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Torsional Vibration Measurement Validated with Dynamic Modeling on a Grinder Rotor Assembly

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Abstract

This work presents a technique for measuring torsional vibrations and validates its application. The objective was to develop and validate a methodology and the necessary tools for investigating torsional vibratory dynamics in rotor assemblies. A bench grinder's rotor assembly and casing were modified to install a high-precision, high-speed incremental rotary encoder. The encoder's output signals were processed by a custom-designed data acquisition card, which recorded the timings of 1024 pulses per revolution. This card was equipped with a microcontroller and timer modules for this purpose. The pulse timings were analyzed to derive the Instantaneous Angular Speed (IAS), representing the velocities of torsional oscillations. Several tests were conducted, and the measured signals were transformed into the frequency domain to identify the Torsional Natural Frequencies (TNFs). A 3-degree-of-freedom (3DOF) dynamic model of the rotor assembly was then used to provide analytical estimates of the first two TNFs. Discrepancies of approximately 22% and 39% were observed between the measured and modeled TNFs, which were considered acceptable given the context of the model. These results are understandable, as the dynamic modeling relied on rough estimations for the grinding disks' material properties, the squirrel cage's geometry and material, and nominal values for the steel rotor material. In conclusion, the developed setup is capable of effectively studying torsional dynamics. Future work will focus on refining the dynamic model to minimize the discrepancies with experimental results.

Keywords: Instantaneous angular speed; Torsional vibration; Rotary encoder, Rotordynamics.

1. Introduction

Torsional vibration is a critical failure mode in rotating machinery, often described as a "silent killer" since it typically occurs without apparent symptoms in lateral vibrations or performance parameters [1, 2]. Consequently, torsional vibrations are rarely monitored routinely, and they are frequently identified only after catastrophic failures of shafts, couplings, or gearboxes caused by excessive torsional stress. While measurement methods for lateral vibrations are common, manufacturers seldom integrate dedicated torsional vibration monitoring devices into critical machinery.

Among the methods for torsional analysis, the measurement of Instantaneous Angular Speed (IAS) is well-established, as it directly reflects the velocities of torsional oscillations [3]. This technique has been widely employed for the health monitoring of rotating systems with angularly periodic disturbances and non-stationary conditions, such as diesel engines [3], jet engines, and grinders. For instance, IAS measurement on a diesel engine's crankshaft has successfully enabled the detection of defective cylinders [4], while similar approaches have facilitated real-time condition monitoring on a FORD F150 engine [5] and the identification of misfire characteristics in large gas engines [6]—a critical malfunction that can lead to severe damage under high temperatures. Beyond internal combustion engines, IAS applications are diverse. In TIG welding, it controls oscillating motions to improve productivity with hard wetting materials like Ni-Cr alloys [7]. It has also been proposed for identifying damaged rolling-element bearings in automotive gearboxes to reduce downtime [8] and for the experimental characterization of partial rub in rotor-stator contact dynamics, a common cause of failure in turbomachinery [9].

The utility of IAS is further enhanced in modern industrial contexts. As noted by [10], the rise in accidental equipment interactions in the Industry 4.0 era creates a need for cost-effective, built-in shock sensing. The authors propose a framework for shock sensing using existing encoder signals, eliminating the need for additional sensors. Similarly, in machining, IAS has proven superior to cutting force measurements for identifying wear and fracture of milling teeth [11] and is effective for detecting interrupted cuts caused by robot flexibility [12]. Because cutting events affect angular speed almost instantaneously, the IAS method presents a highly promising tool for this field.

In this work, Instantaneous Angular Speed (IAS) measurement is integrated with rotor dynamic modeling to validate the experimental approach. This foundational step aims to establish a basis for utilizing torsional vibration signals—as opposed to lateral and axial vibrations—in dynamic modelling and model updating for rotating machinery. The paper is structured as follows: the first section details the experimental setup, which comprises a bench grinder, a rotary encoder, a data acquisition card, and the underlying measurement principle. Subsequently, a lumped dynamic model of the rotor assembly is introduced, and the resulting Torsional Natural Frequencies (TNFs) are reported. These modeled TNFs are then compared with those obtained experimentally through frequency analysis of the signals acquired from the tests. The discrepancies between the modeled and measured TNFs are discussed and justified. Finally, the paper concludes by addressing future work intended to improve the results.

