PID Tuning Methods for Active Magnetic Bearing Systems

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Abstract

The use of active magnetic bearings (AMBs) in place of traditional mechanical bearings for rotating machinery has been increasing. AMBs have been used not only in research studies, but also in a wide range of industrial applications. One of the key challenges in AMB applications is the design of control systems that maintain stability in the presence of external loads and disturbances. The most commonly used controller for AMB systems is the Proportional-Integral-Derivative (PID) controller, whose design involves the tuning of control coefficients. Since manual tuning of these coefficients is time consuming, different systematic tuning methods have been developed and reported on in the literature. This paper examines some of these tuning methods in AMB control systems. Both simulation and experiments are carried out on an AMB test rig to compare the performance of the closed-loop systems resulting from these tuning methods in terms of steady state accuracy and robustness. Simulation and experiments of each tuning method are conducted at rotational speeds ranging from 2,000 rpm to 11,000 rpm to evaluate the response peak (orbital size). Also, the frequency responses of the sensitivity function (S) under PID controllers tuned with different methods are measured and used to compare the robustness of the closed-loop system.

1 Introduction

Active magnetic bearings (AMBs) have been used in place of traditional bearing because of their many advantages. Since AMBs are noncontact support bearings, frictional loss is eliminated. The small losses that occur in AMBs are the eddy current and hysteresis losses. Magnetic bearings use the magnetic field, in the absence of lubricant or oil, to suspend the rotor. Therefore, there is no contamination to the machine environment, which is required in many applications. Since the magnetic force generated by magnetic bearings can be effectively controlled by varying the current signal, the vibration can be controlled reliably. Moreover, the new technology enables users to acquire the real-time data which can be used for an on-line diagnosis of the state of operation. Lower maintenance costs and a longer life time can be expected in the absence of mechanical wear. Operating costs are also lower than using conventional bearings due to lower bearing losses [1]. AMBs have been used not only in research studies, but also in a wide range of industrial applications, which have extended to pumps, compressors, turboexpanders, gas turbines, energy storage flywheel, artificial heart pumps, milling spindles, and many others.

The main feature of an AMB system is the control loop that generates electromagnetic forces to keep the rotor in a stable hovering position with no contact. An AMB system normally consists of position sensors, electromagnets, power amplifiers, power supplies, and controllers. Figure 1 shows the basic principle of an AMB system in one degree of freedom. The position sensor detects how far away the rotor is from the electromagnets. This information is sent to the controller, which determines proper voltage input to the power amplifiers, which in turn apply currents to the electromagnets. The electromagnets then generate magnetic forces to suspend the rotor at the desired position.

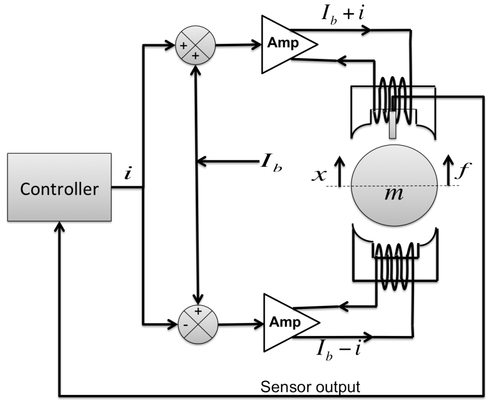


Figure 1: AMB operation in one degree of freedom

For an AMB system, there are many choices for control strategies such as the Proportional-Integral-Derivative (PID) control, H_∞ control, LQG control, and μ-synthesis [2]. The simplest control that has been most widely used in AMB systems is the PID controller. As will be reviewed in the next section, the design of a PID controller entails the tuning of three parameters. A proper choice of these parameters guarantees not only stable suspension of the rotor, but also a certain degree of closed-loop performance in terms of transient response, steady state behavior, and robustness against external disturbances. Manual tuning of three parameters is prohibitive. As a result, many systematic tuning procedures have been developed to determine the values of these parameters [3–9]. These tuning procedures have been widely applied in the process control industry. The objective of this paper is to examine the applicability of some of these tuning procedures in the control of rotor-AMB systems. We will carry out this examination on an existing rotor-AMB test rig.

