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Motion Control of AUV using IMC-PID Controller

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ABSTRACT

This paper highlights on IMC based PID controller for controlling the pitch of the Autonomous Underwater Vehicle (AUV). A detail study of the kinematics and dynamic of an AUV has been described with the help of 6 DOF equations using body and earth fixed frame. To analyze the system better the mathematical model of AUV is reduced to a FOPDT. The simulated shows the efficiency of IMC-PID controller as compared to the conventional PID controller designed and tuned using CHR method.

Key words : AUV, CHR method, Model reduction, PID tuning.

1. INTRODUCTION

Autonomous underwater vehicles (AUVs) as the name suggests are automatic submersible robots that performs different functions without the intervention of human beings. It can also be regarded as a wireless robot. Today AUV has been considered as the most challenging and difficult in the field of research. The AUV technology has found its application in many areas like military, fishery, resource survey, etc. For the motion control of AUV depth changing movement is important. The inherent property of non-linearity in AUV makes it difficult to apply linear control. The complexity in the dynamic characteristics of AUV is the result of its properties like higher nonlinearity, variation in time, uncertainities in hydrodynamic coefficients and the external disturbances. Various controllers have been proposed for modelling the AUVs. This includes linear controllers [8]-[11], which give satisfactory performance; 'Sliding Mode controllers' [12], [13], 'adaptive control' [12]-[14], 'FLC (Fuzzy Logic Control)' [15], 'predictive control' [16], 'static feedback control' [17], have also shown good robustness and tuning ability. Due to the uncertainities in the parameters and coefficients of AUVs, non-linearity of underwater environment due to ocean current disturbances, hydrodynamics drag forces the control of AUV has became a challenging task.

The most difficult challenge for designing a controller for AUVs lies in the mathematical modelling of the vehicle itself. The difficulties in designing basically lies in the fact that non-linearity in vehicle dynamics as well as it is difficulty in finding all hydrodynamic parameters affecting the vehicle dynamics with reasonable accuracy [2]. Motion control of AUV become more difficult due to its complex structure as the motion in all of three axes is in coordinate of fixed frame since each rotation around an axis will result in hydrodynamic forces and rotational torques. A lot of attempts has been made to develop the AUV controller using both conventional as well as modern methods. However due to the complexities the conventional controller could not achieve satisfying control objectives. While the modern methods of control including intelligent control is able to give better performance, since they can effectively match with uncertainty of the hydrodynamic [19].

Keeping these challenges in mind an internal model controller based on PID has been designed and the results of the simulation are compared with a conventional PID controller [21]. Due to the simple design of PID controller and easy to implement it is still used widely.

The paper is arranged as follows. In Section 2 the modeling of an AUV is discussed describing its kinematics and dynamics . Controller design is developed in the Section 3. Section 4 present the simulation graphs and provide a comparative study between the PID and the IMC-PID controller. Finally the conclusion is drawn at the end.

2. AUV MODELLING

In order to find the mathematical model of the vehicle, it is required to study both the kinematics and dynamics of the vehicle. Kinematics describes the geometrical aspects of motion whereas Dynamics defines the analysis of forces which cause the motion. [1]

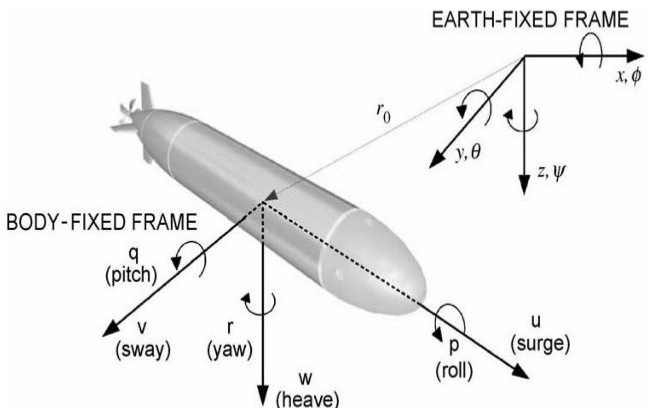

Figure 1: Model of AUV

Table 1: Notation used [1]-[3]

DOF	Motion	Forces and moments	Velocity	Position and Euler angles
1	surge	X	u	x
2	sway	Y	υ	y
3	Heave	Z	w	z
4	roll	K	P	ϕ
5	Pitch	M	q	θ
6	Yaw	N	r	ψ

To determine the position and orientation of the vehicle six degrees of freedom (6-DOF) differential equations of motion are required [4]. The first three are used to represent the position and translation motion along x, y, and z axes, while the last three coordinates are used to describe the orientation and rotational motion.