2. Test-rig

The test rig used in this study is shown in Fig. 1. A preliminary version of this setup was presented in prior work by the authors, which detailed the initial challenges encountered during its commissioning [13]. The current work provides a more comprehensive analysis of the setup and its output signals, specifically focusing on their correlation with dynamic modeling.

2.1 Equipment

The list of equipment used in the setup is shown in Table 1. It consists of a bench grinder with nominal speed of 3000 rpm, a custom-designed data acquisition card, an incremental rotary Baumer encoder and a laptop for post-processing. Rotor assembly of the grinder includes a main shaft with

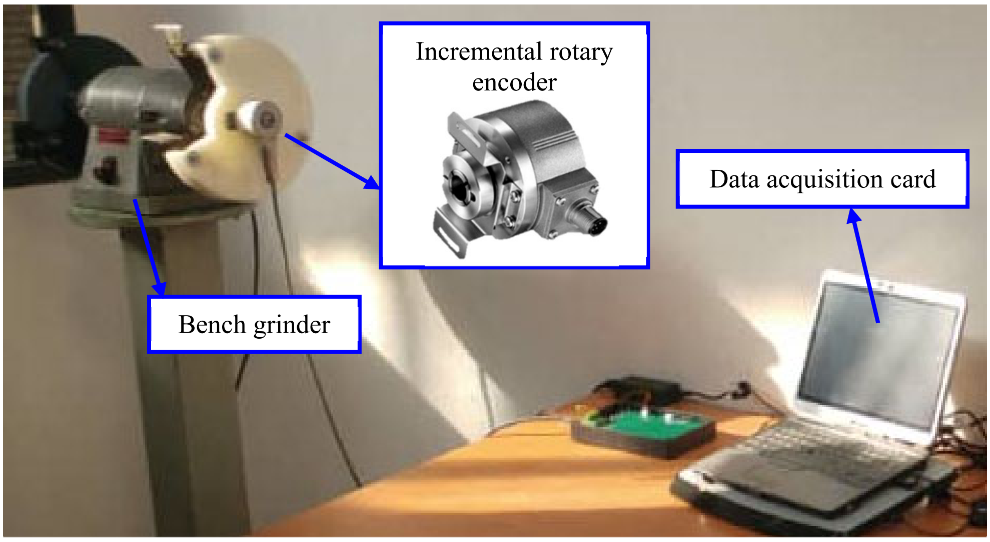


Figure 1: Test-rig

two grinding disks at its ends and a squirrel cage on the middle which receives the rotating torque from the magnetic field of the stator part. This will be further detailed in Section 3 while explaining the dynamic model. An incremental optical rotary encoder programmed with 1024 pulses per revolution is installed on an extension of this rotor assembly which has a shaft runout smaller than 0.03 mm. A custom-designed data acquisition card receives the encoder's output signals. A laptop equipped with MATLAB software is used to process the recorded signals. Some specifications of the rig are listed in Table 1.

Table 1: Equipment used

Equipment	Specifications / Accessories
MAHAK bench grinder, Type GD 200-1	3000 rpm nominal angular speed
Baumer optical encoder, Type EIL580P, hollow shaft	TTL output, resolution 1024 lines per revolution
Data acquisition card, ARM Microcontroller type	8 channels analog to digital converter,
LPC1788	$1/(120 \times 10^6)$ (s) timer accuracy
Laptop, HP Series	Elite book 2760p
Power supply	5 Volt, 1 Ampere

2.2 Measuring principle

The measurement chain is illustrated in Fig. 2. The encoder outputs two analogue square wave pulses with a 90-degree phase shift. These pulses are fed into a DAQ card, which is equipped with an ARM microcontroller and a 120 MHz timer. The DAQ digitizes the pulses, and the timer unit measures the time required to detect 8 edges from the combined pulses. This time value, c_i, is recorded to a storage device, after which the timer resets to zero.