The remainder of the paper is organized as follows. Section 2 explains the characteristics and properties of the PID controller as well as methods for tuning its parameters. Section 3 discusses the implementation of some of these PID tuning methods on the rotor-AMB test rig. Section 4 shows the results from applying these tuning methods as well as the analysis of these results. In section 5, the conclusion is drawn on the closed-loop performance resulting from each tuning method.

2 PID Control and Tuning

This section will review how a standard PID controller works. Main properties of the PID controller in a closed-loop system will be described in terms of the effects of each controller parameter. The typical design criteria for the PID control will be explained. Last part of this section will discuss how the PID controller parameters can be tuned to achieve satisfactory stability and performance specifications.

2.1 PID Controller

The PID control takes the form of

$$U(s) = \left(K_{\text{P}} + \frac{K_{\text{I}}}{s} + K_{\text{D}} s \right) E(s), \tag{1}$$

where $U(s)$ is the controller output, $E(s)$ is the input to the controller, K_P is the proportional gain, K_D is the derivative gain, and K_I is the integral gain. The functionality of the PID controller can be summarized as follows. Proportional control generates its output in proportion to the error signal. The controller gain K_P can be adjusted to make the controller output change as sensitively as desired to deviations between the set point and the measurement. In AMB control, adding K_P leads to increased rotor stiffness and tendency to oscillate. Integral control action is proportional to the time integral of the error. It is used to help to bring the system back to the set point. Derivative control action is proportional to the rate of change of the error. This action indicates where the error is going and provides advance information on the dynamics of the system. In AMB control, increasing the value of K_D reduces the oscillation and leads to increased damping of the rotor.

After understanding the characteristics and effects of each control action, one should find the right combination of K_P, K_I and K_D for the controller to achieve the desired control performance. However, the process of finding these parameters is time consuming. Therefore, many systematic PID tuning methods have been proposed to help with the tuning of the PID parameters.

2.2 Criteria for PID Control

As presented in the previous section, each parameter of the controller has different effects on the closed-loop system. Therefore, having a proper combination of controller parameters is crucial to achieving the desired closed-loop system performance and robustness. Before we tune the controller, we must set design objectives for the tuning process. The two main criteria for PID controllers are the *performance and robustness* of the closed-loop system.

2.2.1 Closed-loop performance

The ideal performance [10] for a closed-loop control system is as follows:

- The closed-loop system is stable: the output of the system is bounded within the specified regions.
- The closed-loop system has good disturbance rejection: the controller can correct errors caused by the disturbance and prevent the system from having a large overshoot.
- The closed-loop system has a smooth and good set-point tracking: the controller can bring the system back to the desired set-point even when the set point is changed.
- The steady state error is eliminated: the steady state value of the system output is equal, if possible, or very close to the desired value.

To be more specific, there is a standard for performance of machinery with AMBs, *ISO 14839-2*.

ISO 14839-2: Specifications on the Closed-loop Performance of an AMB System. This part of the international standard [11] establishes the requirement for the steady state value of rotor vibration of rotating machinery equipped with active magnetic bearings. Evaluation of this ISO will be based on the measurements of shaft vibratory displacements at or close to the AMBs [11].

In the table above, $D_{\max}$ is a maximum displacement of the rotor from the clearance center of the radial AMB and $C_{\min}$ is the minimum gap when moving the rotor in any direction (typically, this value is set to be equal to the displacement from the surface of the shaft to the auxiliary bearing, when the shaft is centered). Different zones are classified by the magnitude of the maximum vibratory displacement. The following are the "zoning" classifications of machines.

- Zone A: newly commissioned machines

Zone limit	Maximum Vibration Displacement ($D_{\max}$)
A/B	$< 0.3C_{\min}$
B/C	$< 0.4C_{\min}$
C/D	$< 0.5C_{\min}$

Table 1: Recommended criteria of zone limits (vibratory displacement)

- Zone B: acceptable for unrestricted long-term operation
- Zone C: unsatisfactory for long-term operation
- Zone D: sufficiently severe to cause damages

