International Journal of Advances in Engineering and Management (IJAEM)
Volume 5, Issue 5 May 2023, pp: 996-999 www.ijaem.net ISSN: 2395-5252

Improve Power train Quality with Predictive PID Controller

Le Thi Minh Nguyet (1) Le Thi Thu Ha (2)
(1) Faculty of Electrical Engineering, College of economics and techniques Thai Nguyen, Vietnam
(2) Faculty of Electrical Engineering, Thai Nguyen University of Technology, Vietnam

Date of Submission: 15-05-2023 Date of Acceptance: 30-05-2023

ABSTRACT
The actuator includes the motor and the mechanical devices that convert its direction of motion, collectively referred to as the transmission system, which is the object used in most production and manufacturing technology lines. Furthermore, most of the powertrain controllers available today are PID-based (97%). Therefore, just a small improvement in PID quality improvement to control the drive system can bring about a very high efficiency in terms of economy and product quality in the entire manufacturing industry, improving the efficiency of the system. product competitiveness.
The article proposes the direction of using predictive control technique to improve the quality of the PID controller for the problem of sustainably sticking to the set values of the drive systems.
Keywords: PID tuning; Predictive control; Drivetrain; Gap; Friction

I. INTRODUCTION

The actuator includes the motor and the mechanical devices that convert its direction of motion, collectively referred to as the transmission system, which is the object used in most production and manufacturing technology lines. Furthermore, most of the powertrain controllers available today are PID-based (97%). Therefore, just a small improvement in PID quality improvement to control the drive system can bring about a very high efficiency in terms of economy and product quality in the entire manufacturing industry, improving the efficiency of the system. product competitiveness.

Therefore, we have set the goal here is to use some new results in the theory of optimal control and predictive control into the PID controller to control the set-value tracking with high quality of the parameters transmission system when there are additional mandatory constraints on the supplied energy, on the working domain of the system such as: working temperature of the device, energy conversion rate..., signal constraints output: torque, speed…

In [4,5,6,7,8] we built a mathematical model of the gap transmission system taking into account the influence of frictional components, gaps and elasticity also propose a number of control methods that improve the quality of the powertrain with gaps such as: PID controller, blurred controller, state adaptive adaptive controller, bound forecast controller.

Specifically in [2,3,5] pointed out the disadvantages of the gap drive system when using a PID controller such as the dynamic quality of the system is still very poor, there is always oscillation, with the set amount as a sinusoidal function, the system still has the difference in amplitude and phase angle, the output response has not been able to follow the input set amount (Figure 1). In [5, 6] show the vibration of the gap drive system when using a sustainable adaptive controller. To overcome the above disadvantages, the author proposes a method to use predictive PID fuzzy controller. This controller is suitable for strong nonlinear objects and has the ability to self-reset the parameters of the PID controller, thereby improving the quality of the drive system.

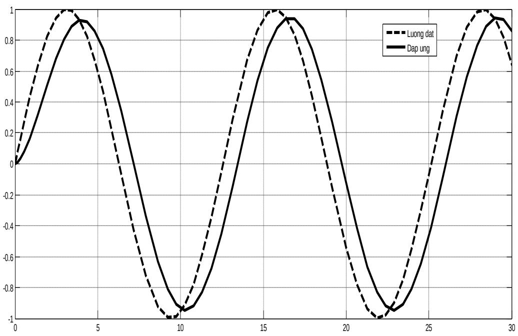

Fig 1. Simulation results of the response of the drive system to the PID controller when the input quantity is a sinusoidal function

Fig 2. Experimental results of the system's working characteristics on WinCC Flexilbe when the set speed changes from 0 v/min to 1000 v/min with PID controller

II. PROBLEM STATEMENT

According to [5, 6], we have a general model of the gear transmission system:

$$\begin{cases} \bar{J}_1\ddot{\varphi}_1 + \hat{c}r_{L1}^2\cos^2\alpha_L(\varphi_1 + i_{12}\varphi_2) = M_d - M_{ms1} \\ J_2\ddot{\varphi}_2 - \hat{c}r_{L2}^2\cos^2\alpha_L(\varphi_2 + i_{21}\varphi_1) = -M_c - M_{ms2} \end{cases} \quad (1)$$

In there: $\bar{J}_1 = J_d + J_1$

r_{L1}, r_{L2} radius of rolling circle of gear 1 and 2.