Subsequently, a processing unit running MATLAB calculates the IAS from the clock counts c_i . Since eight edges from the quadrature pulses (A and B) represent two full periods, the corresponding angular displacement is given by:

$$\Delta\theta = \frac{2\pi}{512}\,(\text{rad}) \tag{1}$$

The time interval Δt_i corresponding to this angular displacement is derived from the clock count c_i and frequency $f_{clk} = 120$ MHz as:

$$\Delta t_i = \frac{c_i}{f_{clk}}\,(\text{s}) \tag{2}$$

Combining Equations (1) and (2) yields the IAS or instantaneous rotational frequency as;

$$\omega_i = \frac{\Delta\theta}{\Delta t_i} = \frac{2\pi \cdot f_{clk}}{512 c_i}\,(\text{rad/s}) \tag{3}$$

$$f_i = \frac{\Delta\theta}{2\pi\Delta t_i} = \frac{f_{clk}}{512 c_i}\,(\text{Hz}) \tag{4}$$

The measured instantaneous angular speed, ω_i, can be visualized in either the time or angular domain. This dual representation is enabled by the angular sampling technique, where data is acquired at constant angular increments instead of constant time intervals. The corresponding cumulative angle θ_i and cumulative time t_i for each ω_i are given by:

$$\theta_i = i\Delta\theta \ \text{(rad)} \tag{5}$$

$$t_i = \sum_{k=1}^{i} \Delta t_k \ \text{(s)} \tag{6}$$

Using Equations (5) and (6), plots of IAS can be created versus either time, angle or alternatively, revolution number of the rotor assembly. In the following sections, all representations will be thus versus the revolution number.

