

## Article

# System Identification and Sensor Calibration Methods for Commissioning Bearingless Machine Control Systems

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## Abstract

This paper presents a practical and comprehensive guide for commissioning the controllers of bearingless machines (BMs). While significant research has been conducted on the commissioning of electric machines (EMs) and magnetic bearings (MBs), relatively little attention has been given to bringing up BMs. Motivated by this gap, the paper presents step-by-step methods for commissioning BM control systems. Two major contributions are outlined in the paper. First, static characterization and sensor calibration techniques, which are necessary for levitating the machine, are introduced. Second, a system identification (System ID) procedure for each plant is presented. Frequency response functions (FRFs) are used to identify the control plant transfer function parameters and evaluate the resulting feedback control performance (command tracking and dynamic stiffness). This paper emphasizes practical testing sequences that are immediately applicable to machines. By following the procedures provided in this paper, engineers can efficiently commission BMs by first ensuring precise sensor calibration and accurate models for control tuning.

**Keywords:** bearingless machines; commissioning; system identification; frequency response function; sensor calibration; static characterization; levitation control; control signal injection; command tracking; dynamic stiffness

## 1. Introduction

This paper provides a comprehensive guide to the commissioning of bearingless machines (BMs) based on the authors' experience of prototyping and testing numerous types of BMs (ac homopolar, consequent pole, PMSM, and induction). The word "commissioning" in this paper refers to the entire set of procedures required to enable the operation of a bearingless machine (BM), including static characterization, sensor calibration, and system identification (System ID). The paper is motivated by the observation that while there are numerous studies on the commissioning and system identification of electric machines (see [1–3]) and magnetic bearings (see [4–6]), there are nearly no studies on bringing up new BMs. A notable reference on BMs is [7], which documents System ID techniques to extract command tracking and disturbance rejection transfer functions of the suspension system; however, it does not provide a procedure to extract the parameters and system properties that are required to commission a new BM. Examples of relevant literature in magnetic bearings (MBs) and electric machines (EMs) include [8], where System ID is used to model



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MIMO system of active magnetic bearing for robust control, and [9], which uses frequency response functions (FRFs) to identify  $d/q$ -axis parameters of a synchronous machine.

The primary contribution of this paper is to provide step-by-step procedures and testing strategies to commission new BMs. The paper presents practical methods for system characterization and sensor calibration that are to be followed prior to levitation and System ID based on FRFs measured by using control signal injection, which are completed as part of the control tuning process once levitation is achieved. This paper is extended from the conference paper that was published in [10]. Compared with [10], this paper adds new results, including validation of the measured machine constants, command tracking evaluation, and dynamic stiffness measurements. Furthermore, this paper provides more comprehensive methodologies than [10], including static test stand set up procedures and an advanced sensor calibration method considering iron loss. These additions provide a more complete commissioning workflow for bearingless machine control systems. To help readers follow the commissioning procedure, Table 1 is provided to summarize the commissioning workflow and indicate where each step is discussed in the paper.

**Table 1.** Summary of the proposed commissioning workflow for BMs.

| Section              | Commissioning Step      | Test Stand                | Objective                                                          | Main Outputs                                                          |
|----------------------|-------------------------|---------------------------|--------------------------------------------------------------------|-----------------------------------------------------------------------|
| Section 3            | Hardware setup          | Static and BM test stands | Prepare the static test stand and BM test stand for commissioning. | Explanation of the required test configuration.                       |
| Section 4            | Static characterization | Static test stand         | Measure the machine constants before levitation.                   | $k_\delta, k_t, k_f, k_\phi, T_m(\theta)$ , and $M_c(\theta)$ .       |
| Section 5            | Sensor calibration      | BM test stand             | Calibrate displacement sensor and encoder for stable levitation.   | Sensor gain/offset, encoder offset, and magnetic center.              |
| Sections 6.1–6.3     | System identification   | BM test stand             | Identify the electrical, levitation, and rotary plant parameters.  | $R_{dq}, L_{dq}, R_{xy}, L_{xy}, m, J, k_\delta, b_p$ , and $T_\mu$ . |
| Sections 6.4 and 6.5 | Closed-loop evaluation  | BM test stand             | Verify controller tuning and evaluate dynamic stiffness.           | Command tracking and dynamic stiffness.                               |

## 2. Overview of Bearingless Machine Control System

This section introduces the standard control architecture for BMs as shown in Figure 1. The outer motion controller consists of the Speed Controller and Suspension Controller, while the inner current control loop includes the Torque Current Controller and the Suspension Current Controller. The six-phase currents are transformed into electrical space vectors in different reference frames via the Generalized Clarke Transform to generate torque and suspension force, as explained in [11]. The Torque and Force Generation block converts six-phase currents into torque and force using the  $T_m$  matrix, which are discussed in Section 4. This control system requires precise rotor angle and displacement feedback for stable levitation, as well as accurate plant parameters for controller tuning. The transfer functions of the electrical plant are provided in the torque  $dq$  frame (Equation (1)) and the suspension  $xy$  frame (Equation (2)), as summarized in Table 2, including the plant parameters of inductance ( $L_{dq}, L_{xy}$ ) and resistance ( $R_{dq}, R_{xy}$ ). The rotational dynamics of the rotor around the  $z$ -axis are described by Equation (5), where  $J$  is the rotor's moment of inertia,  $\omega(t)$  is the angular velocity,  $T_{em}(t)$  is the electromagnetic torque,  $b_p$  is the damping coefficient, and  $T_\mu$  is the friction torque. When the friction torque  $T_\mu$  is small enough to be neglected, which is typically true in BMs, the rotary motion transfer function can be approximated as Equation (6). This paper presents practical methods to determine accurate values for these parameters as observed by the BM drive.

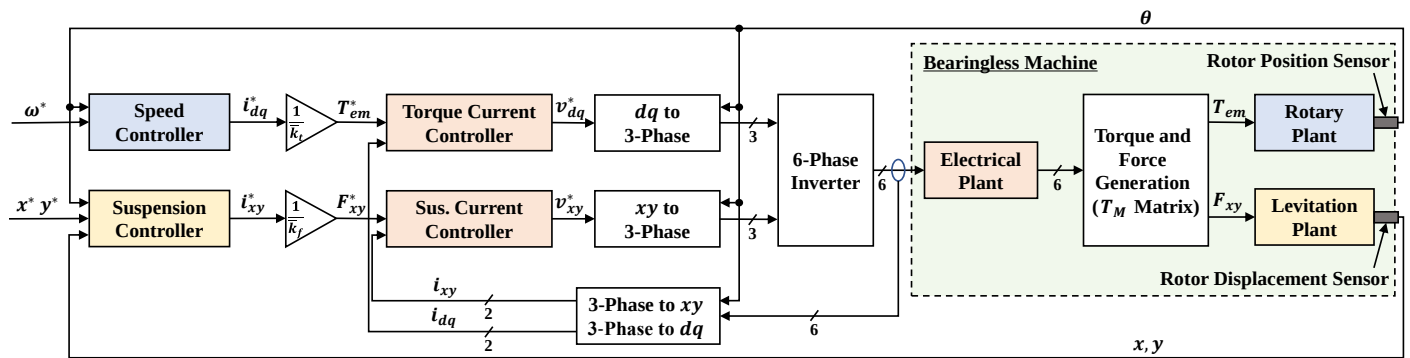


Figure 1. Control architecture for bearingless machine.

