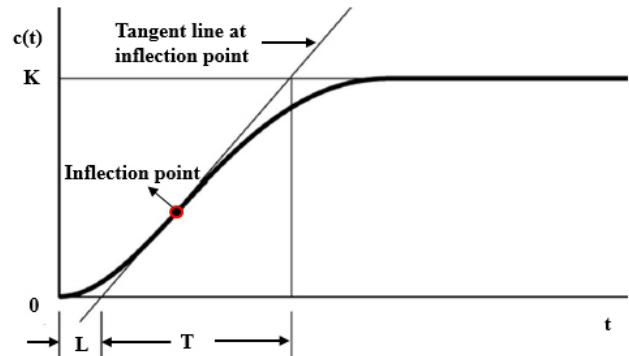


Figure 5. Z-N tuning method (unit step response) for PID controller.

point on the step response curve of the plant. Figure 5 shows the Z-N tuning method for unit step response.

In this study, pump rotational speed was considered as a variable. First, the rated revolution per minute (rpm) was determined using the minimum (1800) and maximum (3500) rpm and the total rpm range was divided into 4 levels of rpm. PID coefficients were determined at the pump rated speed. Finally, the selected coefficients at the pump rated speed was applied to the 4 levels of rpm.

### Determination of optimal PID coefficients

The characteristics of unit step response such as maximum overshoot, settling time, and steady state error of PID controller were considered to select the PID coefficients. Settling time of an output is the required time to reach and stay within the range of the acceptable error band. The acceptable error band range is usually  $\pm 2\%$  of the final value because the settling time is strongly related to the largest time constant of the control system (Ogata, 2002). Steady state error which is the difference between actual response and desired position after being controlled, is indicated at 0%. Percent

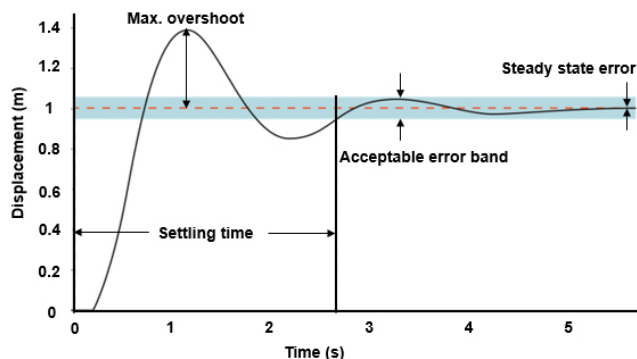


Figure 6. Characteristics of unit step response.

overshoot is determined by the following equation.

$$\text{Overshoot} = \frac{M_p - F_v}{F_v} \times 100\% \quad (20)$$

where  $M_p$  is the peak value of the time response, and  $F_v$  is the final value of the response. For Figure 4, the final value of the response is 1.

Generally, the best or optimum coefficients are chosen on the basis of the overshoot, settling time, and steady state error values. The lower overshoot, settling time, and steady state error indicate the better performance of the actuating system (Kim et al., 2016). The selected coefficient at the rated pump speed was applied for the 4 levels of rpm (1800, 2225, 3075, and 3500) for the stability test and/or evaluating the performance of the controller. Characteristics of unit step response are shown in Figure 6.

## Results and Discussion

### Determination of PID coefficients

The performance of the PID controller has been conducted under various conditions. The actuator displacement was

tracked using the PID controller at 5 determined pump speed levels (1800, 2225, 2650, 3075, and 3500 rpm) for step reference signal. The controller has been used for the changing internal gain of hydraulic system. The controller was also capable of compensating the nonlinearity (temperature) including saturation and friction for satisfactory performance, that verified the linear PID control system. Table 3 represents the tuned PID coefficients including maximum overshoot, settling time, and steady state error that were analyzed for selecting the optimum PID coefficients. It was found that the steady state error was 0% for all conditions. It was also found that the average settling time and maximum overshoot were approximately 0.208 s and 3.766%, respectively. However, the lowest settling time and overshoot at the pump rated speed were calculated as approximately 0.205 s and 2.56%, respectively for  $K_p$ :5.24,  $K_i$ :4.39, and  $K_d$ :1.34. On the other hand, the highest settling time and overshoot were almost 0.210 s and 4.87%, accordingly for  $K_p$ :3.50,  $K_i$ :2.45, and  $K_d$ :1.12.