2.2.2 Robustness

Robustness is the measure of how sensitive the system response is to variations in the system. The robustness of the closed-loop system can be measured in terms of the sensitivity function (S function). S function is defined as the transfer function either from *disturbance* D_1 *to controller input* ($D_1 \rightarrow V_1$) or from *disturbance* D_2 *to plant input* ($D_2 \rightarrow U_2$) as shown in Figure 2,

$$S(s) = \frac{1}{1 + G(s)C(s)}. \tag{2}$$

A related transfer function $T(s)$, called the complementary sensitivity function, is defined as,

$$T(s) = \frac{G(s)C(s)}{1 + G(s)C(s)}. \tag{3}$$

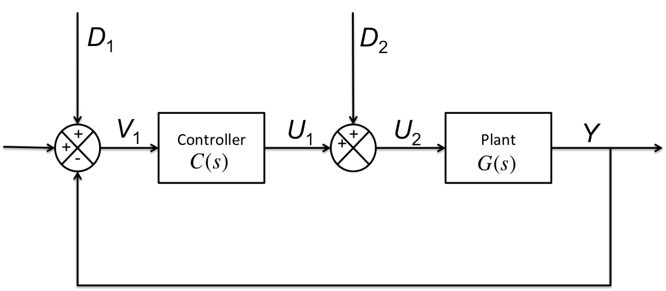


Figure 2: Closed-loop system block diagram with disturbances

The typical values from the magnitude plots of S and T functions that classify the robustness of the system are the following:

- M_S: peak value of the magnitude of the S function. As the value of M_S decreases, the closed-loop system becomes more robust.
- M_T: peak value of the magnitude of the T function.

- ϕ_m: phase margin, which indicates how much time delay can be added before the system becomes unstable. The phase margin ϕ_m is related to M_S and M_T as

$$\phi_m \geq 2\sin^{-1}\left(\frac{1}{2M_S}\right),$$

  or

$$\phi_m \geq 2\sin^{-1}\left(\frac{1}{2M_T}\right).$$

- G_m: gain margin, which indicates by how much the gain can be increased before the system becomes unstable. The gain margin G_m can be related to M_S and M_T as

$$G_m \geq \frac{M_S}{M_S - 1},$$

  or

$$G_m \geq 1 + \frac{1}{M_T}.$$

- ω_{BW}: bandwidth, which indicates the frequency range of set-point tracking of a controller.

The ISO requirement on robustness for machinery with AMBs is stated in *ISO 14839-3*.

ISO 14839-3: Specifications on the Robustness of an AMB System. This part of the international standard [12] establishes the robustness requirement of rotating machinery equipped with AMBs. The requirement is specified in terms of the peak of the sensitivity function as shown in Table 2. As mentioned above, the sensitivity function also embeds the information of the gain margin, the phase margin, and the bandwidth of the closed-loop system, all of which quantify the robustness of the system.

Zone	Level (dB)	Factor (absolute gain)
A/B	9.5	3
B/C	12	4
C/D	14	5

Table 2: Peak values of the sensitivity function (M_S) at zone limits

2.3 Tuning PID Controller

There are two main ways to tune the parameters of a PID controller. The first is manual tuning, which is based on the knowledge of the effects of each parameter as explained earlier. The second is systematic tuning, which provides the specific formula to calculate the value of each parameter of the PID controller.

2.3.1 Manual tuning

Manual PID controller tuning is a trial-and-error process based on the knowledge of the process and the effects of the controller parameters. This type of tuning requires the operation of a system and is thus sometimes called *on-line tuning*. The widely used procedures of manual tuning are suggested in [13], and are summarized as follows:

- Start with the rotor resting on the auxiliary bearing and the power amplifier on, turn off K_D and K_I. Gradually increase K_P in small increments until some oscillation occurs. At this point, the AMB stiffness has already overcome the negative stiffness by having tuned K_p only. Note that having too high a value of K_P will cause higher oscillation as explained in the effects of K_P, which is an undesired action. Therefore, we only need a sufficient amount of K_P to make the rotor begin to oscillate.

- After having some oscillation, add damping K_D into the magnetic bearing in order to reduce the vibration in the rotor. Gradually increase K_D in small increments until oscillation is eliminated, if possible, or until oscillation is very small.

- At this point, the system should be stabilized, have no oscillation, and may or may not suspend at the desired operating point. If the rotor is not at the operating point, we can gradually increase integral gain K_I to eliminate any steady state error. Typically, the amount of integral gain required is very small relative to the magnitudes of the proportional and derivative gains.