Table 2. Mathematical expressions of plant models.

|                  |                                                                                             |                                                                                |
|------------------|---------------------------------------------------------------------------------------------|--------------------------------------------------------------------------------|
| Electrical Plant | $\frac{I_{dq}(s)}{V_{dq}(s)} = \frac{1}{L_{dq}s + R_{dq}} \quad (1)$                        | $\frac{I_{xy}(s)}{V_{xy}(s)} = \frac{1}{L_{xy}s + R_{xy}} \quad (2)$           |
| Levitation Plant | $\frac{x_0(s)}{F_{x_0}(s)} = \frac{1}{ms^2 - k_\delta} \quad (3)$                           | $\frac{\beta(s)}{F_\beta(s)} = \frac{1}{Js^2 - r_\delta^2 k_\delta} \quad (4)$ |
| Rotary Plant     | $J \frac{d\omega(t)}{dt} = T_{em}(t) - b_p \omega(t) - T_\mu \text{sign}(\omega) \quad (5)$ | $\frac{\omega(s)}{T_{em}(s)} = \frac{1}{Js + b_p} \quad (6)$                   |

The control system drawing of Figure 1 corresponds to a single rotor/stator. This paper assumes the control system is extended to have a pair of BMs using centralized control, as discussed in [12]. Figure 2a shows a rigid body shaft model about  $xz$ -axis under centralized control for a 4-degree-of-freedom (4-DOF) system, which is commonly adopted for MB systems [13].  $x_0$  denotes the distance from the  $z$ -axis to the center of mass of the rotor, while  $\beta$  is the angle between the  $z$ -axis and the shaft. For  $yz$ -axis,  $y_0$  and  $\alpha$  are defined analogously to  $x_0$  and  $\beta$ . Based on this, the levitation plant models are shown in the translational Equation (3) and rotational Equation (4) domains, where  $m$  is the rotor mass,  $k_\delta$  is the stiffness coefficient,  $J$  is the rotational inertia, and  $r_\delta$  is the distance between the actuator and the center of mass.

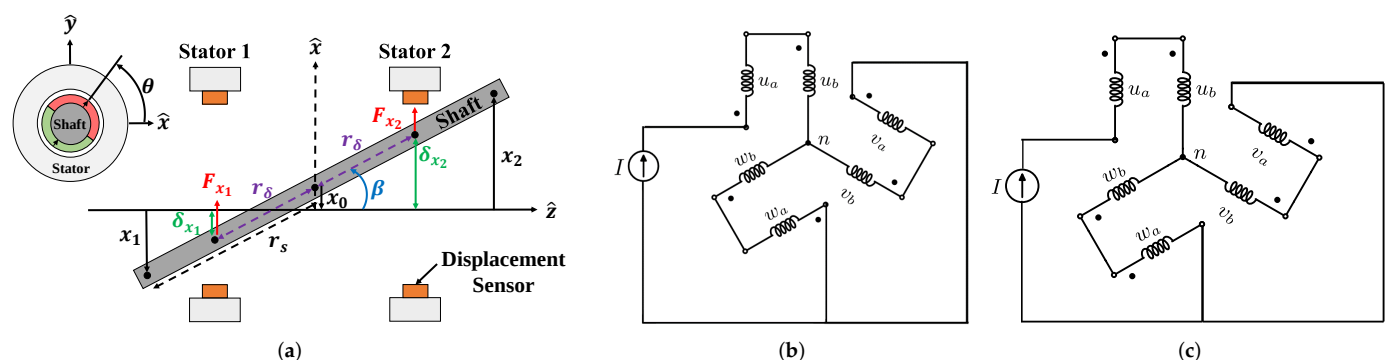


Figure 2. (a) Rigid body shaft model in  $xz$ -axis, and winding connection for measuring (b) torque, and (c) force.

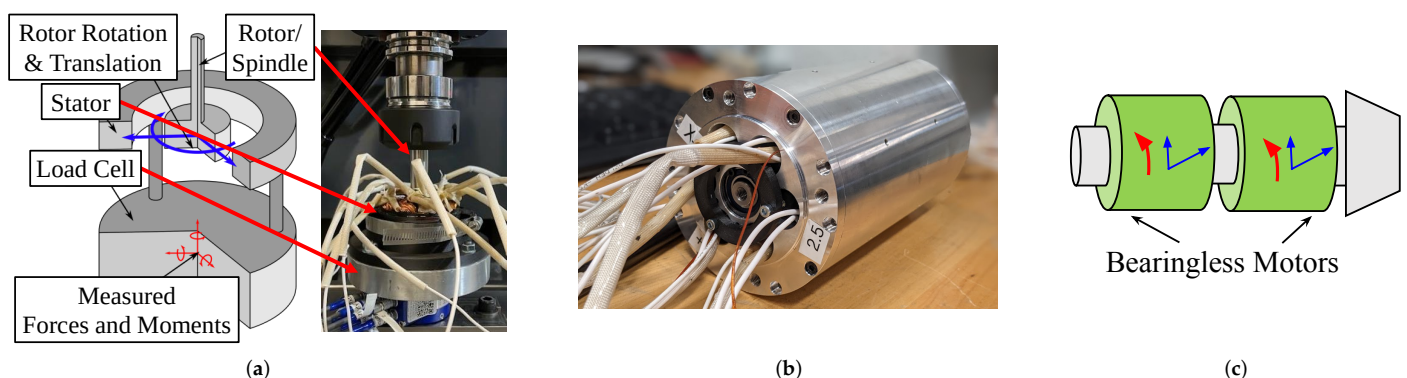
There are several types of winding topologies used to create torque and suspension force in BMs, e.g., separated, multiphase, parallel dual-purpose no voltage (DPNV), middle-point current injection, and bridge DPNV, as presented in [11]. The procedures presented in this paper apply to all of these types. At the early stages of characterization, it is convenient to use a single DC supply instead of a multiphase drive to create shaft torque or force. In separated windings, this is readily accomplished by connecting the supply to

either the suspension or torque winding. This can also be accomplished for any combined winding by connecting the winding phases as indicated in Figure 2b,c. This figure uses DPNV nomenclature (i.e., coil groups  $u_a, u_b$ ) which are mapped to multiphase winding nomenclature (phase 1, 2, etc.) as summarized in [11] (Table 1).

### 3. Description of Required Hardware

This section introduces two test stands that are needed for commissioning BMs: (1) a static test stand with means to measure force/torque and precisely position the rotor and (2) a bearingless motor with appropriate sensors (encoder, eddy current sensors), backup bearings, and a drive that supports signal injection, variable logging, and a programmable encoder offset angle. For the static test stand, a contactless dynamometer is needed to measure multi-axis forces and moments on the stator while separately supporting the rotor with its own bearings. This paper implements this test stand as shown in Figure 3a where the rotor is actuated by a spindle of a CNC mill [14], allowing accurate motion control ( $x - y$  movement and rotary angle). A similar configuration with an appropriate load cell is recommended for precise torque and force measurement. The example results presented in this paper utilize a six-axis load cell [15] mounted to the base of the stator and fixed to the mill's X-Y table. During the measurements, current is applied to the stator coils using a power supply [16] and the voltages are measured using differential probes [17].

For the second test stand, eddy current sensors are used for accurate measurement of radial displacement. In this example test stand, a Kaman KD-5690 digital differential system [18] with 20 N sensor heads [19] is selected. A precise non-contact rotor angle sensor is essential for the accurate control of BMs. In the authors' example hardware, a rotary magnetic encoder [20] with a resolution of 1024 pulses per revolution is used to measure the angular position of the rotor without physical contact. Figure 3b shows the BM prototype used in this paper, which is a one-pole-pair permanent magnet synchronous bearingless machine with a rated power of 15 kW and a rated speed of 160 kRPM. The rotor mass is measured as 0.47 kg. It consists of twin BMs to enable 4-DOF as illustrated in Figure 3c. Each machine uses a multiphase winding configuration and is driven by a SiC inverter with a switching frequency of 88 kHz and a 160 DC bus voltage.



