Development of a Novel Hydrostatic Continuously Variable Transmission with Fast Path Switching

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Abstract

Recently, human-robot collaboration is required in many kinds of fields. To develop robotic systems for collaboration with humans, a wide driving range of robotic actuators is necessary. For example, robots for rehabilitation have to conduct some tasks, like supporting human and moving fast. Backdrivability is also necessary for robotic actuators from a viewpoint of safety. A hydraulic closed circuit called an electro-hydrostatic actuator (EHA) is utilized for robotic actuators since they have high power/weight ratio and backdrivability. However, EHA cannot achieve characteristics of both high velocity driving and high force driving since a reduction ratio is constant. This paper proposes a novel hydrostatic continuously variable transmission that has two oil paths with a solenoid valve. The solenoid valve switches between two discrete reduction ratios by switching two oil paths. Furthermore, this paper realizes a continuously variable reduction ratio by switching the two oil paths quickly. This paper confirms that the reduction ratio changes continuously and widely by simulations.

Keywords: Continuously variable transmission, hydraulic systems, solenoid valve.

1. Introduction

In recent years, cooperation between robots and humans is required not only in industrial fields, but also human assistance such as rehabilitation. Most conventional industrial robots conduct only specific tasks⁽¹⁾. However, robots for rehabilitation fields have to conduct some tasks, like supporting human and moving fast. To develop robotic systems for collaboration with humans, a wide driving range

of robotic actuators is necessary. Robotic actuators need to drive in high speed and output high force since required driving characteristics are different among each task. Furthermore, robots are required to move flexibly against unexpected disturbances by humans. In other words, high backdrivability is needed to develop a flexible actuator⁽²⁾⁽³⁾.

Hydraulic actuators, which exhibit large outputs, have been utilized widely in the field of construction. Currently, they have been actively researched for robotic actuators⁽⁴⁾. Hydraulic systems have two types, open circuits, and closed circuits. Hydraulic actuators are driven by hydraulic pumps and servo valves in open circuits. A servo valve is an electrically operated valve that controls how hydraulic fluid is sent to a hydraulic actuator. Open circuits can control the desired both flow rate and pressure by using servo valves. In other words, the hydraulic actuators have a wide driving range. However, open-loop hydraulic circuits have no backdrivability since the oil direction is one way and servo valves blocked external forces.

On the other hand, a hydraulic closed circuit called an electro-hydrostatic actuator (EHA) is driven by servo pumps coupling with servo motors⁽⁵⁾. EHAs are not needed to put large oil tanks and have backdrivability since oil direction is bidirectional. However, they cannot achieve both high speed driving and high force driving since a reduction ratio determined by a ratio of a displacement between a pump and a motor is constant.

This paper proposes a novel hydrostatic continuously variable transmission (CVT). The proposed CVT has two oil paths with a solenoid valve. The solenoid valve switches between two discrete reduction ratios by switching the two oil paths. Furthermore, the proposed CVT can widely change the reduction ratio continuously by switching the solenoid valve quickly.

(a) Hydraulic circuit.

(b) Mechanism.

Fig. 1. Proposed hydrostatic CVT.

This paper is organized as follows. In section 2, the proposed CVT is presented. The control method of a servo motor and a solenoid valve are introduced in section 3. Simulations are conducted in section 4. Finally, the conclusions are presented in section 5.

2. Proposed Hydrostatic CVT

2.1 Hydraulic Circuit

The proposed hydrostatic CVT is shown in Figures. 1. The proposed CVT is a hydrostatic system similar to EHAs. A servo motor controls a driving of a hydraulic pump and two hydraulic motors are connected by a rotating shaft. A charge pump is connected to keep the minimum pressure of the oil above atmospheric pressure and avoid cavitation. Cavitation is a phenomenon that the air dissolved in the hydraulic cylinder evaporates and forms bubbles when the pressure of the oil drop below the atmospheric pressure in closed hydraulic circuits⁽⁷⁾. The proposed hydraulic circuit is protected by installed a charge pump. The charge pressure of

Fig. 2. Flow path of 3-way solenoid valve.

Table 1. Switching patterns of solenoid valve.

Paths	Connect point	Shuttle valve
Path 1	Tank	A to C
Path 2	Charge Pump	B to C

the hydraulic circuit is lower than the relief pressure by setting a pressure reducing valve. The relationship among the tank pressure p_t , which was assumed to be zero, the charge pressure p_c , and the relief pressure p_r are described as:

$$p_t < p_c < p_r. \quad (1)$$

A solenoid valve and two shuttle valves are used to switch two oil paths. Shuttle valves determine a direction by pressure difference. Figure. 2 shows a flow path of a 3-way solenoid valve. The spool moves and opens each path when the voltage applies to the solenoid valve. The spool position x_{sv} is given as

$$x_{sv} = k_{v0}V, \quad (2)$$

where V and k_{v0} represent the input voltage and the spool position gain. Then, the output oil flow q_b and pressure p_b are given as

