Practical PID Controller Tuning for Motion Control

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Abstract— **The most popular control method in the industry is PID control due to its simple structure and effective performance. Although huge numbers of PID controller tuning methods have been proposed so far, existing ones still do not have the desired performances and the simplicity. Complex system dynamics make it challenging for engineers and students to apply these methods on their applications especially in the motion control and robotics areas. Such systems generally include nonlinearity, friction, varying inertia and unknown disturbances which make the conventional tuning methods ineffective and too complex to be used. There is need for simple and effective PID tuning methods in these areas. In order to solve this problem, this paper proposes two novel practical PID tuning methods for motion control systems. These methods bring the superiority of the 2 degree of freedom control approach to simple PID controller structures analytically. They are very effective in motion control and robust both to parameter uncertainty and unknown disturbances, yet very simple. They can be easily used by the engineers in the industry and the students with very basic control knowledge, so little effort and time. The tuning methods of robust PID and PI controllers with velocity feedbacks are proposed, for position and force control problems of servo systems, respectively. The validities of the proposals are verified by the experimental results.**

Keywords—PID controller, tuning, robustness, motion control robotics, position control, force control

I. INTRODUCTION

Proportional-Integral Derivative (PID) control is the most widely used control in the industry [1]–[2]. The reasons for its popularity are its simple usage and effectiveness [3]. So far, several PID tuning methods have been proposed [4]. However, these existing methods do not have the desired performances and simplicity for many applications i.e., an important part of the PID controllers in the industry have bad performances due to their poor tunings [1].

The most common tuning method was proposed by Ziegler and Nichols [5]. This method needs many iterations for tuning, gives big overshoot and the robustness to varying system parameters is low [6]. There is also the popular Cohen-Coon formula which is derived to give good robustness against load disturbances, but this method is not satisfactory because of its overshoot and oscillatory response [7]–[9].

Apart from these most common methods, there are also more advanced PID tuning methods such as Rivera's method which uses the advantages of 2 degree of freedom (2-DOF) control approach [10] and many intelligent tuning algorithms such as Genetic Algorithms (GA) and Fruit Fly Optimization (FOA) [4], [6]. Although these methods enhance the capabilities of the conventional algorithms, they are too complex to be used by engineers, students and even most of researches. Therefore, these methods are not used in the industry and even not common in the academia.

Complex system dynamics are the main problem of the existing conventional tuning methods. Especially in motion control and robotics, where the main usage of PID controllers is to control servo position and force, this problem is faced more frequently. Such systems generally have nonlinearities, uncertainties, friction, varying system parameters and unknown disturbances. These properties complicate the use of the existing PID tuning methods [11]. The PID controllers tuned according to these methods, may behave very differently when the inertia changes or an unknown load disturbance is applied. This makes the motion control an even harder area for proper PID controller tuning since one must consider these complex system dynamics and uncertainties when tuning. Therefore there is need for effective but very simple PID tuning methods in motion control area.

In order to address this problem, this paper proposes two novel practical PID tuning methods for motion control; tuning of a PID controller with velocity feedback for servo position control and tuning of a PI controller with velocity feedback for servo force control. Although these methods are applied on basic PID controller structures, they use the advantages of 2-DOF control approach. They provide high robustness to varying system parameters, unmodelled dynamics and unknown disturbances. These methods are very effective for motion control and very simple to use. They can be easily used by engineers and students with so little effort and time. Their validity are verified by the experimental results.

The rest of the paper is organized as follows. In Section II, the direct tuning steps are given with explanations. In Section III, the analytical derivations of the methods are presented. In Section IV, the experimental results are given. The Section V is the conclusion part.

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II. Controller Tuning

In this section, tuning algorithms are described for both position and force control problems. A PID controller with velocity feedback and a PI controller with velocity feedback are tuned for position and force control, respectively. The block diagrams of such servo control systems are shown in Fig. 1 and Fig. 2.