**Figure 3.** (a) Mill setup of bearingless dynamometer, (b) BM prototype, and (c) 4-DOF diagram.

### 4. System Characterization Measurements

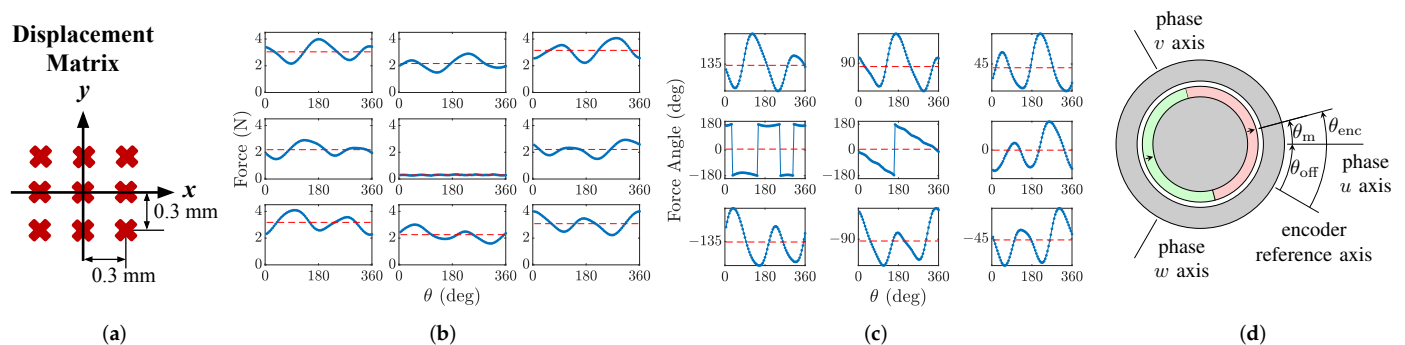
This section proposes procedures to measure the machine's constants using the static test stand described in Section 3. The constants of interest are presented in Table 3 along with governing equations relating these constants to the BM's electrical and mechanical behavior using the notation of [21].

**Table 3.** Static model equations of stiffness, torque, force, back-EMF, and  $T_m$  matrix.

| Displacement Stiffness                | Torque Constant               | Force Constant                | Back-EMF Constant          | $T_m$ Matrix                                                       |
|---------------------------------------|-------------------------------|-------------------------------|----------------------------|--------------------------------------------------------------------|
| $\vec{F} = k_\delta \vec{\delta}$ (7) | $\vec{T} = k_t \vec{i}_t$ (8) | $\vec{F} = k_f \vec{i}_s$ (9) | $v_m = k_\phi \omega$ (10) | $\mathbf{F}(\theta) = T_m(\theta) \mathbf{i}_m + M_c(\theta)$ (11) |

#### 4.1. Set Up for Static Characterization

Prior to taking measurements to determine the static constants, the following steps should be completed to align the test stand. First, the rotor needs to be positioned at the magnetic center of the machine. Doing this is necessary since the control plants are based on models that are linearized about the shaft being centered within the stator. To find the magnetic center, the rotor should be located at various test  $x$ - $y$  positions, for example, by conducting a grid search as shown in Figure 4a. At each position, a complete revolution should be rotated while recording the force and plotting the resulting force data in polar coordinates, as shown in Figure 4b,c. From these data, the magnetic center can be identified as the location where the force vector has a minimum magnitude while the force vector angle rotates 360 degrees over a revolution. For this paper's example machine, the central plot in Figure 4b,c corresponds to the magnetic center.

**Figure 4.** Set up for static characterization: (a) displacement stiffness matrix, (b) force magnitude, (c) force angle, and (d) encoder offset angle definition.

Second, the rotor angle offset needs to be identified. As shown in Figure 4d, the measured encoder angle  $\theta_{enc}$  and the rotor magnetic angle  $\theta_m$  are misaligned by an offset denoted as  $\theta_{off}$ . This offset represents the angle difference between the encoder reference axis in the test stand and the stator reference axis, which is defined here as the phase- $u$  axis. Identifying the rotor angle offset  $\theta_{off}$  is essential to avoid phase error in the measured data. This is used to align the force and torque terms with the cogging torque when completing the  $T_m$  matrix in the following sections. To measure this offset, the test stand is commanded to sweep the rotor's angular position while reading  $\theta_{enc}$  from the test stand encoder. A DC current is applied as shown in Figure 2b to generate torque, which is measured by the load cell. The measured torque should be approximately sinusoidal with rotor angle. The rotor offset angle is determined from the negative zero crossing of the torque waveform, as this corresponds to the  $d$ -axis, and is used for the subsequent measurements.

#### 4.2. Displacement Stiffness

Displacement stiffness  $k_\delta$  in BMs refers to the destabilizing radial electromagnetic force  $\vec{F}$  generated by the stator in response to rotor displacement  $\vec{\delta}$  from the magnetic center. To estimate  $k_\delta$ , the rotor is rotated fully at different radial positions shown in Figure 4a and the average magnitude of the force vector is recorded.  $k_\delta$  is determined by the slope of the linear fit based on Equation (7), as shown in Figure 5. Here, the resulting stiffness is approximately 6.76 kN/m.

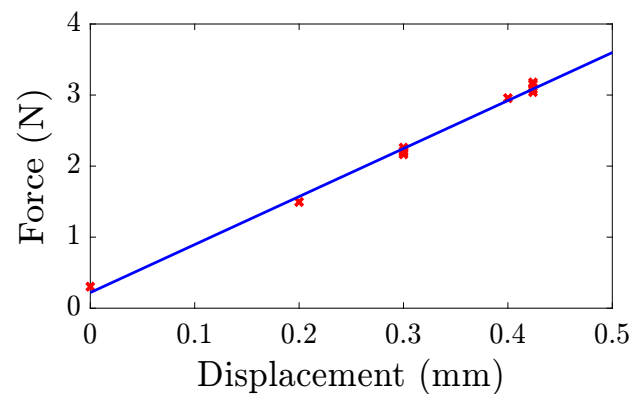


Figure 5.  $k_\delta$  curve fit.

#### 4.3. Torque Constant

The torque constant  $k_t$  quantifies how much torque  $\vec{T}$  an EM produces per torque current  $\vec{i}_t$  as shown in Table 3, Equation (8). The winding configuration in Figure 2b is used to obtain torque due to its capability of torque-only generation with a single DC source. Similar to finding the offset angle in Section 4.1, a DC current is applied to the winding while the rotor is rotating a full revolution and the resulting torque is measured as shown in Figure 6a. The peak torques satisfying the maximum torque-per-ampere condition are extracted to plot the linear fit as shown in Figure 6b. The slope of the linear fit,  $k_t$ , is measured as 0.0164 Nm/A for this prototype.

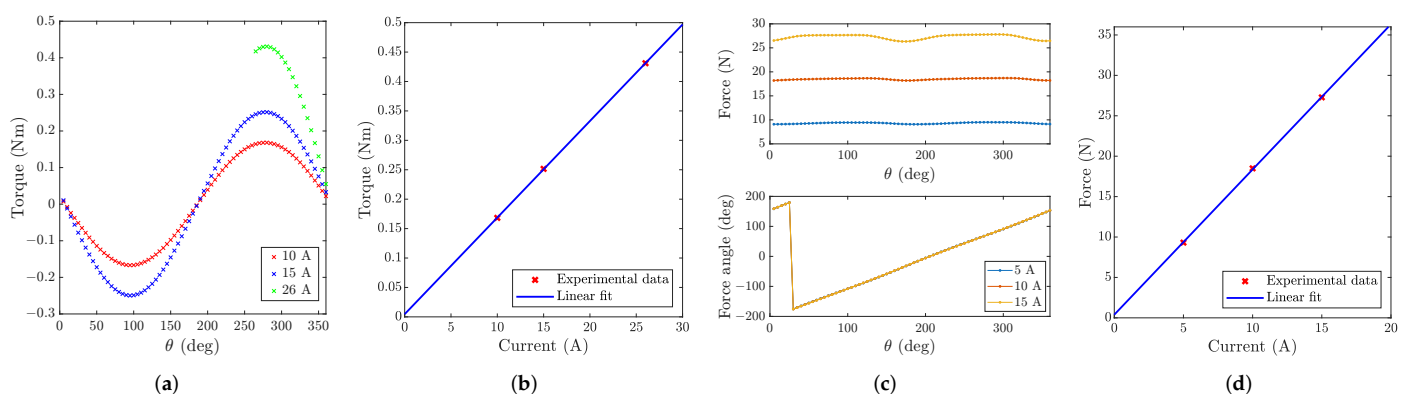


Figure 6. Static characterization results: torque versus (a) angle and (b) current, and force versus (c) angle and (d) current.