The above tuning steps are useful to stabilize the system but the procedure is quite tedious and time-consuming. It does not guarantee the performance and robustness of the system either. Therefore, these steps can be used to obtain the initial values of the parameters. We must further fine-tune these values in order to achieve the desired performance and robustness.

2.3.2 Systematic tuning

Since manual tuning for the PID controller does not involve any performance and robustness criterion, the resulting performance and robustness may not be satisfactory. Therefore, many systematic PID tuning methods have been developed that take into account the control design objectives. Before going further into the details on some of these controller tuning methods, we must introduce the required system dynamics information, which can be obtained theoretically and/or experimentally, since all tuning methods are developed based on this information. The two main methods for obtaining this required system dynamics information are the step response method and the frequency response method. In this paper, we will use the frequency response method proposed by Astrom and Hagglund [14], called the *relay feedback test*, to obtain the dynamics information. This test has a relay connected to the plant instead of a controller as shown in Figure 3. When the loop is closed, sustained oscillation occurs according to the characteristics of the relay component. The goal of this test is to determine the gain and the period as the magnitude of the relay is increased to the point where system becomes unstable (ultimate point). Also, for ease in the implementation, the dead zone band is added in order to prevent the frequent switching caused by noise [10]. The ultimate period, T_u, is the time period of the output signal at the steady state of the test. The approximated ultimate gain can be calculated as in [14]

$$K_u = \frac{4h}{\pi a}, \tag{4}$$

where h is the amplitude of the relay and a is the amplitude of the plant output. This particular test is suited for AMB system because it is a closed-loop test, and therefore, the process will not drift away from the nominal operating point. Moreover, we can choose a proper relay amplitude to constrain the size of the vibration. An example of relay feedback test results is shown in Figure 4. This test method has several advantages such as not forcing the system to its stability limit, not being time-consuming, and allowing the test to be performed without the knowledge of the plant model.

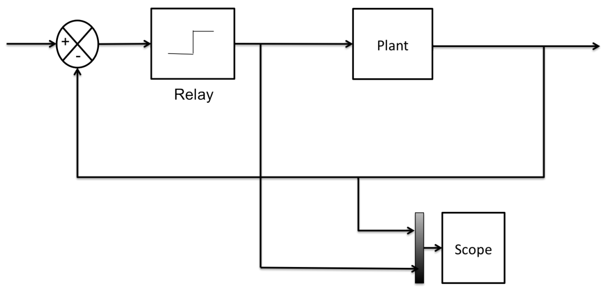


Figure 3: Relay feedback test diagram

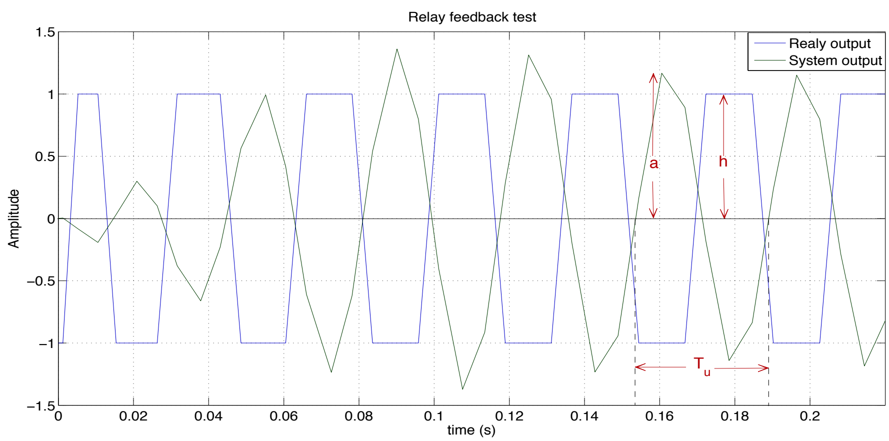


Figure 4: A typical relay feedback test result

Based on K_u and T_u information, many systematic PID tuning methods have been proposed, resulting in different formulae for determining the values of the PID parameters. In this paper, classical tuning methods that require only the ultimate gain and the ultimate period in their formulae are presented due to their simplicity and proven successes. In particular, we will examine the following systematic PID tuning methods:

1. Ziegler-Nichols (ZN) method: This PID tuning method is fundamental to many tuning methods. The goal of this method is to provide the output response of the closed-loop system to have a quarter decay of amplitude ratio when the load disturbance is varied [3].