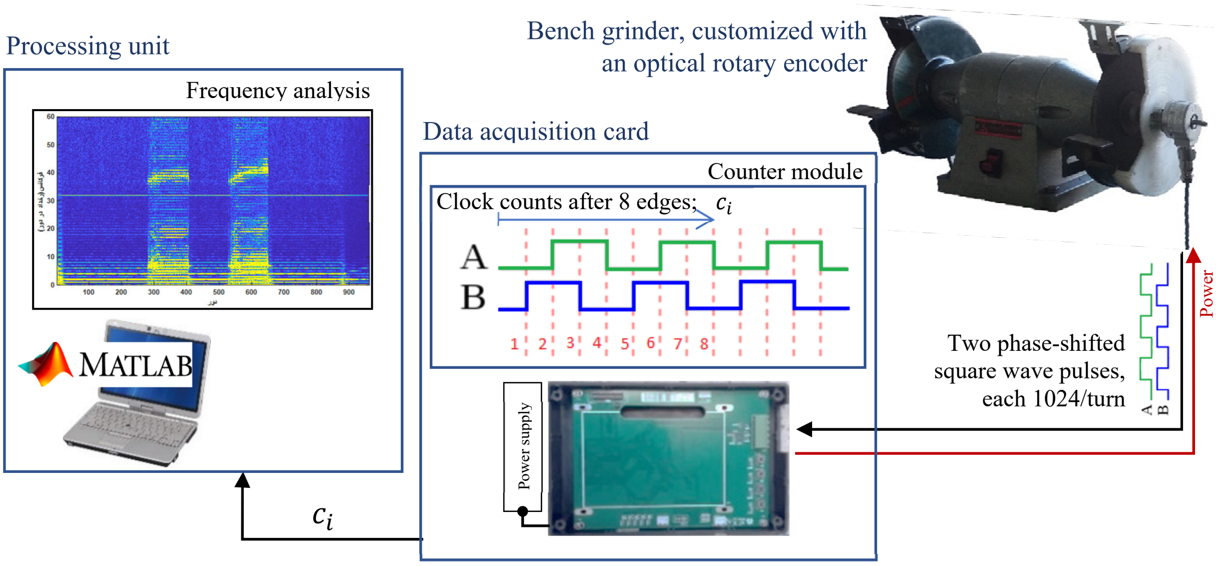


Figure 2: The measurement chain

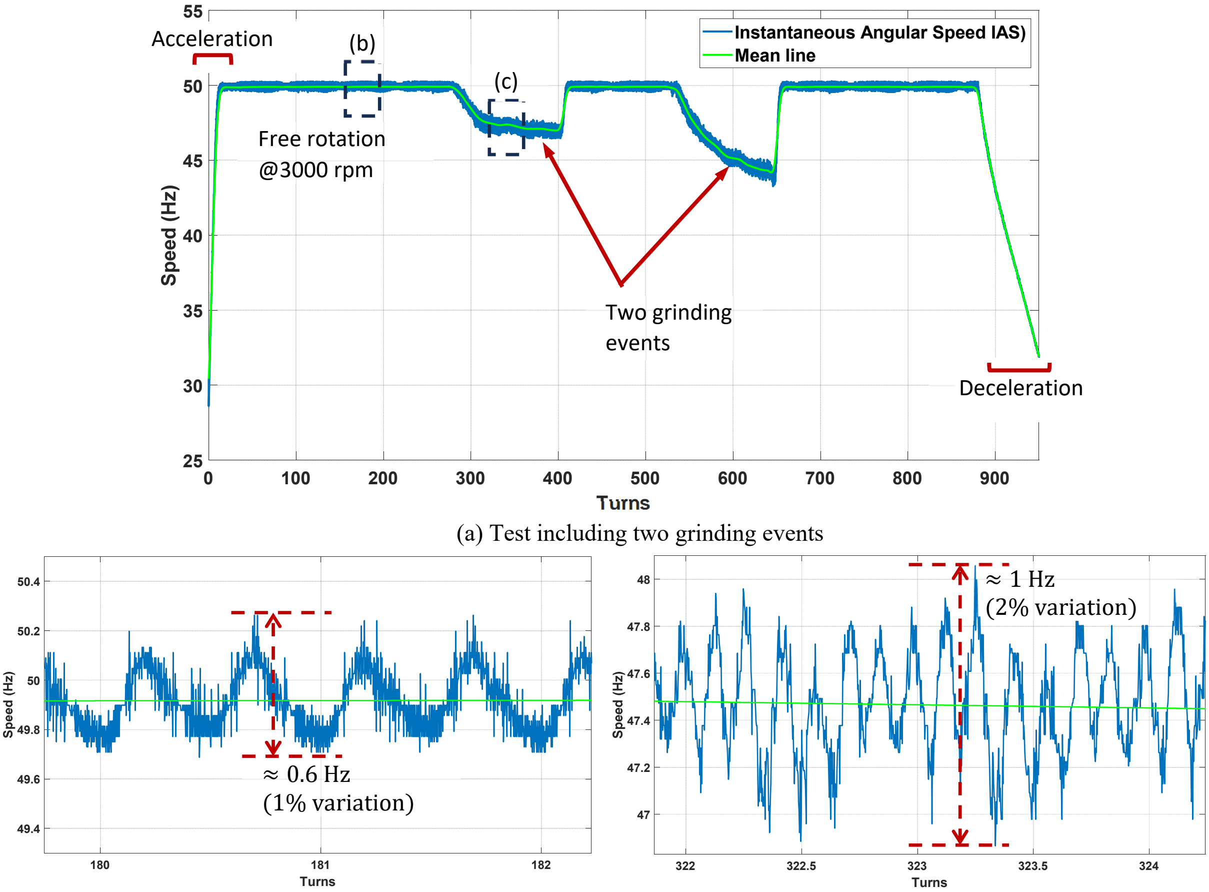
2.3 Tests and experimental analysis

The experimental procedure, detailed in Section 2.2, involved accelerating the system to its nominal speed of 3000 rpm, performing grinding events with a hand-held metallic workpiece, and then manually stopping the grinder. Initial validation of the measurements confirmed distinct acceleration/deceleration trends in rotational frequency during these events. Figure 3-a illustrates a representative test case with two grinding events. As shown in Figures 3-b and 3-c, the angular speed fluctuations are significantly more pronounced during grinding compared to those present during the wheel's free rotation.

Using the described rig, samples of instantaneous angular speed (IAS)—which represents torsional vibration velocity—are acquired at nearly constant angular increments (($\Delta\theta$). The time intervals (Δt_i) corresponding to these increments vary with fluctuations in angular speed. These fluctuations are primarily induced by the torque from cutting forces during the grinding process. During free rotation, however, they are attributed to oscillations in the motor's magnetic field.