#### 4.4. Force Constant

The force constant  $k_f$  defines the relationship between the suspension force  $\vec{F}$  and the suspension current  $\vec{i}_s$  of the levitation system as shown in Table 3, Equation (9). The winding configuration in Figure 2c is used since it enables force-only generation through a single DC current source. To obtain different magnitudes of suspension forces, different levels of DC current are applied to the winding while the rotor rotates a full revolution. The generated forces in  $x$  and  $y$  axes are measured to obtain force magnitudes and angles as shown in Figure 6c. Then, a linear curve fit shown in Figure 6d needs to be generated using the extracted peak force magnitudes to obtain the proportional slope,  $k_f$ . The force constant is found to be 1.797 N/A for this prototype.

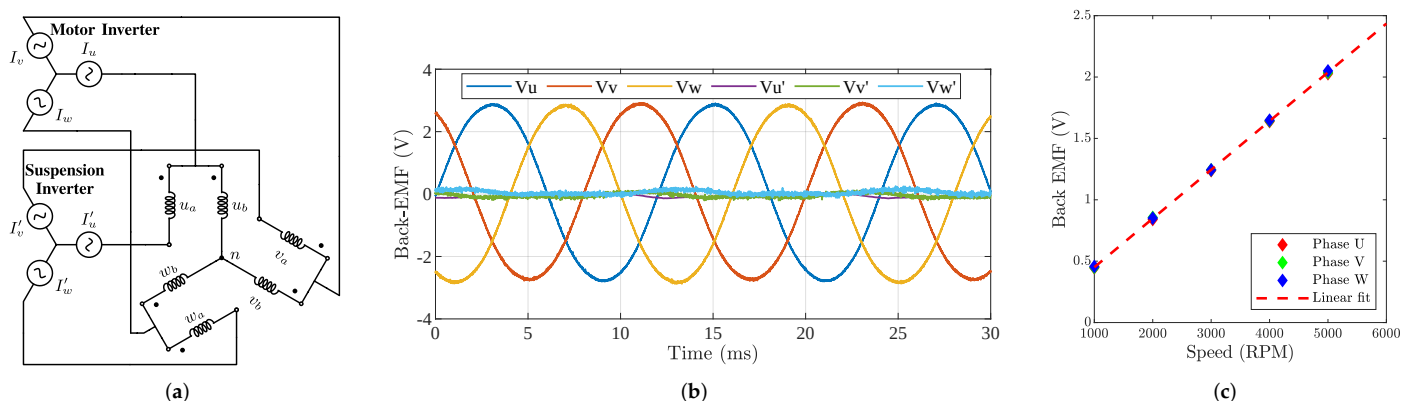
#### 4.5. Back Electromotive Force (Back-EMF) Constant

The back-EMF constant  $k_\phi$  in Table 3, Equation (10) characterizes the voltage  $\vec{v}_m$  induced in a coil per unit angular velocity  $\omega$ . This is necessary for accurately decoupling

back-EMF to achieve high-performance motor vector control and enable back-EMF-based sensorless control, as discussed in [22].

For combined windings, the authors recommend testing in the parallel winding configuration of Figure 7a as this provides convenient diagnostic information: balanced back-EMF should be induced only at the motor inverter terminals and not the suspension terminals. Any voltage imbalance in the motor terminals likely indicates an unequal number of turns among the phases, whereas a non-zero voltage at the suspension side suggests either rotor misalignment or an unequal number of phase turns; see [23].

Figure 7b presents the measured voltage when the rotor is operated at 5000 RPM. From the measured back-EMF, a linear fit of  $v_m$  shown in Figure 7c is obtained and  $k_\phi$  is found as 3.79 mVrms/(rad/s) for this prototype.

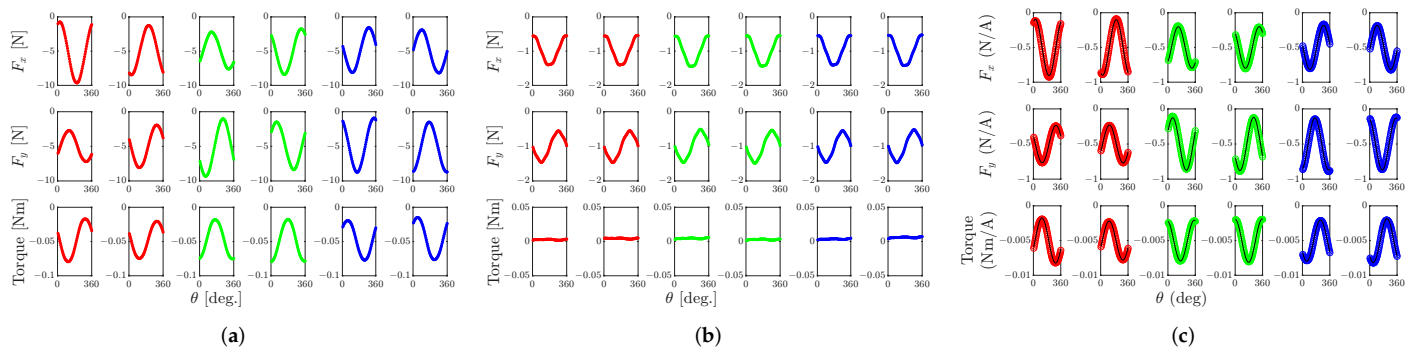


**Figure 7.** (a) DPNV winding configuration, (b) Back-EMF measurements at 5000 RPM, and (c)  $k_\phi$  curve fit.

#### 4.6. $T_m$ and $M_c$ Matrices

The  $T_m$  matrix is a generalized force-torque model that maps coil group currents to the force and torque they create, as described in [24–26]. It relates the electromagnetic force vector  $\mathbf{F} = [F_x, F_y, \tau]^T$  to the coil currents  $\mathbf{i}_m = [i_1, i_2, i_3, i_4, i_5, i_6]^T$  as presented in Table 3 Equation (11), where  $T_m(\theta)$  is a matrix representing the influence of each coil group on the force and torque,  $M_c(\theta)$  is a cogging force and torque vector. To determine the  $T_m$  matrix, the cogging force and torque are first measured without current excitation over a full period of the electrical angle, as shown in Figure 8b. Then, raw force and torque data are measured by injecting a DC current into a single coil group while rotating the rotor over one full electrical angle cycle, as shown in Figure 8a. By subtracting the cogging components from the raw data point by point, the force and torque components produced only by the coil currents are obtained, which correspond to the data points in Figure 8c. To obtain the final  $T_m$  matrix, the remaining force and torque components should be normalized by the applied DC current. Each normalized component is fitted as a function of the electrical angle using a first-order sinusoidal function, which corresponds to the fundamental harmonic component, as shown by the fitted curve in Figure 8c. The acquired  $T_m$  matrix can be used to compute the current reference vector via its inverse matrix  $K_m$  as commonly used in [27–29].

The  $T_m$  matrix is useful as a diagnostic tool and to calculate bearingless motor parameters. Engineers can confirm that each phase's characteristic is uniform and rotated appropriately from other phases. Discrepancies in this can indicate manufacturing issues, such as an inconsistent number of coil turns per phase, or measurement problems. Furthermore, the  $T_m$  and  $M_c$  matrices can be used in Simulink to enable control simulations to include full harmonic content, see [21].