$$q_b = k_{sv}x_{sv} \left(s(x_{sv}) \sqrt{\frac{2|p_r - p_b|}{\rho}} + s(-x_{sv}) \sqrt{\frac{2|p_b - p_t|}{\rho}} \right), \quad (3)$$

$$S(x_v) = \begin{cases} 1, & \text{if } x_{sv} \leq 0 \\ 0, & \text{if } x_{sv} > 0 \end{cases}, \quad (4)$$

$$sC_b p_b = q_b - K_l(p_b - p_c), \quad (5)$$

where q , p , k_{sv} , ρ , C_b , and K_l represent flow rate, pressure, a flow coefficient, oil density, a compression coefficient, and a leakage coefficient respectively. By applying (3)-(5), the pressure p_b changes by switching the solenoid valve. Switching pattern of the solenoid valve is listed in Table 1. In the path 1 connected to the tank, the port B pressure becomes lower than the charge pressure. The port A and the port C are connected and the port B is blocked. On the other hand, the pressure of the port B becomes higher than the

Fig. 3. Path 1.

Fig. 4. Path 2.

one of the port C in the path 2 since the charge pressure is lower than the relief pressure. The port A and the port B are connected and the port C is blocked. The model of the shuttle valve can be described as

$$p_a = \max(p_b, p_c). \quad (6)$$

2.2 Modelling

Figure. 3 shows the path 1 in the proposed CVT. The red line is the flow path. The motor 1 and the motor 2 rotate by the input pump. The path 2 is shown in figure. 4. The blue line is the flow path. In the path 2, the only motor 2 rotates by input pump. Oil flow displaced by the motor 1 flows the loop.

The displacement of the hydraulic motor in the path 2 is different from the one of the path 1. Therefore, the reduction ratio can be switched between two discrete value.

The dynamics of the motor side is given as

$$J_m \ddot{\theta}_m + B_m \dot{\theta}_m = \tau_m^{\text{ref}} - \frac{D_m}{2\pi} \Delta p - \tau_m^{\text{dis}}, \quad (7)$$

where J , B , D , θ , τ , and Δp represent inertia, viscous coefficient, displacement, angle, torque, and pressure

Fig. 5. Two-inertia resonant system.

difference between the input and the output ports respectively. The subscript m stands for an index for the motor side. The superscript ref and dis stand for an index for the reference value and the disturbance value respectively.

The dynamics of the load side is given as,

$$J_l \ddot{\theta}_l + B_l \dot{\theta}_l = \frac{D_l}{2\pi} \Delta p - \tau_l^{\text{dis}}, \quad (8)$$

$$D_l = (D_{l1} + D_{l2}, D_{l2}), \quad (9)$$

where D_{l1} and D_{l2} represent the displacement of the motor 1 and the motor 2 respectively. The subscript l represents an index for the load side. The pressure difference Δp is expressed by the oil compression as follows:

$$\begin{aligned} s\Delta p &= \frac{1}{c_b}(q_m - q_l - K_l \Delta p), \\ &= \frac{1}{c_b} \left(\frac{D_m}{2\pi} \dot{\theta}_m - \frac{D_l}{2\pi} \dot{\theta}_l - k_l \Delta p \right), \\ &= \frac{D_m}{2\pi c_b} \left(\dot{\theta}_m - \frac{D_l}{D_m} \dot{\theta}_l - \frac{2\pi}{D_m} \Delta p \right). \end{aligned} \quad (10)$$

Finally, the dynamic of the torsional torque τ_s is given as

$$\tau_s = K_s \left(\theta_m - R\theta_l - \frac{1}{B_s} \int \tau_s \right), \quad (11)$$

$$K_s = \frac{1}{c_b} \left(\frac{D_m}{2\pi} \right)^2, \quad (12)$$

$$B_s = \frac{1}{K_l} \left(\frac{D_m}{2\pi} \right)^2, \quad (13)$$

$$R = \left(\frac{D_{l1} + D_{l2}}{D_m}, \frac{D_{l2}}{D_m} \right), \quad (14)$$

where K_s , B_s , and R represent a spring coefficient of oil, a viscous coefficient, and reduction ratios respectively. Modeling and parameters except the reduction ratios are the same between the path 1 and the path 2. By applying (7)-(14), the proposed CVT is modeled as two-inertia resonant systems as shown in Figure. 5.

3. Control

3.1 Position Control

The motor-side angle is controlled by the servo motor. The block diagram of the position control is shown in Figure. 6. K_p and K_v represent feedback gains of position and velocity for a position controller. The superscripts cmd stands for

Table 2. Parameters of simulations.