It is noticed that the displacement tracking of hydraulic actuator raised dramatically and stayed between the acceptable error band within 0.210 s for each coefficient

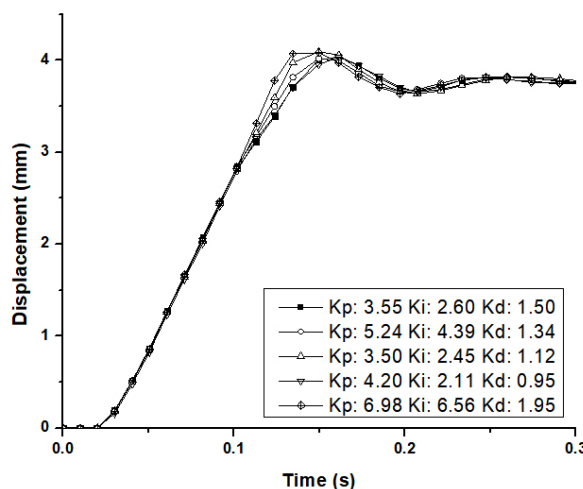


Figure 7. Actuator displacement tracking at the pump rated speed.

Table 3. Determination of PID coefficients at the pump rated speed

RPM	$K_p$	$K_i$	$K_d$	Max. overshoot (%)	Settling time (s)	Steady state error (%)
2650 (Rated)	3.55	2.60	1.50	3.85	0.210	0
	5.24	4.39	1.34	2.56	0.205	0
	3.50	2.45	1.12	4.87	0.210	0
	4.20	2.11	0.95	2.58	0.206	0
	6.98	6.56	1.95	4.97	0.209	0
Average				3.766	0.208	0

at the rated pump speed, shown in Figure 7, and the maximum overshoot was less than 5%.

Kim et al., (2016) stated the lowest values of the settling time, overshoot and steady state error that are referred to as the best performance of the actuating system. These results indicate that the controller can compensate the unexpected inherent parameters of the hydraulic system with accuracy.

### Stability test of the controller

The selected coefficient at the pump rated speed was applied for all level of the pump rotational speeds, listed in Table 4. It is found that the best performance of the PID controller was at the pump rated rotational speed among all levels of rpm. Approximately 0.205 s of the settling time and 2.56% of the max. overshoot were the lowest. However, the highest values of the settling time and overshoot were almost 0.210 s and 4.97%, respectively at 1800 rpm.

The stability test results are also revealed that the max. overshoot and steady state error were less than 5% and 1%, respectively. In addition, settling time was also less

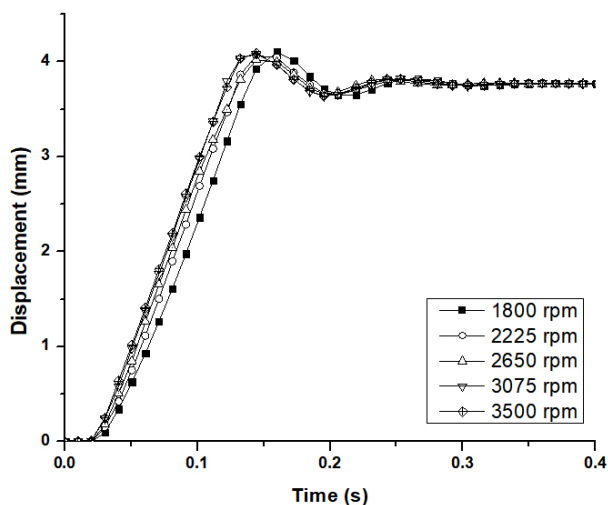
than 0.210 s shown in Figure 8. Li et al., 2017 reported that settling time was 4.5 s, and there was no overshoot and steady state error for hydraulic proportional valve in the position control mode. This study also reported that in the case of traction control mode of proportional valve, settling time was 4.5 s, there was small overshoot and no steady state error. This study showed that the simulation results are realistic for agricultural machines.