α_L The angle of engagement of two gears, is a measure of the clearance between the gears. In the case of two standard gears and no central displacement, the engagement angle $\alpha_L = \alpha = 20^0$. For a system with a gap $18^0 \le \alpha_L \le 25^0$.

c is a measure of the stiffness of the gear. The smaller the c value, the greater the gear flexibility.

M_d depends on the motor type selected, for example when selecting DC motor with parallel excitation: $M_d = M_0 - b_0\dot{\varphi}_1 = M_0 - b_0\omega_1$

M_c depending on the form of the load: for example: $M_c = M_c(\dot{\varphi}_2, \dot{\varphi}_1, t)$

M_{ms1}, M_{ms2} is the frictional moment component in the shaft bearings, assuming the frictional moments depend only on speed: $M_{ms1} = \gamma_1\dot{\varphi}_1$ và $M_{ms2} = \gamma_2\dot{\varphi}_2$

Just as the system in operation has close gears, approximated by $\hat{c} \approx c$, $\forall t$ incorporating the input hysteresis $u(t-\tau) = M_d$ whose τ is small enough to compensate for this approximation, with additional symbols: $cr_{L1}^2\cos^2\alpha_L = c_{z1}$, $cr_{L2}^2\cos^2\alpha_L = c_{z2}$

The general model of the powertrain is rewritten as:

$$\begin{cases} \bar{J}_1\ddot{\varphi}_1 + \gamma_1\dot{\varphi}_1 + c_{z1}(\varphi_1 + i_{12}\varphi_2) = u(t-\tau) \\ J_2\ddot{\varphi}_2 - \gamma_2\dot{\varphi}_2(\varphi_2 + i_{21}\varphi_1) + M_c = 0 \end{cases} \quad (2)$$

Besides, from the second equation and when the system is a slow variable we have: $\ddot{\varphi}_2 = 0$

I will be with $M_c = 0$:

$$\begin{cases} c_{z2}i_{21}\varphi_1 = -c_{z2}\varphi_2 + \gamma_2\dot{\varphi}_2 \\ c_{z2}i_{21}\dot{\varphi}_1 = -c_{z2}\dot{\varphi}_2 \end{cases}$$

$$\Leftrightarrow c_{z2}i_{21}\begin{pmatrix}\varphi_1 \\ \dot{\varphi}_1\end{pmatrix} = \begin{pmatrix}-c_{z2} & \gamma_2 \\ 0 & -c_{z2}\end{pmatrix}\begin{pmatrix}\varphi_2 \\ \dot{\varphi}_2\end{pmatrix}$$

$$\Leftrightarrow \begin{pmatrix}\varphi_2 \\ \dot{\varphi}_2\end{pmatrix} = c_{z2}i_{21}\begin{pmatrix}-c_{z2} & \gamma_2 \\ 0 & -c_{z2}\end{pmatrix}^{-1}\begin{pmatrix}\varphi_1 \\ \dot{\varphi}_1\end{pmatrix}$$

$$= \frac{c_{z2}i_{21}}{c_{z2}^2}\begin{pmatrix}-c_{z2} & -\gamma_2 \\ 0 & -c_{z2}\end{pmatrix}\begin{pmatrix}\varphi_1 \\ \dot{\varphi}_1\end{pmatrix} \quad (3)$$

So: $\varphi_2 = \beta_1\varphi_1 + \beta_2\dot{\varphi}_1$ (4)

In there: $\beta_1 = -i_{21}, \beta_2 = -\frac{c_{z2}i_{21}\gamma_2}{c_{z2}^2}$ (5)

are constant parameters.

Substituting (4) into the first equation of (2) we get:

$$u(t-\tau) = \bar{J}_1\ddot{\varphi}_1 + \gamma_1\dot{\varphi}_1 + c_{z1}\left[\varphi_1 + i_{12}(\beta_1\varphi_1 + \beta_2\dot{\varphi}_1)\right]$$
$$= \bar{J}_1\ddot{\varphi}_1 + (\gamma_1 + c_{z1}i_{12}\beta_2)\dot{\varphi}_1 + c_{z1}(1 + i_{12}\beta_1)\varphi_1$$

Using the notation $y = \varphi_2$ for the first signal to be output to the continuous transfer function describes the approximate linear model of the gear drive system as follows:

$$G(s) = \frac{Y(s)}{\Phi_1(s)}.\frac{\Phi_1(s)}{U(s)} = \frac{\beta_1 + \beta_2 s}{\alpha_1 s^2 + \alpha_2 s + \alpha_3}e^{-\tau s} \quad (6)$$