**Figure 8.**  $T_m$  matrix: (a) raw data including cogging components, (b) cogging force and torque, and (c)  $T_m$  matrix.

#### 4.7. Validation of Measurements and Machine Performance

In the preceding sections, the fitted linear curves used to determine the machine constants showed strong agreement with the experimental data. This indicates the measured data were reliable and consistent with the extracted machine constants. This section shows how further validation can be obtained by cross-checking the consistency among the identified machine parameters. Additionally, ripple performance metrics of the bearingless motor are evaluated using the preceding measurements. First, the independently measured control constants are compared to validate the measurement results. Then, the machine performance is assessed by calculating the torque ripple and force vector error from the  $T_m$  and  $M_c$  matrices.

##### 4.7.1. Measurement Validation

The  $k_\phi$  value and  $k_t$  obtained from static characterization can be validated using a power balance between electrical and mechanical power. These machine constants can be measured using any combined winding topologies, which can be converted into equivalent multiphase constants (see [21]). For the six-phase multiphase winding, the total electrical power can be expressed as

$$P = 6V_{ph}I_{ph} \cos \phi \quad (12)$$

where  $V_{ph}$  is the line-to-neutral RMS phase voltage and  $I_{ph}$  is the RMS phase current. Using the amplitude-invariant space vector transformation, the RMS quantities are related to the space vector magnitudes as

$$V_{ph} = \frac{|\vec{V}|}{\sqrt{2}}, \quad I_{ph} = \frac{|\vec{I}|}{\sqrt{2}} \quad (13)$$

By exciting only the  $q$ -axis current and assuming an ideal case with no phase shift between the  $q$ -axis current and voltage, the electrical power can be written as

$$P = 3v_q i_q \quad (14)$$

Since the electromagnetic torque is  $\tau = k_t i_q$  and the voltage from the back-emf constant is  $v_q = \sqrt{2}k_\phi \omega$  (V) from Table 3 Equation (10), the power balance yields

$$k_t i_q \omega = 3(\sqrt{2}k_\phi \omega) i_q \quad (15)$$

which leads to

$$k_t = 3\sqrt{2}k_\phi \quad (16)$$

Significant differences in this value indicate a problem in either the voltage or torque measurement that should be resolved before proceeding.

The machine constants can also be validated through the  $T_m$  matrix from Section 4.6. Phase current vectors  $\mathbf{i}_m$  can be calculated to excite force or torque over a rotor revolution; see [11]. By multiplying the phase current vectors with the  $T_m$  matrix, the resulting force and torque are obtained and can be compared against the earlier measured  $k_f$  and  $k_t$  to confirm that the machine is behaving linearly. From [21] (Equation (6)), the constants are related as  $k_f = 3\hat{f}$  and  $k_t = 3\hat{\tau}$  where  $\hat{f}$  and  $\hat{\tau}$  denote the amplitudes of the fundamental sinusoidal components of the generated force and torque in the  $T_m$  matrix. These relationships can be used for a consistency check on the experimental results to verify whether the identified  $k_f$  and  $k_t$  values are reasonable.

#### 4.7.2. Machine Performance Metrics

To assess the expected torque and force that the machine can generate when it is controlled under conventional sinusoidal excitation, torque ripple and force vector error need to be evaluated. To evaluate these metrics, sinusoidal currents are first calculated as a function of rotor angle over one full revolution by using the previously measured  $k_t$  and  $k_f$ , over a range current magnitudes. Then, the currents are applied to Equation (11) to calculate the resulting torque and force vectors. To assess the performances of the expected torque, torque ripple is calculated based on the maximum deviation from the average torque as

$$T_r = \frac{\max(|T(\theta) - T_{avg}|)}{T_{avg}} \quad (17)$$

where  $T_r$  is the torque ripple,  $T(\theta)$  is calculated torque from the  $T_m$  matrix, and  $T_{avg}$  is the average torque over one revolution.

Next, force vector error can be calculated using the computed force vectors. The BMs are typically designed to achieve a force vector error angle below  $5^\circ$  to ensure stability of the levitation system (see [30–32]). Validating this metric through static testing is recommended to further characterize the prototype prior to levitation testing. The force vector error can be obtained using the following equations:

$$E_m = \frac{\max(|F - F_{avg}|)}{F_{avg}} \quad (18)$$

$$E_a = \max(|\phi - \phi_{avg}|) \quad (19)$$

where  $E_m$  and  $E_a$  represent the force magnitude error and angle error, respectively.  $F$  and  $\phi$  denote the magnitude and angle of the force. The average force magnitude  $F_{avg}$  is calculated as the root sum square of the average force components, i.e.,  $F_{avg} = \sqrt{F_{x,avg}^2 + F_{y,avg}^2}$ . The average force angle  $\phi_{avg}$  is obtained from  $\phi_{avg} = \text{atan2}(F_{y,avg}, F_{x,avg})$ .

## 5. Sensor Calibration Methodology

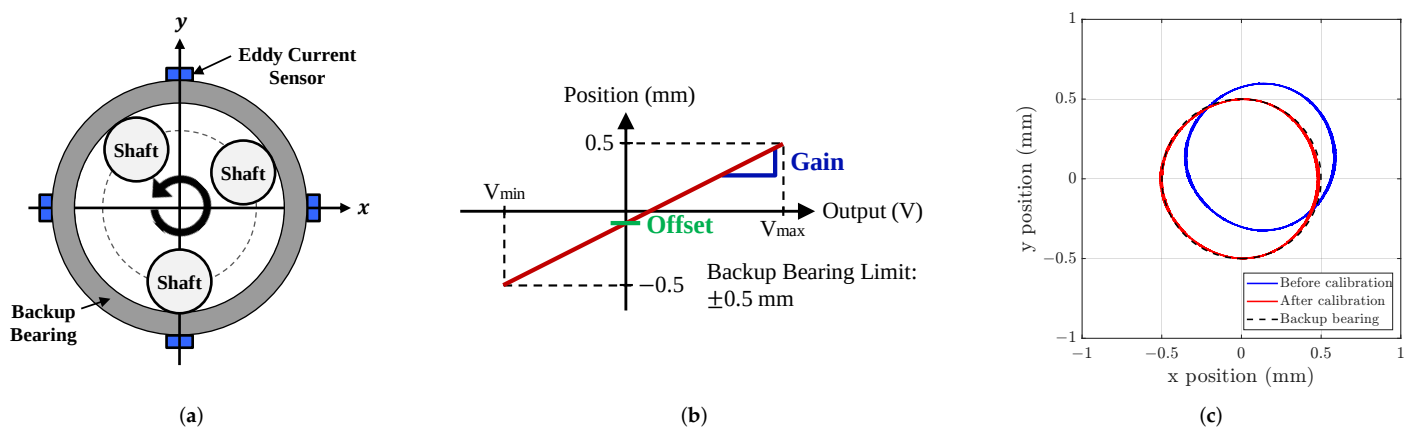
This section introduces techniques to calibrate the sensors on the second test stand described in Section 3. Following these procedures ensures accurate feedback for stable levitation control. This section provides methodologies for shaft displacement and rotary position sensors as well as a procedure for finding the magnetic center. While Section 4 similarly identifies the magnetic center and rotor angle offset on the static test stand, this section identifies them on the assembled BM test stand by calibrating the controller feedback signals needed for levitation control.

### 5.1. Eddy Current Sensor Calibration

Accurate eddy current sensor calibration is essential for acquiring accurate rotor displacement feedback. Errors in the displacement measurement can lead to incorrect

magnetic center identification and System ID results. To obtain accurate displacement feedback, precise conversion of sensor voltage to displacement measurement is required. Even when factory calibration data are available, the sensor mounting tolerances and use of a curved target may mean that it cannot be accurately applied to the bearingless motor.