In these figures and the following tuning steps, the following definitions apply;

x	Position;
$\dot{x}$	Velocity;
τ_{ext}	External force;
x^{ref}	Position reference;
τ_{ext}^{ref}	External force reference;
e	Error;
J	Inertia;
b	Viscous friction coefficient;
K_p^p, K_p^f	Proportional gain;
K_d^p	Derivator gain;
K_i^p, K_i^f	Integrator gain;
K_v^p, K_v^f	Velocity feedback gain;
τ	Control signal;
$\tau_{int}^{dis}, \tau_{ext}^{dis}, \tau_{frc}^{dis}$	Interactive, external and friction disturbances;
τ^{d^p}, τ^{d^f}	General disturbances;
$K_p^{des^p}, K_p^{des^f}$	Desired proportional gain;
$K_d^{des^p}$	Desired derivator gain;
K_{env}	Environmental stiffness coefficient;
D_{env}	Environmental damping coefficient;
J_n	Nominal inertia;
ω_n	Natural frequency;
ζ	Damping coefficient;
R	Robustness variable.

A. Position Control

Consider an ideal position control system which is linear and having no friction or any disturbance, having nominal inertia and PD controller. Select the nominal inertia as close as possible to the upper limit of the exact varying inertia, for the reason that selecting the nominal inertia higher makes the system more stable [12], [13]. Select the desired proportional and derivator gains according to the desired natural frequency and damping coefficient;

- $K_p^{des^p} = J_n \omega_n^2$;
- $K_d^{des^p} = J_n 2\zeta\omega_n$.

Select R value considering that the higher this value, the more robust the system will be to the unmodeled dynamics, friction and external disturbances. Set the real controller gains according to the following relations;

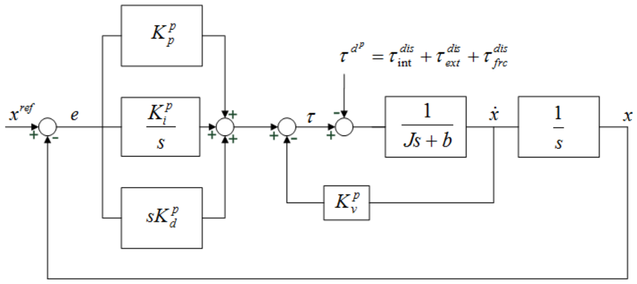

Fig. 1. Block diagram of the PID control system with velocity feedback for position control.

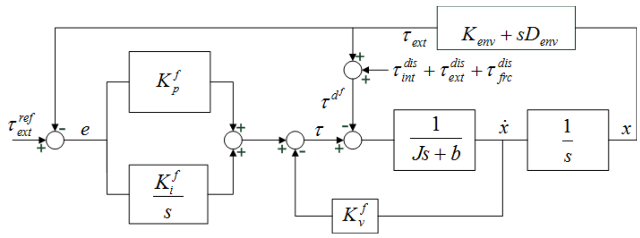

Fig. 2. Block diagram of the PI control system with velocity feedback for force control.

- $K_p^p = K_p^{des^p} + K_d^{des^p} R$;
- $K_i^p = K_d^{des^p} R$;
- $K_d^p = K_d^{des^p}$;
- $K_v^p = J_n R$.

Increase R while updating the controller gains according to it, until the system starts to be influenced negatively by practical constraints such as noise.

B. Force Control

Similarly, consider an ideal force control system with P controller, assuming that there is no external or interactive disturbance, the system inertia is nominal and close to the upper limit of the exact varying inertia. Select the desired proportional gain accordingly;

- $K_p^{des^f} = \frac{b^2}{4J_n K_{env} \zeta^2}$.