For frequency domain analysis, the measured signals are processed using angular frequency analysis instead of the classical temporal approach. Classical frequency analysis employs a constant time-based sampling interval, with results expressed in Hertz (Hz), or events per second. In contrast, our angular domain analysis uses a sampling rate of $F_s = 512$ samples per turn. Consequently, the frequency domain outputs are also expressed in the angular domain, with the abscissa plotted in units of "events per turn" rather than Hz. This method is consistent with prior research on IAS measurement [14].

Figure 4 presents the Short-Time Fourier Transform (STFT) of the measured signal, computed using angular frequency analysis. A key TNF of the rotor assembly represents a mechanical oscillation independent of the rotor's angular speed. Consequently, during shaft acceleration or deceleration, the angular frequency of this component is expected to shift inversely. This trend is clearly observed in Fig. 4-a and 4-b for the measured TNFs at approximately $\omega_{n1}^{m} = 4$ events/turn and $\omega_{n1}^{m} = 37$ events/turn. Using the nominal rotational frequency of 50 Hz, these measured angular frequencies are converted to Hertz and reported in Table 2.



(a) Test including two grinding events

(b) Variations during free rotation

(c) Variations during grinding

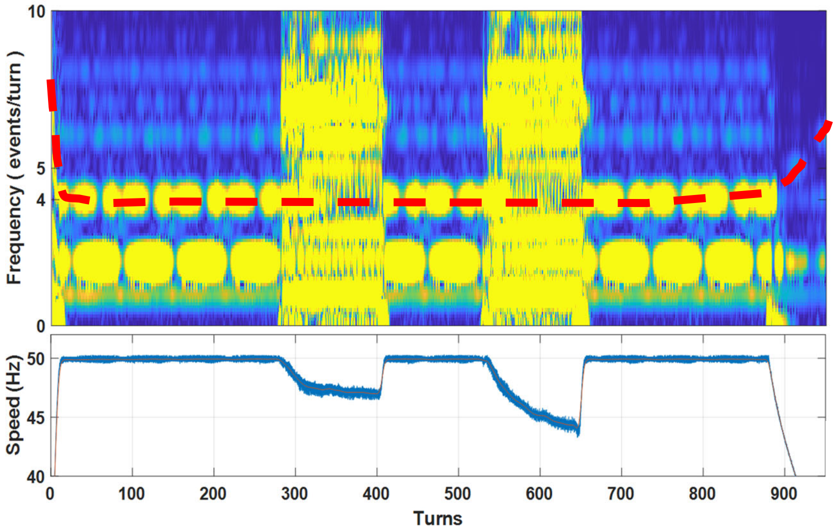
Figure 3: Measured Instantaneous Angular Speed (IAS)

3. Torsional dynamic model of the grinder rotor assembly

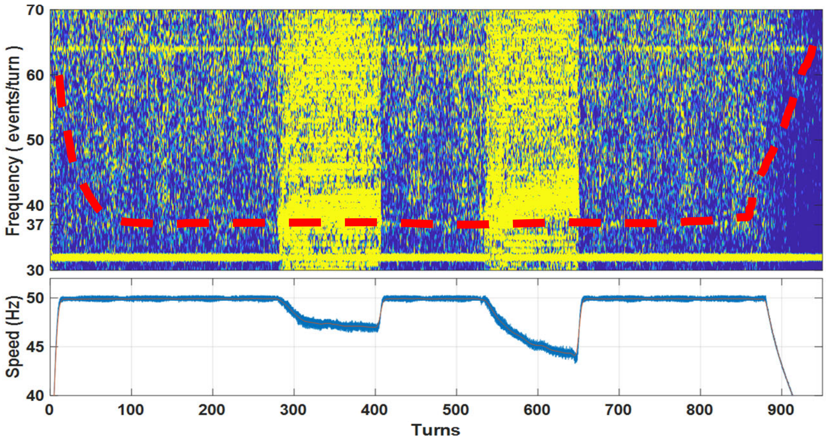
A 3DOF torsional dynamic model of the rotor assembly is developed to validate the measurements from the test rig described in Section 2. Future developments will include a more comprehensive model, such as one based on the Finite Element method. However, for this initial step, the three primary lumped inertias on the shaft (the two grinding disks and the squirrel cage) justify the use of a 3DOF approximation, which is deemed sufficient. The model's predicted TNFs are compared with the experimental TNFs to assess the validity of the measurement system. Figure 5 shows an exploded CAD view of the assembly alongside its equivalent torsional vibration model. The model comprises three lumped inertias: the two overhung grinding wheels at the shaft ends and the squirrel cage at the mid-span. The main shaft provides the torsional stiffness and damping for the system. Since the assembly could not be dismantled to directly measure all necessary parameters, their values were determined through a combination of methods: some were estimated from CAD software, others were assumed, and a few were measured directly, as detailed in Table 3.