Parameters	Values	Parameters	Values
J_m	0.00063 kg · m ²	k_{vo}	0.075 mm/V
J_l	0.5 kg · m ²	k_{sv}	1.07 E-5
B_m	0.0003 Nms/rad	V_0	10.0 V
B_l	5.0 Nms/rad	g_{dob}	150 rad/s
B_s	0.0915 Nms/rad	K_p	6400
K_s	0.099 Nm/rad	K_v	160
R_1	100	K_f	100
R_2	20	K_r	0.01
ρ	900 kg/m ⁻³		

when the relative angular velocity is a positive value. It means that the time opening the path 1 in a switching period needs to decrease and the time opening the path 2 in a switching period needs to increase. On the other hand, the average reduction ratio of the proposed CVT has to be increased when the relative angular velocity is a negative value. It means that the time opening the path 1 in a switching period needs to increase and the time opening the path 2 in a switching period needs to decrease. When the positive voltage value is higher than the negative one in the PWM control, the time opening the path 1 becomes long and the time opening the path 2 becomes short. Therefore, the input voltage is controlled as follows :

$$V = s(t)V_0 + \frac{K_{iv}}{s}W_v, \quad (24)$$

where K_{iv} represent the integral gain. To track the desired reduction ratio in a short time, the voltage gains set higher as the desired reduction ratio becomes large.

In the torsional torque control, the mechanism to control the desired reduction ratio is the same. The relative torque W_f is given as

$$W_f = R_v\tau_s - \tau_1^{dis}. \quad (25)$$

W_f should be zero when the average reduction ratio is same with the desired one. The average reduction ratio of the proposed CVT has to be increased when the relative angular torque is a positive value. average reduction ratio of the proposed CVT has to be decreased when the relative angular velocity is a negative value. Then, the input voltage is controlled as follows:

$$V = s(t)V_0 - \frac{K_{if}}{s}W_f, \quad (26)$$

where K_{if} represents the integral gain. To track the desired reduction ratio in a short time, the voltage gains set higher as the desired reduction ratio is small.

4. Simulation

4.1 Condition

Simulations were conducted to confirm the continuously variable reduction ratio of the proposed CVT. Parameters for simulations are listed in Table 2. By assuming the switching period of the solenoid valves was 60 ms, the duty ratio was set as 0.5. The variable reduction ratio was set as 40, 60, and 80. A servo motor was actuated after time 1.0 in the simulations.

4.2 Results

Figures. 9 show responses of the motor-side angle. θ_m^{cmd} was set as,

$$\theta_m^{cmd} = 1.0t, \quad (27)$$

where t represents time. As shown in Figure. 9 (a), the response of the load-side angular velocity changed by switching the two oil paths quickly. Figures. 9 (b)-(d) show the enlarged view of Figure. 9 (a) for 40 ms when R_v was set as 40, 60, and 80. As shown in Figures. 9 (b)-(d), the angular velocity responses were oscillating as the two oil paths were switched.

The load-side disturbance torque responses in the torsional torque control are shown in Figures. 10. The command value of the torsional torque was set as 1 Nm with a ramp-shaped trajectory in this simulation. As shown in Figure. 10 (a), the response of the load- side disturbance torque changed by switching the two oil paths quickly. Figures. 10 (b)-(d) show the enlarged view of Figure. 10 (a) for 40 ms when R_v was set as 40, 60, and 80. As shown in Figures. 10 (b)- (d), the torsional torque responses were oscillating as the two oil paths were switched.

The comparisons of the reduction ratios between the simulation value and the theoretical one are listed in Table 3. The simulation value is calculated by the load-side angular velocity and the torsional torque from 4 seconds to 5 seconds. From Table 3, the simulated reduction ratios corresponded with the theoretical one. As result of the simulations, the reduction ratio could change continuously in a wide range by switching the solenoid valve quickly and changing the input voltage of the solenoid valve.

(a) Load-side angular velocity

(a) Load-side disturbance torque.

(b) $R_v=40.0$.

(b) $R_v=40.0$.

(c) $R_v=60.0$.

(c) $R_v=60.0$.

(d) $R_v=80.0$.

(d) $R_v=80.0$.

Fig. 9. Responses of load-side angular velocity.

Fig. 10. Responses of load-side disturbance.

Table 3. Comparisons of reduction ratios.

Theoretical value	Figures. 9	Figures. 10
40.0	40.0	39.9
60.0	60.4	59.9
80.0	80.0	79.8

5. Conclusions

In this paper, the novel hydrostatic CVT was proposed. The solenoid valve of the proposed CVT switched between the two discrete reduction ratios. Furthermore, this paper realized that the reduction ratio changed to the desired value by switching the two oil paths quickly and controlling the input voltage of the solenoid valves. This proposed method was confirmed by the simulations. The wide range of variable reduction ratios is achieved by the proposed hydrostatic CVT.

Actuators with CVT can switch between the high speed driving and the high force driving. In other words, a driving range of actuators can expand. Moreover, the proposed mechanism had higher backdrivability than the conventional hydraulic actuators of open circuits.

Future work includes to validate the continuously variable reduction ratio in experiments with comparing the result of the simulation. Hydrostatic systems included nonlinearities. Control scheme compensating these effects had to be developed to achieve the desired reduction ratio.

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