K_{env} and D_{env} are the environment parameters that are defining the material properties of the contact point in force control. The force controlled, τ_{ext}, is a function of these parameters and position. If these parameters are unknown, they can be estimated by an adaptive algorithm such as in [14]. In practice, it is harder to tune the controller gain in force control due to unknown environment. Therefore, selecting $K_p^{des^f}$ lower at first and increasing it slowly, until the system performance deteriorates, is a safer method in terms of stability. Increase R until the system is affected by the noise and set the real controller gains according to the following relations;

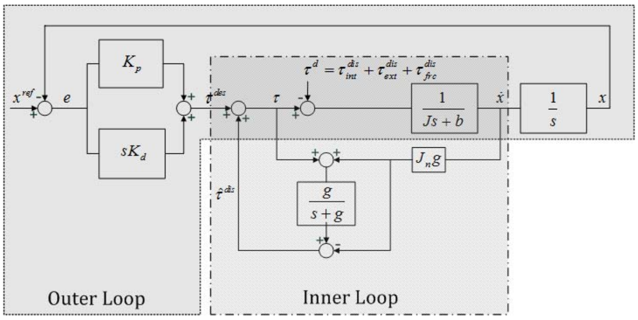

Fig. 3. Block diagram of a DOB control system.

- $K_p^f = K_p^{des^f}$;
- $K_i^f = K_p^{des^f} R$;
- $K_v^f = J_n R$.

III. Analytical Derivation

Disturbance observer (DOB), which was proposed by K. Ohnishi, is a robust control tool that is widely used in motion control systems due to its simplicity and efficiency [15]. In a DOB based control system, robustness and performance goals are achieved in the inner and outer loops, respectively. As the bandwidth of the DOB is increased, not only external disturbances are suppressed but also stability and performance are improved [12], [14]. The block diagram of a DOB based position control system is shown in Fig. 3.

These additional definitions apply in the figure and derivation;

g	Bandwidth of the DOB;
$\tau^{dis} = \tau^d + \Delta J \ddot{x} + b\dot{x}$	Total disturbance;
$\hat{\tau}^{dis}$	Estimated disturbance;
τ^{des}	Desired torque.

According to the DOB system, the equation of motion is;

$$J_n \ddot{x} = \tau - \tau^{dis}. \tag{1}$$

Where control signal is;

$$\tau = \tau^{des} + \hat{\tau}^{dis}. \tag{2}$$

The estimated disturbance can be expressed as;

$$\hat{\tau}^{dis} = \frac{g}{s}(\tau^{des} - J_n \ddot{x}). \tag{3}$$

By considering the figure, (1), (2) and (3), the control signal can be rewritten as;

$$\tau = e\left(K_p + K_d\, g\right) + \int e\,\left(K_p\, g\right) + \dot{e}\, K_d - J_n\, g\, \dot{x}. \tag{4}$$

If the controller gains of the DOB based control system and g are selected as $K_p^{des^p}$, $K_d^{des^p}$ and R, respectively, (4) is equal to the control signal of the PID controller for position control which is tuned according to the proposal. This control signal can be expressed as;

$$\tau = e\, K_p + \int e\; K_i + \dot{e}\, K_d - K_v\, \dot{x}. \tag{5}$$

Similarly, if proportional gain and g are selected as $K_p^{des^f}$ and R, respectively, the control signal of a DOB based force control system can be written as;

$$\tau = e\left(K_p^{des}\right) + \int e\,\left(K_p^{des} R\right) - J_n R\, \dot{x}. \tag{6}$$

This is the same as the control signal of the PI controller for force control which is tuned according to proposal.

IV. Experiments

Position and force control experiments were conducted using two linear motors which are shown in Fig. 4. These motors have constant inertia which are 0.4 kg each. Motor 1 was used for control experiments, and Motor 2 was used to apply external sinusoidal disturbance on Motor 1. A KYOWA LUR-A-50NSA1 force sensor was used to measure the external force applied on Motor 1, in the force control experiment. The specifications for the experimental setup are given in Table I.

A. Position Control Experiment

The first part of the position control experiment was performed without the external sinusoidal disturbance. A ramp reference input, which increases to 0.03 meters from 0 meters in 0.2 seconds, is applied at 1 second. The position control responses with the PD controller, which have $K_p^{des^p}$ and $K_d^{des^p}$ as the controller gains and the PID controller, which was tuned according to the proposal with the same gains and 600 as R, are shown in Fig. 5. As it is seen from the figure, there is some steady state error due to the disturbance of the system such as friction. The PD controller alone cannot eliminate this steady state error due to the lack of an integrator. On the contrary, the PID controller eliminates this steady state error while having the same dynamic response.