Using the parameters listed in Table 3, the governing equation of motion for the torsional vibratory dynamics is derived as follows:

$$[M]\{\ddot{q}\} + [C]\{\dot{q}\} + [K]\{q\} = 0 \tag{7}$$



(a) First torsional natural frequency at 4 events/turn (equivalent to 200 Hz)



(b) Second torsional natural frequency at 37 events/turn (equivalent to 1850 Hz)

Figure 4: STFT on measured IAS to detect torsional resonances

Table 2: Measured torsional natural frequencies

TNFs	Value (Hz)
f_{n1}^{m}	200
f_{n2}^{m}	1850

where $[M]$ is the inertia matrix formed by the polar moments of inertia, I_d and I_r, of the disks and the squirrel cage about the shaft's rotational axis. The stiffness matrix $[K]$ is formulated from the torsional stiffness, K_t, of the shaft components. This work employs proportional damping, where the damping matrix is defined as $[C] = \alpha[M] + \beta[K]$, with coefficients $\alpha = 0$ and $\beta = 0.001$. The torsional vibratory degrees of freedom associated with the grinding disks and the squirrel cage are represented by the vector $\{q\}$.

$$\{q\} = \begin{Bmatrix} \theta_1 \\ \theta_2 \\ \theta_3 \end{Bmatrix},\ [M] = \begin{bmatrix} I_d & 0 & 0 \\ 0 & I_r & 0 \\ 0 & 0 & I_d \end{bmatrix},\ [K] = \begin{bmatrix} k_t & -k_t & 0 \\ -k_t & 2k_t & -k_t \\ 0 & -k_t & k_t \end{bmatrix},\ [C] = 0.001[K] \quad (8)$$

Once the parameters from Table 3 are used to populate the matrices in Eq. (7), the resulting eigenvalue problem is solved. This solution yields the two TNFs of the rotary system, which are reported in Table 4.

Table 3: Constant parameters used in the 3DOF dynamic model

	Parameter	Value	Units	Method obtained
Main shaft	Young's modulus ($\boldsymbol{E}$)	$\mathbf{2 \times 10^{11}}$	Pa	Assumed as steel
	Poisson's ratio ($\boldsymbol{\nu}$)	0.266	-	Assumed as steel
	Diameter ($\boldsymbol{D_s}$)	21	mm	Measured
	Effective length ($\boldsymbol{l_s}$)	171.5	mm	Measured
Grinding disks	Mass ($\boldsymbol{m_d}$)	1.770	Kg	Measured
	Inside/Outside diameter ($\boldsymbol{D_{id}/D_{od}}$)	37.5/200	mm	Measured
Squirrel cage	Mass ($\boldsymbol{m_r}$)	0.794	Kg	Estimated using CAD
	Inside/Outside diameter ($\boldsymbol{D_{ir}/D_{or}}$)	21/54	mm	Measured