2. Two modifications of ZN method called No Overshoot (NO-OV) method and Some Overshoot (SO-OV) method: These tuning methods are modified from the ZN method in response to the concern of the overshoot [4]. The reduction of the amount of overshoot results from a smaller proportional gain as shown in the summary table below. Also, the derivative time constant is adjusted to be larger in order to maintain the stability condition.

3. Tyreus-Luyben (TL) method: The idea behind this tuning method is to find the controller parameters such that the peak value of the frequency response of the complementary sensitivity

function T is no bigger than 2dB [5]. In other words, the main objective of the TL tuning method is to improve robustness.

4. Shinskey (SH) method: This tuning method is directly modified from the frequency response of the test proposed by Ziegler and Nichols. In most cases, the ZN method is aggressive, providing large proportional gain to the closed-loop system. Shinskey proposed a less aggressive controller by reducing the value of the proportional gain to $0.25K_{\mathrm{P}}$, which causes a slight change in the derivative gain to $0.12\frac{K_{\mathrm{D}}}{K_{\mathrm{P}}}$ [6].

A summary of the tuning formulae is shown in Table 3, where the PID controller is given in the form of

$$U(s) = K_{\mathrm{P}}\left(1 + \frac{1}{T_{\mathrm{I}}s} + T_{\mathrm{D}}s\right)E(s), \tag{5}$$

with $T_{\mathrm{I}} = \frac{K_{\mathrm{P}}}{K_{\mathrm{I}}}$ and $T_{\mathrm{D}} = \frac{K_{\mathrm{D}}}{K_{\mathrm{P}}}$.

Method	K_p	T_i	T_d
ZN	$0.60K_u$	$0.50T_u$	$0.125T_u$
SO-OV	$0.33K_u$	$0.50T_u$	$0.330T_u$
NO-OV	$0.20K_u$	$0.50T_u$	$0.330T_u$
TL	$0.46K_u$	$2.20T_u$	$0.159T_u$
SH	$0.25K_u$	$0.50T_u$	$0.120T_u$

Table 3: Summary of PID tuning methods

3 Implementation

Generally, the implementation of an AMB system with a PID controller can be shown in a block diagram as illustrated in Figure 5, Closed-loop 1. To obtain the required system information for systematic tuning, the rotor must be levitated for testing. This implies that we cannot directly apply the relay feedback test to Closed-loop 1 in Figure 5. One possible way to implement the systematic tuning method is to include an initial PID controller, manually tuned for rotor levitation, as part of the plant. We then treat the PID controller to be tuned as an outer loop controller, as shown in Figure 5, Closed-loop 2.

3.1 The Test Rig

The test rig used in this experiment, shown in Figure 6, consists of two heteropolar radial magnetic bearings. There are four sets of electromagnets on each bearing working in a differential mode. The initial PID controllers are four analog controllers used to control each axis in both magnetic bearings. The outer-loop PID controllers are implemented by using National Instrument Labview 10.0, Analog to Digital (NI 9234), and Digital to Analog (NI 9263). Labview 10.0 is used to construct the PID controller algorithm and the user interface for inputing the controller parameters. The sampling rate of this controller is 11.2 kHz. Eight eddy current sensors are mounted to measure the displacement of the rotor. The sensor gain is 6010 V/m. Eight power amplifiers, Model 422, manufactured by Copley Controls Corp., are installed in the AMB system. The amplifier gain is 0.94 A/V.