Fig. 4. Experimental setup.

TABLE I
EXPERIMENTAL SETUP SPECIFICATIONS

Parameters	Descriptions	Values
dT	Sampling period	$10^{-4}s$
J_n	Nominal motor inertia	$0.4\ kg$
$K_p^{des^p}$	Desired proportional gain for position control experiment	800
$K_d^{des^p}$	Desired derivator gain for position control experiment	60
$K_p^{des^f}$	Desired proportional gain for force control experiment	0.2
g_{vel}	Cut-off frequency of the velocity measurement filter	$600\frac{rad}{s}$
g_{sen}	Cut-off frequency of the force measurement filter	$600\frac{rad}{s}$

The second and third parts of the position control experiment were conducted with the external sinusoidal disturbance applied to Motor 1. This disturbance was started to be applied at 2 seconds in 2nd part, and at 4 seconds in 3rd part. The frequency of the disturbance is increased every two seconds to the following values respectively; 0.1 Hz, 0.5 Hz, 2 Hz, 4 Hz. In the second part, the same reference input as in the first part was applied. The position control responses of the PID controller, which was tuned with different R values, are shown in Fig. 6. As it is seen, increasing the R value improves the system robustness. While for low values of R only the low frequency disturbance is suppressed, for high values of R also the high frequency part of the disturbance is suppressed, making the system position response closer to the disturbance-free position response.

In the third part of the position control experiment, instead of the a ramp reference input, a sinusoidal reference input, with 0.03 m as the offset, 1 Hz as the frequency and 0.01 m as the amplitude, was applied to the Motor 1. The same external disturbance is also applied. The position control responses with different values for R are shown is Fig. 7. As in the ramp input case, increasing R improves system robustness, making the position control response converge to the disturbance-free control response.

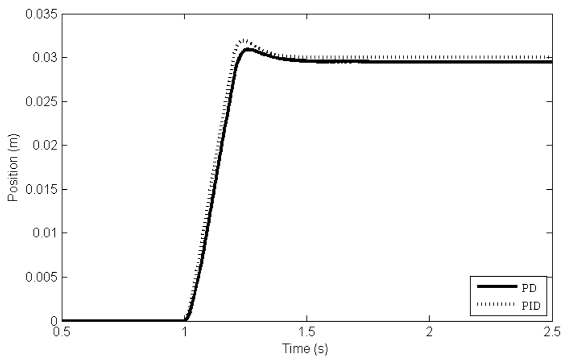

Fig. 5. PD and PID position control responses without external disturbance.

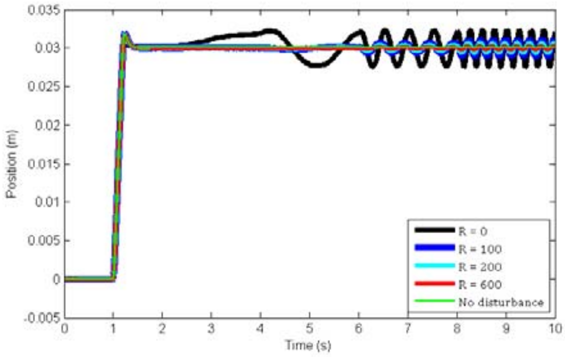

Fig. 6. PID position control responses for ramp input with different values for R.

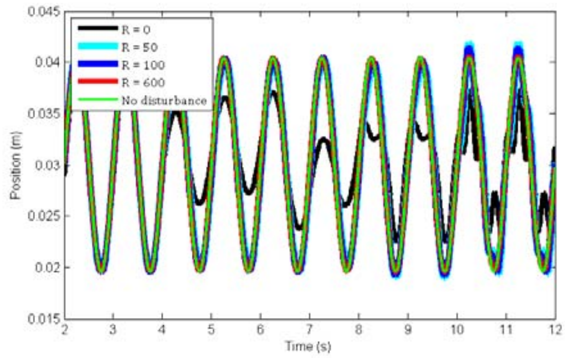

Fig. 7. PID position control responses for sinusoidal input with different values for R.