2.1 Vehicle kinematics

A two co-ordinate frames is chosen to analyze the motion of the vehicle in six degree of freedom. The moving reference frame is known as body-fixed reference frame because it is fixed to the vehicle. The motion of the body-fixed frame is described with respect to an inertial frame. Incase of marine vehicles, the acceleration wrt a point on the Earth's surface can be neglected and the Earth fixed frame is considered to be an inertial frame. This implies that the linear and angular velocities of the vehicle has to be defined in terms of body-fixed frame while position and orientation should be described with respect to inertial frame [1]. In a very general form, the motion of vehicle in 6DOF can be described by the following vectors: [1]

$\eta = \begin{bmatrix} \begin{bmatrix} \eta_1 \\ \eta_2 \end{bmatrix} \end{bmatrix}$ $\eta_1 = \begin{bmatrix} x \\ y \\ z \end{bmatrix}$ Position vector $\eta_2 = \begin{bmatrix} \phi \\ \theta \\ \psi \end{bmatrix}$ Euler angles vector

$v = \begin{bmatrix} v_1 \\ v_2 \end{bmatrix}$ $v_1 = \begin{bmatrix} u \\ v \\ w \end{bmatrix}$ linear velocity vector $v_2 = \begin{bmatrix} p \\ q \\ r \end{bmatrix}$ Angular velocity vector

$\tau = \begin{bmatrix} \tau_1 \\ \tau_2 \end{bmatrix}$ $\tau_1 = \begin{bmatrix} X \\ Y \\ Z \end{bmatrix}$ forces vector $\tau_2 = \begin{bmatrix} K \\ M \\ N \end{bmatrix}$ moments vector.

The mapping between the two coordinate frames is given by the Euler angle transformation $\dot{\eta} = J(\eta_2)v$, where $J(\eta_2)$ is the Jacobian matrix [18].

2.2 Vehicle Dynamics

Similar to kinematics of the vehicle, its dynamics is also divided into translational and rotational motion. The transltional equation of motion is given as below:

$$m(v_0 + \omega * v_0 + \dot{\omega} * \dot{r_g} + \omega * (\omega * r_g)) = f_0 \quad (1)$$

and the rotational equation of motion is as follows:

$$I_0\dot{\omega} + \omega * (I_0\omega) + mr_g * (v_0 \dot{+} \omega * v_0) = m_0 \quad (2)$$

Where m is the mass of the body (vehicle) and I_0 is the moment of inertia.

Using 'Newton's and Euler's equation' the six degree of freedom equation can be written as: [2,18]

$$m\left(\dot{u} - vr + wq - x_g\,(q^2 + r^2) + y_g\,(pq - \dot{r}) + z_g(pr + \dot{q})\right) = X \quad (3)$$

$$m\left(\dot{v} - wp + ur - y_g\,(r^2 + p^2) + z_g\,(qr - \dot{p}) + x_g(qp + \dot{r})\right) = Y \quad (4)$$

$$m\left(\dot{w} - uq + vp - z_g\,(p^2 + q^2) + x_g\,(rp - \dot{q}) + y_g(rq + \dot{p})\right) = Z \quad (5)$$

$$\begin{aligned} &I_x\dot{p} + (I_z - I_y)qr - (\dot{r} + pq)I_{XZ} + (r^2 - q^2)I_{yz} + (pr - \dot{q})I_{xy} \\ &\quad + m[y_g\,(\dot{w} - uq + vq) - z_g(\dot{v} - wp + ur)] = K \end{aligned} \quad (6)$$

$$I_y\dot{q} + (I_x - I_z)rp - (\dot{p} + qr)I_{Xy} + (p^2 - r^2)I_{XZ} + (qp - \dot{r})I_{yz} + m[z_g\,(\dot{u} - vr + wq) - x_g(\dot{w} - uq + vp)] = M \quad (7)$$

$$I_z\dot{r} + (I_y - x)pq - (q + rp)I_{yz} + (q - p^2)I_{xy} + (rq - \dot{p})I_{xz} + m[x_g\,(\dot{v} - wp + ur) - y_g(\dot{u} - vr + wq)] = N \quad (8)$$

Assuming the heave velocity is being very small and is neglected the state space equation of the system will be

$$\begin{bmatrix} I_y - M_q & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix} \begin{bmatrix} \dot{q} \\ \dot{z} \\ \dot{\theta} \end{bmatrix} + \begin{bmatrix} -M_q & 0 & M_\theta\theta \\ 0 & 0 & u_1 \\ -1 & 0 & 0 \end{bmatrix} \begin{bmatrix} q \\ z \\ \theta \end{bmatrix} = \begin{bmatrix} M_{fs} \\ 0 \\ 0 \end{bmatrix} \quad (9)$$

From the above matrix representation, the transfer function for the inner pitch loop is found as [3]

$$G_\theta(s) = \frac{\theta(s)}{f_s(s)} = \frac{\frac{M_{fs}}{I_y - M_q}}{s^2 - \frac{M_{fs}}{I_y - M_q}s - \frac{M_\theta}{I_y - M_q}} \quad (10)$$

Substituting the data given for REMUS AUV as given in [4],[18], we get

$$G_\theta(s) = \frac{-3.18}{s^2 + 1.09s + 0.52} \quad (11)$$

3. CONTROLLER DESIGN

The relatively complex AUV can be broken into separate layer to simplify the controller design [20]. The controlling scheme of an AUV is divided into three

1 heading control

2. dive plane control

3. speed control

3.1 PID CONTROLLER

A proportional–integral–derivative controller (PID controller) is a control loop feedback mechanism (controller) commonly used in industrial control systems. A PID controller continuously calculates an error value as the difference between a desired set point and a measured process variable. If the parameters of a conventional PID controller are tuned properly then it can give a good performance [5]. The PID controller is expressed as follows:

$$C(s) = K_p + \frac{K_i}{s} + K_d s \quad (12)$$

Where K_p = proportional gain , K_i = integral gain and K_d= derivative gain.