Where $Y(s)$, $\Phi(s)$, $U(s)$ is the laplace image of y, φ_1, u :

$\alpha_1 = \bar{J}_1$

$\alpha_2 = \gamma_1 + c_{z1}i_{12}\beta_2$ và $\alpha_3 = c_{z1}(1 + i_{12}\beta_1)$

Converting the continuous linear model (6) to the discontinuous linear model with the sampling period $T_a = \tau$ and the input signal having the form of a constant fragment, we get the equivalent delay discontinuous transfer function as follows:

$$G(z) = Z\{G(s)\} = \frac{b_1 z^{-1} + b_2 z^{-2}}{1 + a_1 z^{-1} + a_2 z^{-2}} \quad (7)$$

Where the constant parameters b_1, b_2, a_1, a_2 are respectively inferred from $\alpha_1, \alpha_2, \alpha_3, \beta_1, \beta_2$.

Consider a gear transmission system with the following parameters:

Transmission ratio: $i_{12} = 2$

Base radius of two gears: $r_{01} = 50$mm; $r_{02} = 100$mm

Mutation angle between two gears: $\alpha_L = 30^0$

The moment of inertia of the two gears is: $J_1 = 0,01kgm^2$; $J_2 = 0,02kgm^2$

Elasticity coefficient: $c = 10N$

Ignoring the input delay, the gear drive system with the transmission function is described as follows:

$$G(s) = \frac{0.07s + 0.5}{1.5s^2 + 1.75s + 0.375}$$

We move to the operator domain Z, we get the transfer function: $G(z)=\dfrac{0.1z+1}{z+0.9}$

III. SIMULATION AND EXPERIMENTAL RESULTS

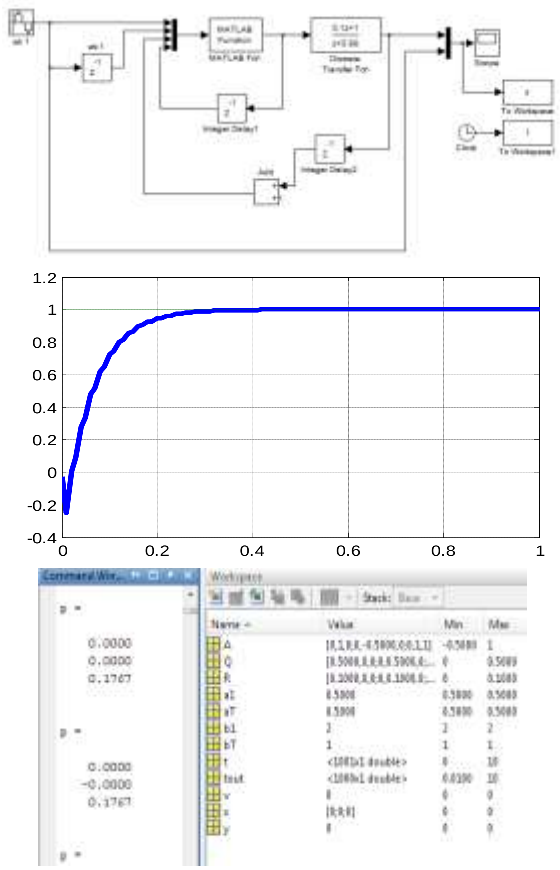

Fig 3: Simulation diagram of transmission system and transient function with jump excitation

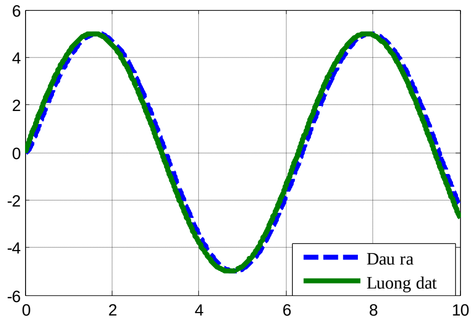

Fig 4: Simulation results of transient function with sinusoidal excitation

In the absence of interference, the response of the system to the step quantity is quite good. From the graph of the transient function, we can see that the transient time is small, there is no excess of the oscillation and the output follows the required amount.

When we replace the set signal as $\omega_k=\sin(10^{-2}k)$ shown in a solid line, from the transient function of the system we see that the output signal is a dashed line that has followed the input signal and there is almost no phase difference between the output signal and the input signal.