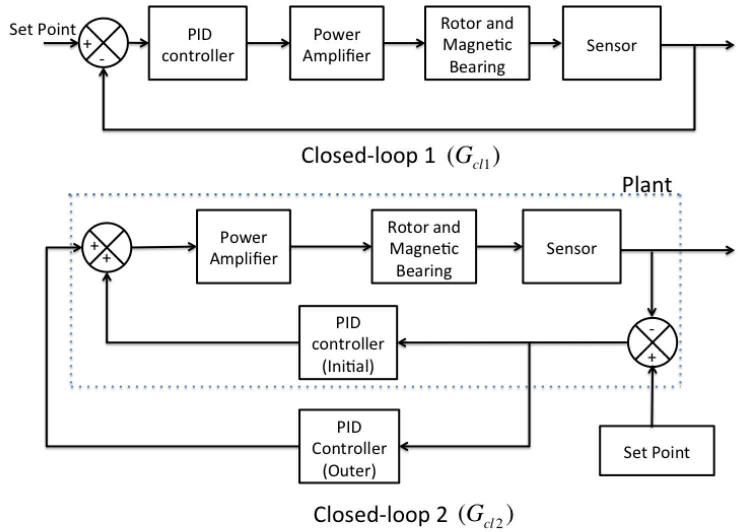
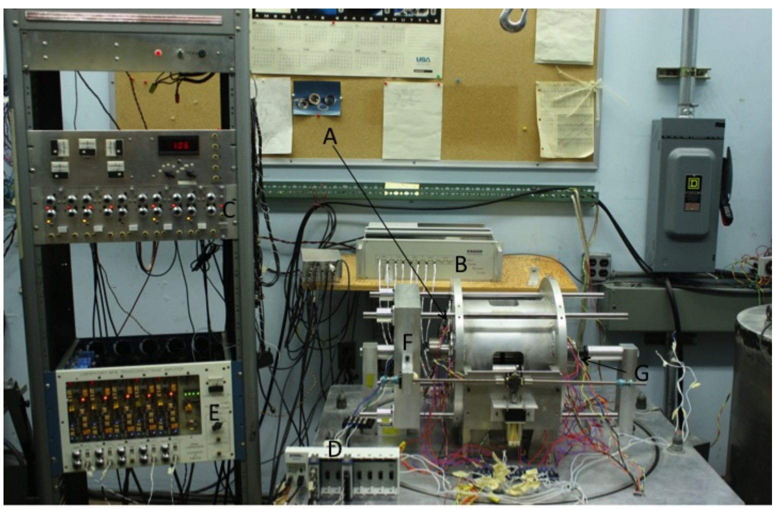


Figure 5: Closed-loop AMB systems, G_{cl1} and G_{cl2}



A: AMB B: Sensors unit C: Analog Controller D: A/D and D/A unit

E: Power amplifiers unit F: Induction motor G: Shaft (0.39 m)

Figure 6: A photograph of the test rig

3.2 Criteria for PID Tuning

We consider two criteria for the tuning of the PID controller.

1. Rotor displacement: This measures the performance of the closed-loop AMB system. A smaller rotor displacement indicates smoother operation of the system.

2. Frequency response of the sensitivity function: This measures the robustness of the closed-loop AMB system. A smaller peak value of the sensitivity function means a more robust system.

4 Results and Analysis

This section presents the simulation and experimental results of the rotor displacement and frequency response of the sensitivity function. The first part contains the results of maximum rotor displacements and the analysis of the data in order to determine the steady state performance of the closed-loop system resulting from different tuning methods. The second part contains the results of the frequency response of the sensitivity function S, from which the peak value M_s and the bandwidth ω_{BW} can be determined. For comparison, the smallest peak value of M_s of the sensitivity function resulting from an H_∞ controller is also calculated.

4.1 Rotor Displacements

The following are the results of maximum rotor displacements resulting from different tuning methods at various speeds ranging from 2,000 rpm to 11,000 rpm.

4.1.1 Simulation results

The simulation was based on the model of test rig. The results of the peak response values at various speeds are shown in Table 4 and in Figure 7.

rpm	No tuning	ZN	TL	SO-OV	NO-OV	SH
2000	3.501	2.892	1.858	1.795	2.010	2.646
3000	6.489	6.146	5.668	5.426	5.840	6.093
4000	7.126	6.992	6.338	6.042	6.608	6.866
5000	6.951	5.979	4.130	3.498	4.443	5.460
6000	3.286	1.601	1.438	1.132	1.494	1.530
7000	2.770	2.160	1.886	1.711	1.913	2.042
8000	2.098	1.835	1.511	1.343	1.594	1.767
9000	1.773	1.611	1.289	1.143	1.332	1.543
10000	1.697	1.325	1.061	0.953	1.104	1.273
11000	1.531	1.208	0.967	0.923	1.010	1.162

Table 4: Simulation results of the maximum rotor displacements in mils with each tuning method at various speeds

These simulation results show that every tuning method improves over the steady state performance of the closed-loop system under the initial PID controller. The tuning method that results in the smallest vibration is the SO-OV method and the tuning method that results in the largest vibration is the ZN method.