B. Force Control Experiment

The same force control experiments were conducted for environments with high and low stiffness. The first part of these experiments was performed without the external sinusoidal disturbance and on an environment with high stiffness. A step force reference, which has a magnitude of 2N, was applied at 0 second. A distance between the motor and environment contact points was kept in order to observe the impact. Force control responses with the P controller, which has $K_p^{des^f}$ as the controller gain, and the force response with the PI controller, which is tuned according to the proposal, having the same controller gain value and 100 as R, are shown in Fig. 8. As it is seen, the P controller is influenced by the disturbances too much and it cannot eliminate the huge steady state error due to the lack of the integrator. On the other hand, there is no steady state error in the PI controller case. Moreover, response is faster. The huge overshoot magnitude is due to the distance between motor and environment contact points and can be reduced by decreasing this distance.

In the second and third parts of the force control experiment, which were also conducted on environment with high stiffness, an external sinusoidal disturbance was applied. Again, this disturbance was started to be applied at 2 seconds in 2nd part, and at 4 seconds in 3rd part. The frequency of the disturbance is increased every two seconds to the following values respectively; 0.1 Hz, 0.5 Hz, 2 Hz, 4 Hz. In the second part, the same step reference input as in the first part was applied. The force responses of the PI controller, which was tuned with different R values, are shown in Fig. 9. As it is seen, the higher the R, the more robust the system becomes, and the response becomes closer to the disturbance-free case.

In the third part, a sinusoidal force reference input was applied instead of a step input, with an offset of 3N, amplitude of 1.5N, and frequency of 1Hz. The external disturbance properties were kept the same. Like in the second part, high R values give more robust control performances, as expected.

The fourth, fifth and sixth parts of the force control experiment are almost same as with the first, second and third parts, respectively. However, the only difference is that instead of an environment with high stiffness, these experiments were conducted on an environment with low stiffness. Force control responses with the P controller and the PI controller, are shown in Fig. 11. As in the environment with high stiffness, the steady state error is eliminated also in this case due to the robustness of the controller tuning method.

Force control responses of the PI controller, in the presence of the external sinusoidal disturbance, for varying R values, with step and sinusoidal reference inputs are shown in Fig. 12 and Fig. 13, respectively. The results are similar to the high stiffness cases. As R is increased, the system becomes more robust, eliminating the external disturbances with higher success. However, the response becomes more oscillatory for high values of R. This is due to the fact that low environmental stiffness value changes the system dynamics. If this situation is creating a problem for applications, $K_p^{des\,f}$ or R can be reduced.

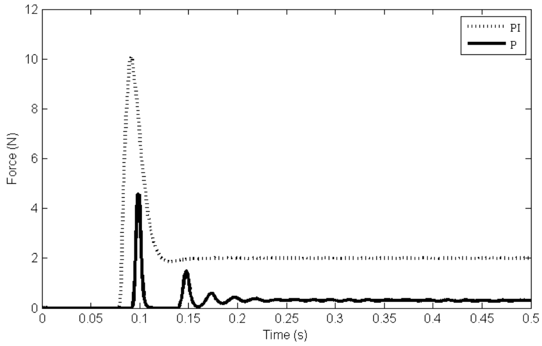

Fig. 8. P and PI force control responses without external disturbance on an environment with high stiffness.

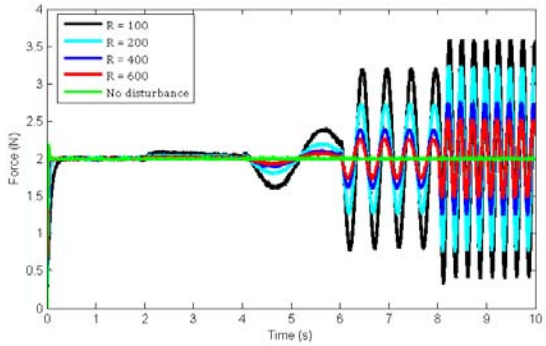

Fig. 9. PI force control responses for step input with different values for R on an environment with high stiffness.