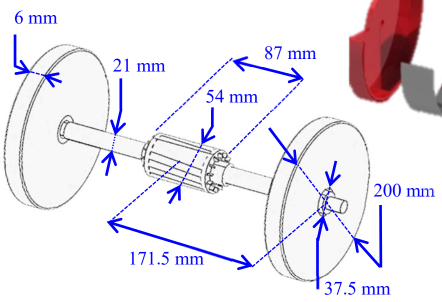


Table 4: Analytical torsional natural frequencies

TNFs	Value (Hz)
f_{n1}^{a}	156
f_{n2}^{a}	1167

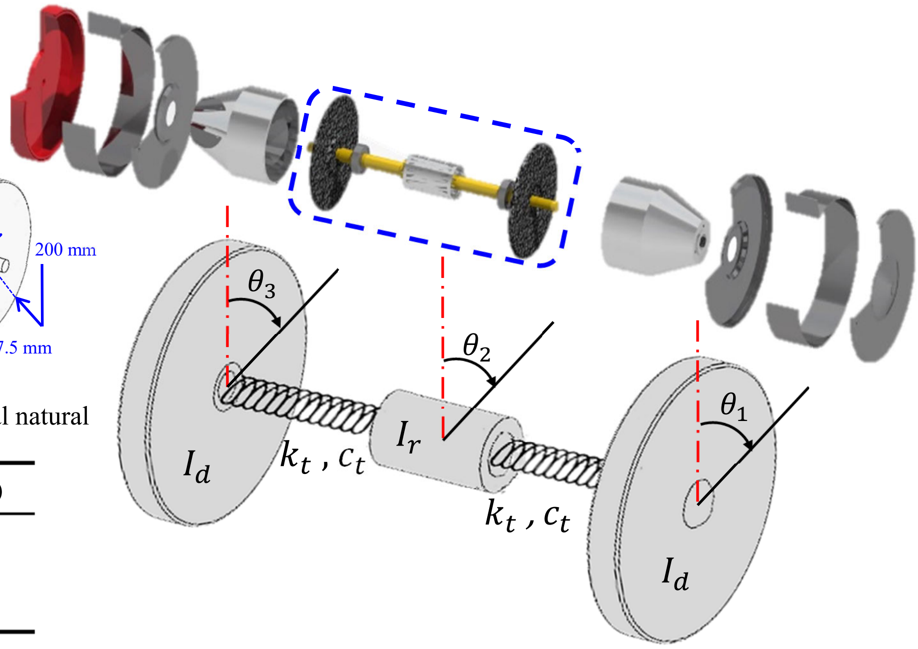


Figure 5: 3DOF torsional dynamic model of rotor assembly for bench grinder

4. Results and discussion

Table 5 compares the measured values of the first two TNFs with their analytical counterparts from the dynamic model. The discrepancies of 22% and 39% are considered reasonable, given the assumptions in the dynamic model and the uncertainties inherent in the measurement process.

Table 5: Analytical torsional natural frequencies

TNFs	Measured (Hz)	Analytical (Hz)	Error (%)
f_{n1}	200	156	22
f_{n2}	1900	1167	39

The results indicate that the developed measurement rig successfully captures the essential physical characteristics of the torsional resonances. The observed discrepancies between measurement and model are attributable to several factors. First, the 3DOF analytical model incorporates significant approximations; for instance, the material properties and some geometrical dimensions of the squirrel cage were obtained from CAD software, and the limited number of degrees of freedom cannot fully represent the real system. Second, the measurement system itself introduces noise and disturbances. The addition of an extension shaft for encoder installation inherently adds mass and compliance, which perturbs the system's dynamics. Signal fidelity could be significantly improved by using an encoder integrated directly onto the main shaft, enabling a clearer measurement of the torsional resonances.

The parameters of the dynamic model, initially based on engineering estimates, should be updated using optimization procedures to match the measured values. Parameters such as shaft's stiffness and damping coefficients, the squirrel cage's material properties and grinding disks' material which defines their inertia may contribute to the observed errors. Therefore, a primary focus for future work is model updating based on the measured torsional vibration data.

5. Conclusion

This paper presents the development and validation of a test rig for measuring torsional vibrations via Instantaneous Angular Speed (IAS). A bench grinder was modified by fitting an extension shaft to its rotor to accommodate a high-speed incremental rotary encoder. The encoder pulses are processed by a custom-designed data acquisition (DAQ) card, which utilizes a microcontroller and timer modules to record variations in pulse periods. These angle-stamp counts are processed offline to derive the IAS. The measured IAS signals are validated through frequency analysis in the angular domain, which identifies candidate torsional natural frequencies (TNFs). These experimental results are compared against TNFs extracted from a 3DOF dynamic model of the rotor assembly. Discrepancies of up to 22% and 39% between the measured and analytical TNFs are within expected margins, confirming the functionality of the measurement system and affirming the simplifications inherent in the dynamic model. Future work will focus on updating the dynamic model to improve its correlation with the experimental data.

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