This procedure assumes that the bearingless motor is fitted with backup bearings that provide a known precision clearance for shaft displacement. Suspension currents are applied to shift the rotor along the backup bearing as illustrated in Figure 9a. The sensor voltages are recorded to extract the gain and offset; see Figure 9b. Figure 9c shows the calibration result, where the calibrated output closely matches the backup bearing profile. After the calibration, repeated trials showed that the rotor displacement consistently followed the backup bearing, confirming the reproducibility of the calibration.



**Figure 9.** Eddy current sensor: (a) calibration procedure, (b) linear map, and (c) calibration result.

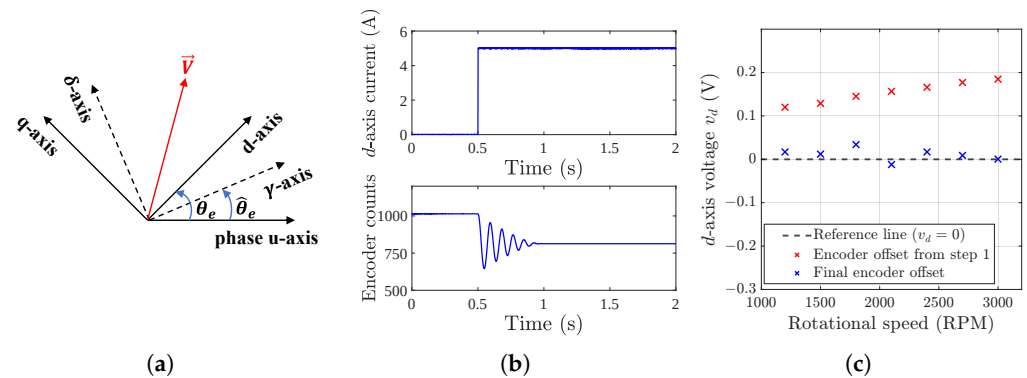
## 5.2. Encoder Calibration

Similar to finding the rotor offset in the static test stand in Section 4.1, the encoder offset in the BM test stand must be identified through angular position calibration to separate  $d$ ,  $q$ ,  $x$ , and  $y$  axes in vector control.

The angular position calibration is carried out in two sequential steps. The first step is without levitation:  $\vec{i}_t = I_0 \angle 0^\circ$  to generate a stator field that causes the  $d$ -axis of the rotor to align with the  $0^\circ$  electrical angle  $\theta_e$  of the field direction. This alignment is used to determine the initial encoder offset, and Figure 10b shows that the initial offset was determined as 813 encoder counts.

$$\begin{bmatrix} v_\gamma \\ v_\delta \end{bmatrix} = \begin{bmatrix} R + pL & -\hat{\omega}_e L \\ \hat{\omega}_e L & R + pL \end{bmatrix} \begin{bmatrix} i_\gamma \\ i_\delta \end{bmatrix} + \hat{\omega}_e \lambda_{pm} \begin{bmatrix} -\sin \tilde{\theta}_e \\ \cos \tilde{\theta}_e \end{bmatrix}, \quad \tilde{\theta}_e = \theta_e - \hat{\theta}_e \quad (\text{position error}) \quad (20)$$

The next step is to perform rotational encoder calibration while the rotor is levitated and spinning to account for dynamic factors that cannot be captured during the static calibration. The position error in Equation (20) is the angular difference between the  $d-q$  and  $\gamma-\delta$  reference frames from [22] (Equation (10)), as shown in Figure 10a where  $\gamma-\delta$  is an estimated  $d-q$  frame. The equations show that if the estimated angle offset is accurate and position error is zero,  $v_\gamma$  becomes zero and equals  $v_\delta$  when both  $i_\gamma$  and  $i_\delta$  are set to zero. Therefore, the offset is determined as the value that minimizes  $v_\delta$  to nearly zero under no load torque (where  $i_q \approx i_d = 0$  A). After Step 2, Figure 10c shows that a final encoder offset of 790 results in  $v_\delta \approx 0$  V across a range of speeds verifying the repeatability of the calibrated encoder offset over different speed ranges.



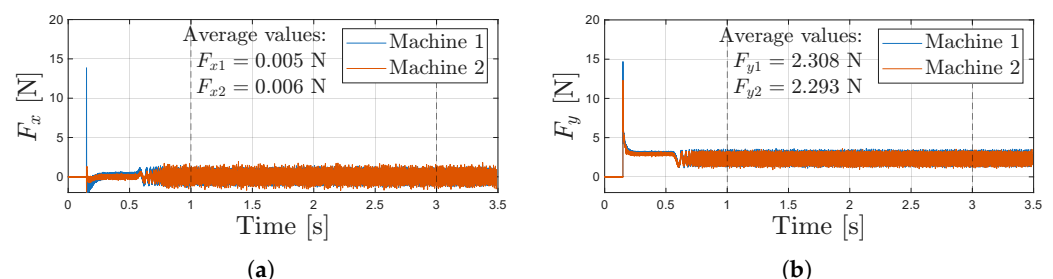
**Figure 10.** Encoder calibration: (a)  $dq$  and  $\gamma\delta$  reference frames, (b) static calibration, and (c) rotational calibration result.

Although this rotational calibration method works well to initially levitate the BM and spin it at low speed, a more advanced calibration technique is needed at high speed. Since Equation (20) does not consider any iron loss, it cannot accurately model the voltage expression. In practice, even if there is no angular error,  $d$ -axis voltage still exists due to the voltage contributing to the iron losses, which increase as electrical angular speed increases. To address this, ref. [33] suggests to calibrate the encoder by rotating the machine in the opposite direction. The offset that makes the identical  $d$ -axis voltage at the same speed but in different directions ensures the alignment of the PM rotor's  $d$ -axis with the stator phase U-axis. This calibration method is useful since it takes the iron loss into account, enabling accurate rotor position feedback at high speed.

### 5.3. Find Magnetic Center

The magnetic center (MC) is the rotor displacement where suspension forces on the shaft match the expected load. In the example test stand, the  $x$ -axis suspension force is zero ( $F_x = 0$ ) and the  $y$ -axis force should equal gravitational loading ( $F_y = mg$ , where  $m$  is the rotor mass and  $g$  is the gravitational acceleration). Identifying MC minimizes control effort and provides a reliable reference point for consistent control tuning.

The suspension forces are calculated using  $k_f$  measured in Section 4.4 until finding the MC that yields balanced forces. Figure 11 shows the forces measured at 3000 RPM at MC. The suspension forces in  $x$ -axis ( $F_{x1}$ ,  $F_{x2}$ ) for both machines are nearly zero, while the forces in  $y$ -axis ( $F_{y1}$ ,  $F_{y2}$ ) are nearly 2.3 N, which corresponds to the weight of the prototype rotor.



**Figure 11.** Suspension forces at MC of (a)  $x$ -axis, and (b)  $y$ -axis.

## 6. System Identification

Bearingless motor System ID procedures are now presented for the second test stand of Section 3. The tests require accurate machine parameters (Section 4) and sensor calibration (Section 5). The outcome is a measurement of the transfer functions in Table 2 as seen by the motor drive, which enables high-performance controller tuning. The System ID technique in this section is intended for the commissioning stage and was therefore performed at

a single representative operating condition under no-load conditions. However, since the identified parameters may vary with operating conditions, the identification can be repeated under the target conditions when higher parameter accuracy is required.

### 6.1. Electrical Plant System Identification

The transfer function for the electrical plant is given in Table 2, Equations (1) and (2). To estimate the electrical plant parameters of  $R_{xy}$ ,  $L_{xy}$ ,  $R_{dq}$ , and  $L_{dq}$ , the control block for electrical plant System ID is shown in Figure 12a. This structure is based on the control system architecture in Figure 1. To estimate the parameters, a voltage chirp signal across a frequency range is injected while the shaft is levitated, and the response of currents is obtained to construct FRFs. As an example, the curve fitted FRFs of  $x$  and  $d$  axis are presented in Figure 13a,b. The plot of the FRF demonstrates strong alignment between the experimental data and the fitted curve, and high reliability of the measured data is implied since the coherence is unity across the entire frequency range.