4.1.2 Experimental results

The original measurements of each channel's displacement from the Labview data acquisition were in terms of sensor output voltage. These results were converted by eddy current sensor gain into

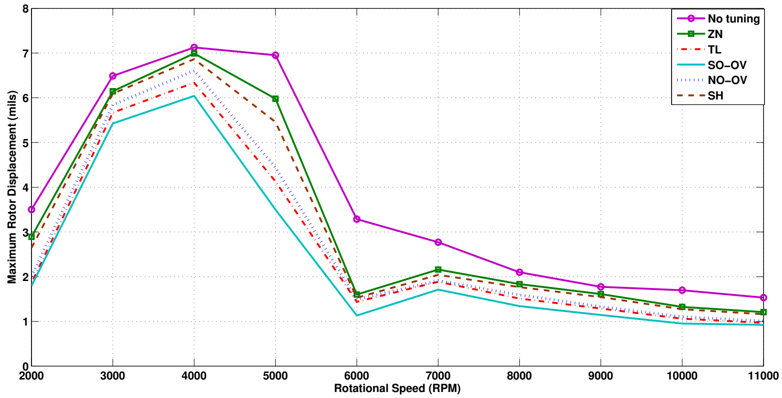


Figure 7: Comparison of simulation results of the maximum rotor displacements

mils. The maximum rotor displacements of each tuning method at different speeds are summarized in Table 5 and the trend comparison of each tuning method is shown in Figure 8. The rotor displacements were measured at various speeds ranging from 2,000 rpm to 11,000 rpm. Within this range, as can be noticed from the results, two rigid body modes were also in this rotational speed range and cause large vibrations. According to the rotordynamic analysis, the two rigid body modes of this test rig were at 4,030 rpm and 5,413 rpm, which correspond to large vibrations between 3,000 rpm and 5,000 rpm.

rpm	No tuning	ZN	TL	SO-OV	NO-OV	SH
2000	3.688	3.456	2.522	2.456	2.606	2.688
3000	6.736	6.359	6.035	5.881	6.190	6.293
4000	7.443	7.348	6.657	6.423	6.961	7.045
5000	6.969	3.233	2.446	2.375	2.584	3.076
6000	2.982	1.916	1.375	1.283	1.555	1.888
7000	2.230	1.817	1.410	1.176	1.456	1.610
8000	2.424	2.114	2.087	1.937	1.950	2.099
9000	2.084	1.634	1.351	1.168	1.458	1.465
10000	1.437	1.341	1.221	1.204	1.320	1.337
11000	1.391	1.122	0.811	0.630	0.904	0.923

Table 5: Experimental results of the maximum rotor displacements in mils with each tuning method at various speeds

These results reveal that all chosen PID controller tuning methods help to improve the performance of the AMB system in terms of the reduction of rotor vibration at various rotational speeds. The tuning method that results in the smallest vibration is the SO-OV method (39.79% improvement over the untuned case). The tuning method that results in the largest vibration is the ZN tuning method. Also, both simulation and experimental results are consistent in terms of their relative performance in reducing rotor vibration.

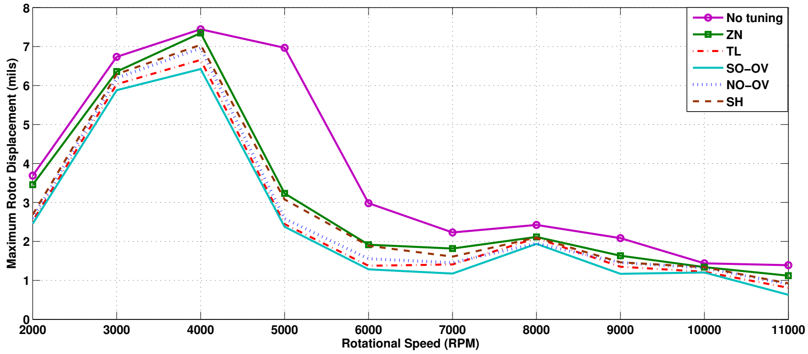


Figure 8: Comparison of experimental results of the maximum rotor displacements

4.2 Frequency Response of the Sensitivity Functions

This section presents the simulation and experimental results of the frequency response of the sensitivity function for different systematic tuning methods. In addition, the simulation results of the frequency response of the sensitivity functions with an H_∞ controller is presented.