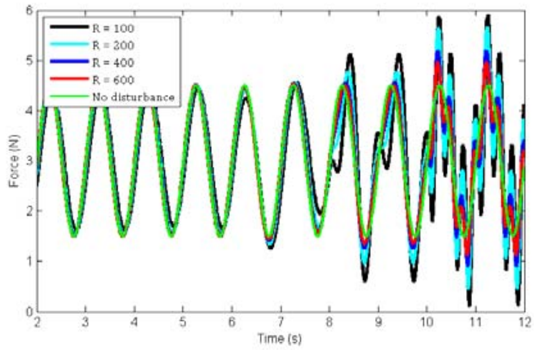

Fig. 10. PI force control responses for sinusoidal input with different values for R on an environment with high stiffness.

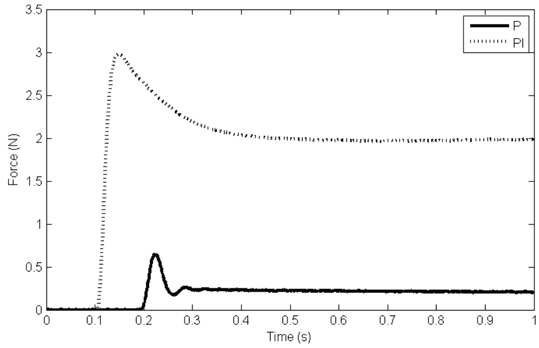

Fig. 11. P and PI force control responses without external disturbance on an environment with low stiffness.

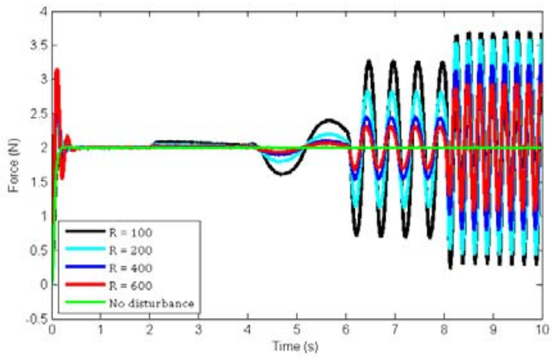

Fig. 12. PI force control responses for step input with different values for R on an environment with low stiffness.

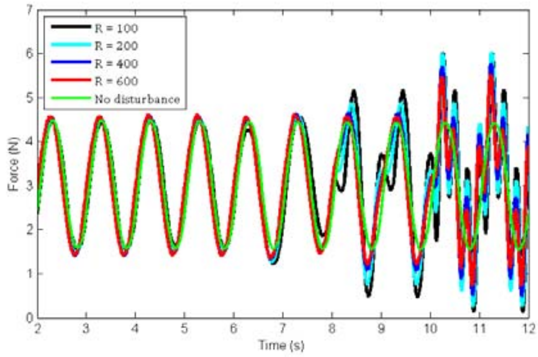

Fig. 13. PI force control responses for sinusoidal input with different values for R on an environment with low stiffness.

V. Conclusion

In this paper, two novel practical PID tuning algorithms for motion control systems are proposed. These tuning methods are for robust PID and PI controllers with velocity feedbacks, for position and force control problems of servo systems, respectively. The analytical derivations of these methods are based on the 2-DOF control approach which considers the performance and robustness separately. This superior property is achieved on simple PID structures with these algorithms in a way that the tuning takes so less effort and time. Therefore, there are very effective for motion control and robust both to parameter uncertainty and unknown disturbances, yet very simple. They can be used by the engineers and the students with very basic control knowledge just by following the direct tuning steps described in Section II. Moreover, this approach can be applied into different problems. Hence, it has a high impact on the industry and the academia. The validity of the proposal is verified with the experiments.

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