Kim et al., (2016) reported that average settling time and max. overshoot were found 0.44 s and 2.51%, respectively for shuttle shifting actuator of Automated Manual Transmission (AMT). The steady state error was also calculated to be approximately 0.29%. Rice transplanter are comparatively slower and more precise machines than tractors. Therefore, this control algorithm is applicable to the ascending and descending system of a rice transplanter.

However, the limitation of this study is that only a simulation model is used for evaluation. In future, it is necessary to conduct an actual field test for the validation of this control algorithm in a real rice transplanter. Finally, it can be seen that the developed PID controller could be feasible for the ascending and descending system of a rice transplanter.

**Table 4.** Results of applying the selected coefficients to each engine revolution speed

RPM	Max. overshoot (%)	Settling time (s)	Steady state error (%)
1800	4.97	0.210	0
2225	3.85	0.207	0
2650	2.56	0.205	0
3075	4.61	0.208	0
3500	4.36	0.209	0



**Figure 8.** Actuator displacement tracking for optimum coefficient.

### Conclusions

This study was conducted to develop a linear PID control algorithm and select the optimum PID coefficients for the ascending and descending system of a rice transplanter to improve its stability, comfort and accuracy. To control the displacement tracking of a hydraulic actuator and proportional valve, a PID controller was applied. A hydraulic simulation model and a control algorithm were developed using the electro hydraulic proportional valve. To study the performance of PID controller, 5 sets of coefficients were determined at the rated pump speed and the optimum coefficients were selected for the stability test. The test results of this study showed that the steady state error was 0% (zero) at all conditions. It is found that the highest settling time and max. overshoot were less than 0.210 s and 5% at all level of pump speeds. This controller can suitably compensate the effects of nonlinear gains of the hydraulic system. The results indicated that the developed PID control algorithm could be useful for the ascending and descending system of a rice transplanter. Finally, this control algorithm could



be helpful in minimizing the tilting of seedling planted using the rice transplanter. Ultimately, it would uniformly improve the transplanter performance.

## Conflict of Interest

The authors have no conflicting financial or other interests.

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## List of Nomenclatures

Symbol	Parameters	Unit
$Q_a$	Flow rate to the actuator	$m^3/s$
$Q_o$	Flow rate to the head end of the cylinder	$m^3/s$
$\delta Q$	Increment of pump flow supply	
$A$	Cylinder area at the head end	$m^2$
$A_o$	Valve orifice area	$m^2$
$P$	Cylinder pressure at the head end	$N/m^2$
$\delta P$	Increment of cylinder pressure	
$\delta P'$	Increment of pressure changing rate in the cylinder	
$V$	Volume of the cylinder	$m^3$
$v$	Velocity of the actuator	$m/s$
$\delta v$	Increment of actuating velocity	
$\beta$	Fluid bulk modulus	$N/m^2$
$K_{spring}$	Spring rate	$N/m$
$X_{act}$	Length of stroke	$m$
$m$	Mass of the actuator	$kg$
$F$	External force acting on the cylinder	$N$
$C_d$	Orifice coefficient	
$C_g$	Gain coefficient of the solenoid	
$C_f$	Flow gain coefficient	
$P_s$	Supply pressure of pump	$N/m^2$
$u$	Control input voltage signal	
$\rho$	Fluid density	$kg/m^3$
$K_{input}$	Input signal	
$K_{pressure}$	Gain factor of cylinder chamber pressure	
$y$	Displacement of the actuator	$m$
$x'$	Velocity of linear actuator	$m/s$
$\delta x'$	Increment of cylinder velocity	
$\gamma$	Dumping factor	
$K_p$	Coefficient of proportional	
$K_i$	Coefficient of integral	
$K_d$	Coefficient of derivative	
$\omega_n$	Valve natural frequency	$Hz$
$\zeta$	Valve damping ratio	
$T$	Time constant	$s$
$L$	Delay time	$s$
$M_P$	Peak value of the time response	
$F_V$	Final value of the response	