4.2.1 Simulation results

The simulations were performed in MATLAB with a set-up similar to the diagram illustrated in Figure 2. In this case, $C(s)$ in the diagram is the outer-loop PID controller resulting from different tuning methods. The simulation results are summarized in Table 6.

Methods	M_s (absolute)	% improvement	Bandwidth (rad/s)
No tuning	11.2	-	531
ZN	7.17	35.98	602
SO-OV	4.25	62.05	584
NO-OV	5.65	49.55	563
TL	6.11	45.45	636
SH	9.34	16.61	559

Table 6: Comparison of the sensitivity function responses obtained from simulation

These results reveal that every tuning method has a smaller M_s value than the initial PID controller. The SO-OV method results in the smallest M_s value and the SH method results in the largest M_s value.

4.2.2 Experimental results

The frequency responses of the sensitivity function were measured by using dynamic signal analyzer. In the absence of controller tuning, the outer-loop PID controller was turned off. The data obtained from the dynamic signal analyzer was imported to MATLAB in order to plot both the S and T function responses. Table 7 summarizes all the results from measurements of frequency responses of the S functions.

Methods	M_s (absolute)	% improvement	Bandwidth (rad/s)
No tuning	12.01	-	603
ZN	7.71	35.80	617
SO-OV	4.42	63.20	616
NO-OV	5.64	53.04	620
TL	6.22	48.21	623
SH	9.06	24.56	611

Table 7: Comparison of the sensitivity function responses obtained from experiment

The experimental results of the sensitivity function responses reveal that every tuning method leads to a smaller M_s value than the initial PID controller. The SO-OV method results in the smallest M_s value (62.05% improvement over the untuned case) and the SH method results in the largest M_s value. Again, both simulation and experimental results are consistent in terms of their relative M_s values.

In addition, to determine the optimal value of the sensitivity function, an H_∞ controller is designed to achieve the smallest peak value of the sensitivity function. The simulation result when applying this H_∞ controller is shown in the Table 8 below. The result shows that the H_∞ method reduces the value of M_s to 1.79, which is smaller than those resulting from all the five tuning methods considered in this paper. This value of M_s lies in zone A of *ISO 14839-3*.

Methods	M_s (absolute)	% improvement	Bandwidth (rad/s)
H_∞	1.79	84.02	431

Table 8: Sensitivity function response under an H_∞ controller

5 Conclusions

The simulation and experiments were carried out on an existing rotor-AMB test rig to examine the closed-loop performance of the AMB control systems resulting from different systematic PID tuning methods in terms of steady state accuracy and robustness. The examination started from obtaining the ultimate periods and ultimate gains by using the relay feedback test. These ultimate values were used to calculate the controller gains for each systematic tuning method. The tests of each tuning method were conducted at various rotational speeds ranging from 2,000 rpm to 11,000 rpm.

We first compared a maximum rotor displacement in order to determine the steady state performance of the closed-loop system with different tuning methods. The results from the simulation and the experiment agreed and revealed that all of the chosen tuning methods improved the steady state performance. The tuning method that results in the smallest vibration is the Some Overshoot (SO-OV) method and the tuning method that results in the largest vibration is the Ziegler-Nichols (ZN) tuning method.

Next, we compared the frequency response of the sensitivity function, whose peak magnitude was used to determine the robustness according to the criteria given in *ISO 14839-3*. The results from the simulation and the experiment again agreed and revealed that all of the chosen tuning methods improved the robustness of the closed-loop system. The SO-OV method results the smallest M_s value and the Shinskey (SH) method results in the largest M_s value. However, none of the tuning methods result in zone A of robustness criteria as recommended in *ISO 14839-3*. To identify the best possible robustness, an H_∞ controller was designed. The simulation results show that this H_∞ method reduces the value of M_s to 1.79, which falls in zone A of robustness. This implies that the systematic tuning methods improve the robustness but their sensitivity function values could still be significantly away from the optimal value.

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