To test the system's noise resistance, we add a block to generate white noise signals back to the input. From the transient function of the system, for the set signal as the step function, the system shows a rather poor noise resistance, the output signal cannot follow the set signal, and the system is unstable.

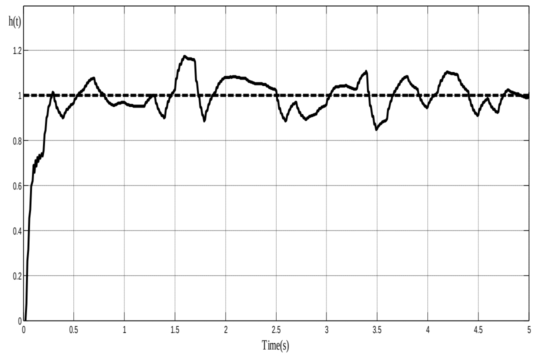

Fig 5: Simulation results of transient function with jump excitation in the presence of noise

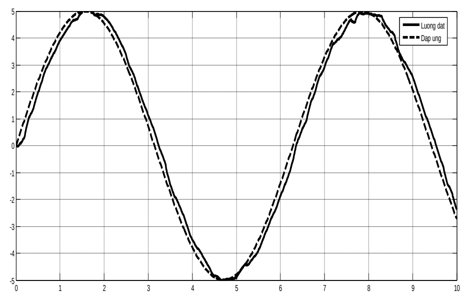

Fig 6: Simulation results of transient function with sinusoidal excitation in the presence of noise

With the set signal being sinusoidal $\omega_k=\sin(10^{-2}k)$, from the graph of the transient function of the system, we can see that the noise resistance of the system is quite good. The output signal can still follow the set signal and the deviation is very small.

Fig 7: Experimental model

Fig 8: System working characteristics on WinCC Flexilbe when set speed 1400 rpm (with change of dummy load) with predictive PID controller

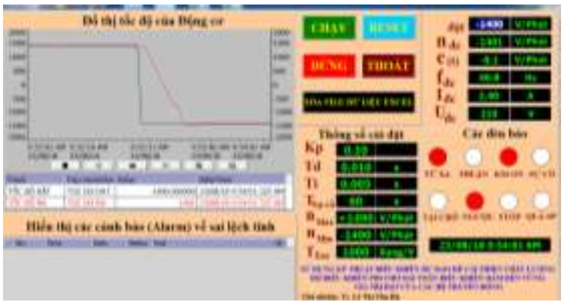

Fig 9: Working characteristics of the system on WinCC Flexible when the set speed changes from 1400 rpm to -1400 rpm with predictive PID controller

IV. CONCLUSION

The article has studied and built a predictive PID controller. This PID controller has also been tested in the article on gear transmission system that takes into account the gap, elasticity and friction factors with the input response of the step function and the sinusoidal harmonic function. The simulation results on MatLab and the experimental results show that the predictive PID controller has the ability to self-adjust the parameters to suit the object so that it can accurately control the object. From the promising results, this will be a widely adopted control method to achieve better quality and precise control.

REFERENCES

[1] Mallesham,G. and Rajani,A. Automatic Tuning of PID Controller using Fuzzy Logic. Proceedings of 8th Int. Conference on Development and Application Systems, Rumani 5.2006, pp.120-127.

[2] Le Thi Thu Ha, Lai Khac Lai and Le Thi Minh Nguyet, Quality examination of gearing transmission system, Journal of science and technology Thai nguyen university, Book 051, 2009, pp. 124-130.

[3] Lai Khac Lai, Le Thi Thu Ha, Le Thi Minh Nguyet and Nong Le Huy (2009), One adaptive control method for gear system, Journal of science and technology Thai nguyen University, Book 88, 2011, pp. 163-167.

[4] Le Thi Thu Ha and Nguyen Doan Phuoc (2015), Design a predictive PID controller with infinite horizon foradaptive control of gearing systems, Proceeding of VCCA-2015, pp. 7-12.

[5] Lê Thị Thu Hà (2013), Some solutions to improve performances of gearing transmission systems with backlash created on robust and adaptive control, Doctor Dissertation. Thai Nguyen University of Technology.

[6] Le Thi Thu Ha (2014), Modelling of gearing transmission system, Journal of science and technology Thai nguyen University, Book 118, No. 4, 2014, pp. 67-78.

[7] Dang Tuan Nam (2017), Master Thesis, Thai Nguyen University of Technology.

[8] Le Thi Thu Ha (2018), University level topic, Thai Nguyen University of Technology.