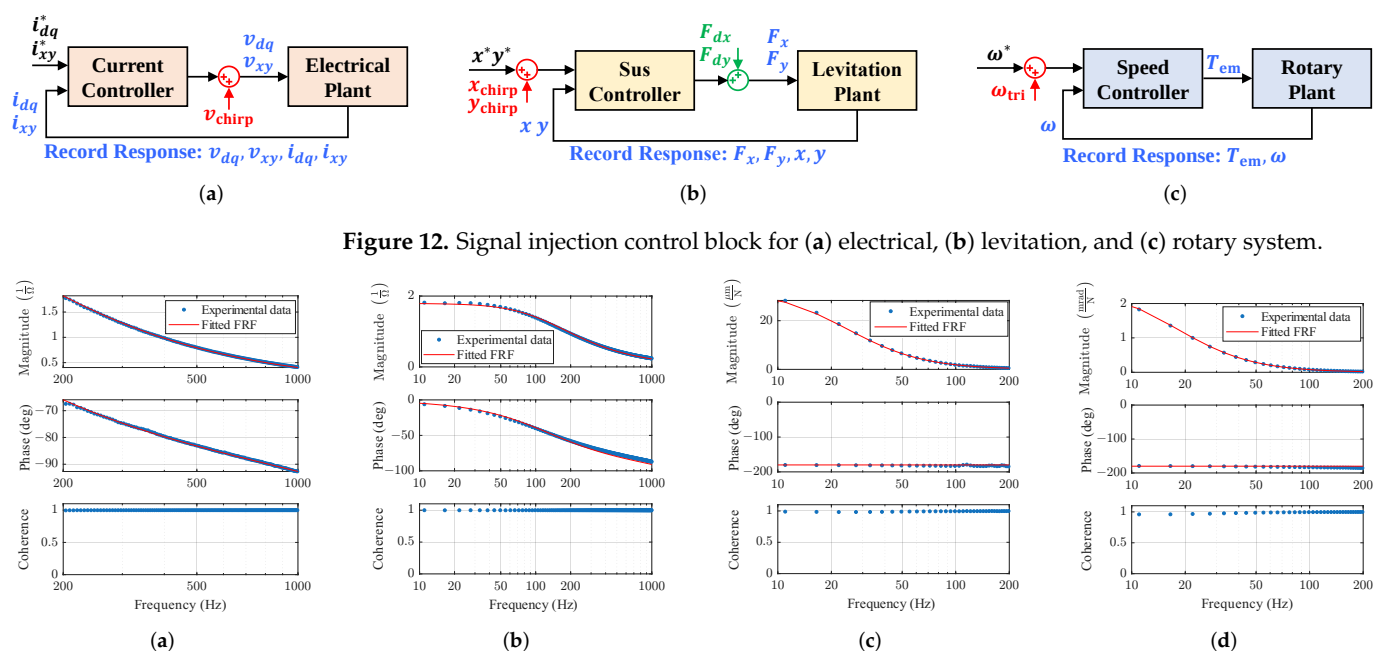


Figure 12. Signal injection control block for (a) electrical, (b) levitation, and (c) rotary system.

Figure 13. FRF results: (a)  $\frac{I_x(s)}{V_x(s)}$  and (b)  $\frac{I_d(s)}{V_d(s)}$  of the electrical plant, and (c)  $\frac{y_0(s)}{F_{y_0}(s)}$  and (d)  $\frac{\alpha(s)}{F_\alpha(s)}$  of the levitation plant.

### 6.2. Levitation Plant System Identification

As shown in Figure 12b,  $x$  and  $y$  displacement chirp signals are injected to collect the displacement response for the levitation plant System ID. Then, the suspension force  $F_x$ ,  $F_y$  is recorded to construct the FRF using the displacement response to identify the parameters  $k_\delta$  and  $m$  in Table 2, Equations (3) and (4). For centralized control depicted in Figure 2a, the same System ID methodology should be conducted for  $\alpha$  and  $\beta$ -axis to determine  $J$  and  $k_\delta$  in  $\alpha\beta$  axis. As an example of the experimental results, the FRFs of  $y$  and  $\alpha$  axes are shown in Figure 13c,d. The experimental data for both  $y$  and  $\alpha$  axes match well with the fitted curves of FRFs, which validates the identified  $k_\delta$  and  $m$  parameters.

### 6.3. Rotary Plant System Identification

To identify the parameters of the rotary plant system in Table 2, Equations (5) and (6), a triangular speed ( $\omega_{tri}$ ) can be commanded as shown in Figure 12c to obtain the flag plot in Figure 14a; see [34]. Its slope corresponds to damping  $b_p$ , width relates to inertia  $J|\omega_m|$ , and vertical offset indicates friction torque  $T_\mu$ . Sufficiently high peak speed ensures capturing

the full span of the damping effects, and constant acceleration helps to obtain a decent estimation of inertia. A triangle-shaped speed command sweeping from  $-20$  krpm to  $+20$  krpm was applied to obtain Figure 14b. The flag plot shows symmetric slopes without an offset, indicating negligible friction torque. The estimated parameters listed in Table 4 confirm the accuracy of the modeling assumptions.

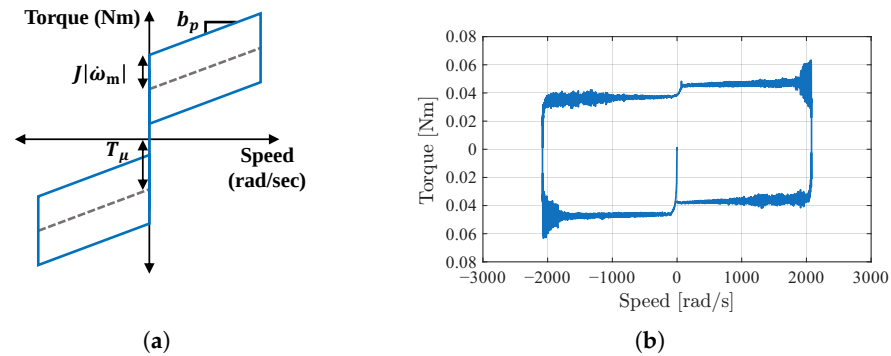


Figure 14. Flag plot of (a) ideal case, and (b) experimental result.

Table 4. Rotary System ID results.

| Parameter                                        | Estimate | Expected |
|--------------------------------------------------|----------|----------|
| $J$ ( $\mu\text{kg}\cdot\text{m}^2$ )            | 29.54    | 31.53    |
| $b_p$ ( $\mu\text{kg}\cdot\text{m}^2/\text{s}$ ) | 4.28     | 4.53     |
| $T_\mu$ (N·m)                                    | 0        | 0        |

#### 6.4. Command Tracking

To verify whether the plant parameters are accurately identified through system ID, command tracking performance is evaluated by comparing the resulting system bandwidth with the target bandwidth used for controller tuning. This can be achieved by comparing a chirp signal injected at the input of the controller and a response at the plant output. For the electrical system, a current chirp signal across a frequency range is injected as a controller command. Similarly, a position command chirp signal is injected for the levitation system. Then, the current and position response for each system is obtained to construct FRFs. Bode plots in Figure 15 show that the  $-3$  dB crossover frequencies closely match the designed bandwidths of both systems,  $1$  kHz for electrical system and  $200$  Hz for levitation system, indicating accurate identification of the plant parameters.