The PID is tuned for its gain value using C-H-R technique will be as shown in Table 2. In this technique, the PID controller tuned with respect to set-point and disturbance rejection [4]. Closed-loop response which is more heavily damped, guarantees for an ideal plant and the one which is having high response speed without overshoot is considered as overshoot of 0% [6].

3.2 IMC-PID DESIGN

In process control application IMC design has become famous [7]. In this G(s) is FOPDT, in IMC it is suitable for open-loop stable control systems. The Internal model control as shown in consists of a stable internal model controller parameter Q(s) and ^ G s( ) is the model of the plant. F(s) is internal model controller filter selected to make Q(s) F(s) proper by improving the robustness.

Table 2: CHR Tuning Method

Controller type	Overshoot of 0%		
	K_p	τ_i	τ_d
P	0.3/a		
PI	0.35/a	1.2T	
PID	0.6/a	T	0.5T

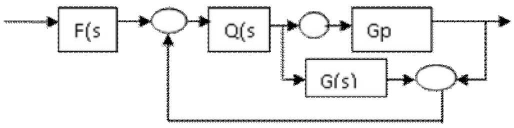

Figure 2: Block Diagram of IMC

$$C(s) = \frac{F(s)Q(s)}{1 - F(s)Q(s)\widehat{G(s)}} \quad (13)$$

4. RESULT AND DISCUSSION

In order to validate the effectiveness and stability of IMC-PID controller designed in this paper, the transfer function of the pitch control of AUV can be approximated as:

$$G_\theta(s) = \frac{6.1154}{1.425s + 1}e^{-0.95s} \quad (14)$$

4.1 CHINE-HRONES-RESWICK (CHR) TUNING

By adjusting the parameters of PID controller using Chine-Hrones-Reswick tuning method, the step response is as shown in Figure 3. The best response of the controller can be obtained with K_p = 0.13 , K_i= 0.09 , K_d = 0.

4.2 IMC-PID TUNING

By adjusting the parameters of PID controller using IMC-PID tuning method, the step response is as shown in Figure 4. The best response of the controller can be obtained with K_p= 0.14 , K_i= 0.08 , K_d= 0.

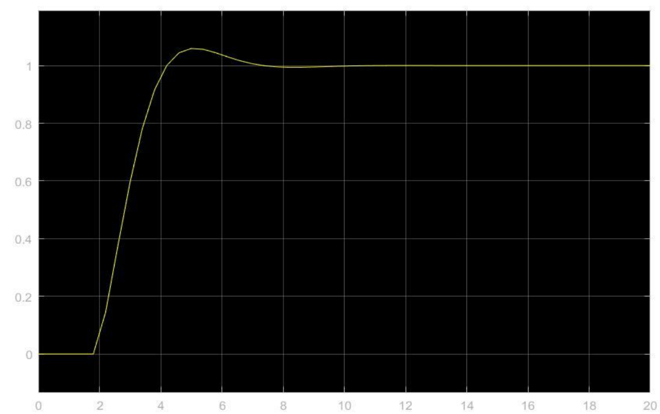

Figure 3: Step Response of Conventional PID Controller

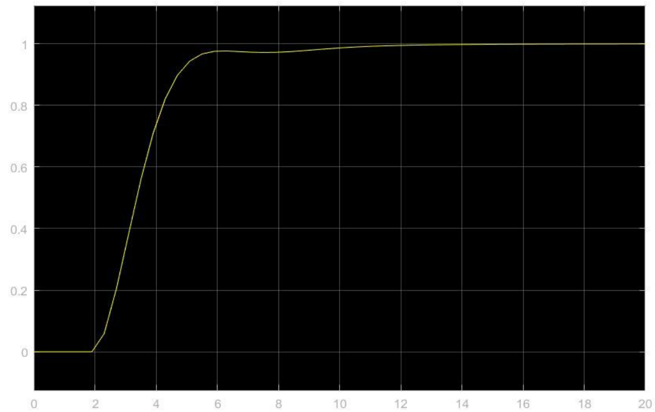

Figure 4: Step Response of IMC-PID Controller

Table 3: Parameter Comparison of CHR-PID And IMC-PID Controller

Sr No	Parameter	CHR-PID	IMC-PID
1	Peak Overshoot	5.87%	2.97%
2	Rise Time	1.6	1.58
3	Settling Time	5.39	4.36

5. CONCLUSION

In this paper Control systems for pitch control of an AUV is designed. A mathematical model was designed to control the pitch of the AUV and the same was simulated in MATLAB/SIMULINK to test the stability and performance characteristics. The simulation results shows that the IMC-PID controller gives better performance as compared to CHR tuned PID controller.

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