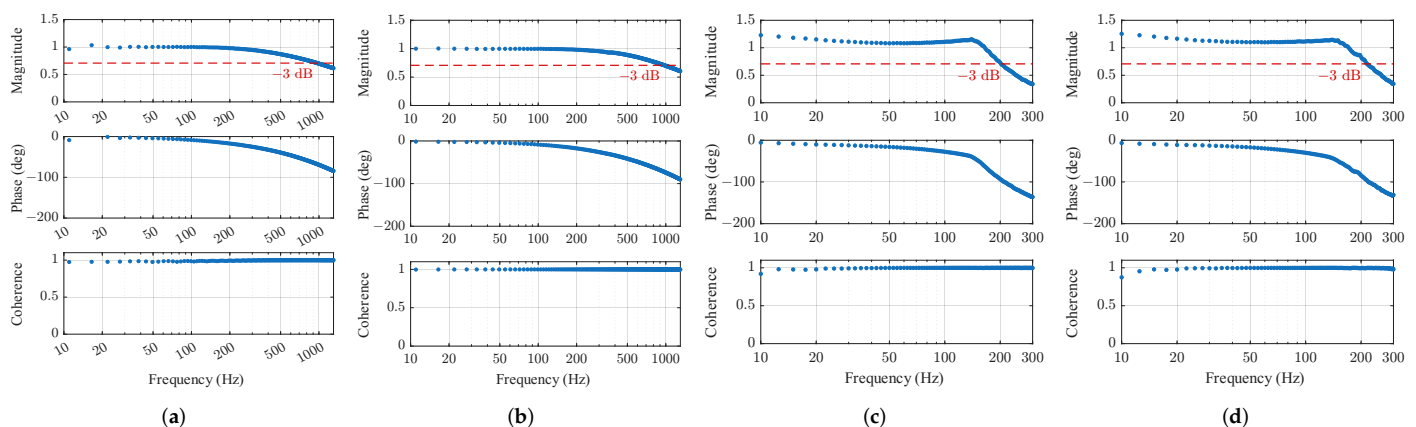
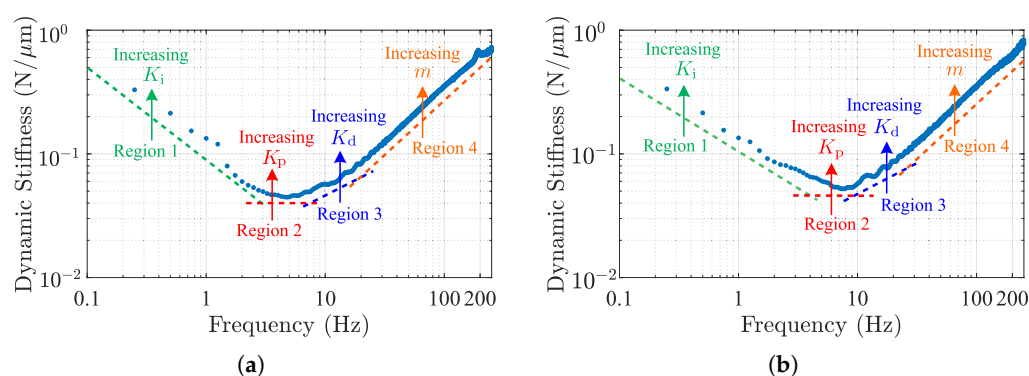


Figure 15. Command tracking results: (a)  $\frac{I_d(s)}{I_{d\text{cmd}}(s)}$  and (b)  $\frac{I_x(s)}{I_{x\text{cmd}}(s)}$  of the electrical plant, and (c)  $\frac{x_0(s)}{x_{0\text{cmd}}(s)}$  and (d)  $\frac{y_0(s)}{y_{0\text{cmd}}(s)}$  of the levitation plant.

### 6.5. Dynamic Stiffness

After stable levitation is achieved and the controllers are verified to achieve the expected command tracking performances, characterizing the disturbance rejection capability via dynamic stiffness is recommended as a final step to refine the controller tuning. In a magnetic levitation control system, the primary goal is to stably position the rotor at the magnetic center. Therefore, disturbance rejection is critical, and dynamic stiffness serves as an important metric to evaluate this capability of the levitation system in BMs (see [35]).

Dynamic stiffness is defined as the ratio of the applied disturbance force to the resulting displacement response (see [36]). To obtain dynamic stiffness FRF, disturbance forces along the  $x$  and  $y$  axis, which are denoted as  $F_{dx}$  and  $F_{dy}$  in Figure 12b, are injected as chirp signals while levitating. Then, the dynamic stiffness FRF is obtained from dividing the applied disturbance force by the measured rotor displacement response. The resulting dynamic stiffness curve of the prototype in each axis is obtained as shown in Figure 16.



**Figure 16.** Dynamic stiffness plot of (a)  $x$ -axis, and (b)  $y$ -axis.

Figure 16 shows that the stiffness curve can be separated into four regions. Each curve in each region can be shaped by tuning the different controller gains. Consider the following PID controller for the magnetic levitation system:

$$G_c(s) = K_p + \frac{K_i}{s} + K_d s \quad (21)$$

where  $G_c(s)$  is the transfer function of the Sus Controller (see Figure 12b),  $K_p$  is the proportional gain,  $K_i$  is the integral gain, and  $K_d$  is the derivative gain. Implementing this controller to the levitation plant model in Table 2, Equation (3), the transfer function of dynamic stiffness in the magnetic levitation system is expressed as the following equation:

$$\frac{F_{dx}(s)}{x_0(s)} = ms^2 + k_f K_d s + (k_f K_p - k_\delta) + \frac{k_f K_i}{s} \quad (22)$$

where  $F_{dx}(s)$  is the disturbance force and  $x_0(s)$  is the displacement response. The same approach applies to the other levitation plant model in Table 2, Equation (4). From Equation (22), the integral stiffness  $K_i/\omega$  dominates at low frequency, which corresponds to Region 1 in Figure 16. As the frequency increases, the  $K_d$  term dominates, and the stiffness increases, as shown in Region 3. In the high-frequency region, denoted as Region 4, stiffness increases with the square of the frequency and is proportional to the mass of the object. Region 2 shows the lowest stiffness since neither the control gain  $K_i/\omega$  nor the mass sufficiently suppresses the disturbances.

Note that tuning controller gains for dynamic stiffness is often in tradeoff with closed-loop stability. For example, the stiffness in Region 2 can be improved by increasing  $K_p$  at the expense of lowering the phase margin. Caution must be used to avoid unreasonably

stability margins or sensitivity to parameter uncertainty. Further consideration of these trade-offs in levitation controller tuning is presented in [37].

## 7. Conclusions

Poorly calibrated sensors and inaccurate plant parameters can all lead to unstable levitation when first commissioning a BM. Due to the high amount of interdependency in the control architecture, it is difficult to identify the root cause of failed levitation in new BMs. This paper addresses this problem by providing a structured approach to commissioning BMs that adapts proven techniques from EMs and MBs. The procedure is organized into three core phases: (1) system characterization, (2) sensor calibration, and (3) system identification, which engineers can incorporate into their workflow when bringing up new BMs.

System characterization is introduced to determine the fundamental electromechanical parameters required for control design, including the displacement stiffness, torque constant, force constant, back-EMF constant, and the generalized force-torque  $T_m$  matrix. Techniques are presented to cross-check the measured machine constants across multiple sensor and experiment types to prevent measurement errors leading to inaccurate parameter values. After determining static constants, calibration methodologies for displacement and angular position sensors are provided to ensure accurate feedback signals, and a procedure is presented to find the MC while levitating. The accuracy of the preceding measurements depends on the experimental setup used to obtain them, and the drive may observe the system differently due to inaccuracies in its own sensors. Therefore, it is important to perform a control System ID to tune the drive based on the quantities it actually sees. This paper shows how to obtain the exact transfer functions seen by the controller using System ID and presents experimental techniques to assess the controller's command tracking and dynamic stiffness performance.

The proposed commissioning workflow can serve as a practical foundation for future BM development by supporting more systematic bring-up with verifiable steps and reducing the iterative commissioning effort. Ultimately, this paper presents comprehensive and practical guidelines for commissioning BMs by providing each procedure and its expected outcome, which will be a valuable reference for engineers developing their own BMs.

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