

Quantifying the Impact of Drivetrain Flexibility on PID-Based MPPT Performance in Modern Wind Turbines: A Simulation-Based Comparative Study [†]

Mohamed Tahiri ^{1,*} , Yassine Lakhal ² , Soulaïman Louah ³  and Abdelaziz Mimet ⁴

¹ Laboratory of Energetic, Faculty of Science, Abdelmalek ESSAIDI University, Tetuan 93000, Morocco

² Laboratory of Engineering and Applied Physics, Higher School of Technology of Beni Mellal, Sultan Moulay University, Beni Mellal 23000, Morocco; lakhal.yassine.doc@gmail.com

³ Information System and Logistics Engineering National School of Applied Sciences, Abdelmalek ESSAIDI University, Tetuan 93000, Morocco; soulaïman.louah@etu.uae.ac.ma

⁴ Faculty of Science, Abdelmalek ESSAIDI University, Tetuan 93000, Morocco; amimet@uae.ac.ma

* Correspondence: mtahiri921@gmail.com; Tel.: +212-665789314

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Abstract

The following is a detailed account of how the flexibility of a drive train affects maximum power point tracking (MPPT) for wind turbines. The study uses the National Renewable Energy Laboratory 5 MW reference turbine as a base model with Kaimal spectrum having felt waves creating turbulent winds. This study compares both rigid and flexible drive trains that undergo identical turbulence conditions (average wind speed 9.0 m/s and flight weather turbulence, 16% T.I.). The results show that the flexible unit provides an 8.95 percent area increase in mean power coefficient, ($p < 0.001$, Cohen's $d = 2.108$) while the rigid unit produces a 4173 percent ($p < 10^{-47}$, Cohen's $d = 31.8$) sudden rise in power during catastrophic loss of control and produce a direct torque oscillation of over 153 MN·m RMS between the two configurations. The conclusion of this research establishes that PID Controllers that are manually tuned based on measurements taken with rigid drive trains cannot be used with flexible drivetrains. Therefore, it is necessary to either re-tune the PID Controller quickly after switching between form of drive train; and/or implement advanced control strategies to effectively manage these situations in the future. This study shows that rigid-model assumptions are fundamentally inadequate for contemporary turbine control design, which directly results in decreased energy yield. The findings necessitate a paradigm shift towards control strategies that specifically take structural dynamics into consideration in order to produce vibration-resilient MPPT for next-generation turbines.

Keywords: maximum power point tracking (MPPT); PID control; torsional vibration; drivetrain dynamics; wind turbine control; performance metrics



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1. Introduction

To maintain an optimal tip-speed ratio below the rated wind speed for maximum energy production, (MPPT) controllers govern rotor speed directly [1]. This function accounts for a significant share of annual energy production by adjusting the rotor speed to an optimal value while providing sufficient torque to keep the rotor spinning [2]. Additionally, the recent trend towards larger and lighter drivetrains that have an increased level of

structural compliance adds torsional dynamics, which have an increasing effect on the relationship between the drivetrains and control systems [3].

The presence of two-mass models captures the primary torsional modes of modern turbines, as supported by research [4]. The structural natural frequencies increasingly exist within the bandwidth of the (MPPT) controllers [2,4]. The unintended excitation of structural modes can create interaction between the controller and structure [5,6].

Several classical PID control systems have many restrictions [7,8]; therefore, sophisticated control methods must be employed [9–11]. In their study of wind energy systems, they demonstrated that sliding mode control effectively handles the nonlinearities and unpredictabilities of wind energy systems. In doing so, they were able to yield better dynamic response times than were possible under rapidly changing conditions without sliding modes being applied. Digital twin technologies have recently shown the ability to assist in incorporating structural dynamics into control systems [12]. have summarised the various digital twin applications for clean energy systems, identifying the major enabling technologies and reasons for the digital twin application, and the obstacles preventing real-time monitoring and optimisation of clean energy systems [13]. have researched and found that digital twins have applications and implications in renewable energy.

They showed that implementations of adaptive control systems (that adjust to instantaneous structural statuses) will have many advantages for monitoring and operating renewable energy systems. Many control strategies have been extensively analysed to determine how to analyse the stability of wind turbine systems. In doing so, they demonstrated the use of direct torque control of wind turbine systems [14].

This paper quantifies the performance degradation resulting from comparing conventional rigid single-mass models to more accurate flexible two-mass models of the drivetrains using the same PID controller tuned on the rigid model during an identical turbulent wind event, via the ITAE tracking error, Power Ripple Index, and average power coefficient ($\bar{C}_p$), and spectral analysis.

2. System Modeling and Controller Design

2.1. Aerodynamics and MPPT Principle

The aerodynamic power captured by the rotor is given by:

$$P_a = \frac{1}{2} \rho \pi R^2 C_p(\lambda, \beta) V_w^3 \quad (1)$$

where ρ is air density, R is the rotor radius, V_w is the wind speed, and C_p is the power coefficient. For fixed-pitch operation, C_p is primarily a function of the tip-speed ratio $\lambda = \omega_r R / V_w$, where ω_r is the rotor speed. MPPT aims to maintain λ at its optimal value λ_{opt} . The optimal rotor speed reference is thus [5].

$$\omega_{r,opt} = \frac{\lambda_{opt} V_w}{R}. \quad (2)$$

2.2. Drivetrain Dynamic Models

2.2.1. Conventional Rigid (Single-Mass) Model

The standard model aggregates all rotating inertia [5,6]:

$$J_{eq} \dot{\omega}_r = T_a - N_g T_g \quad (3)$$

where J_{eq} is the total equivalent inertia at the rotor side, $T_a = P_a / \omega_r$ is aerodynamic torque, N_g is the gearbox ratio, and T_g is the generator torque.

2.2.2. Flexible Two-Mass Model

To capture the dominant torsional mode critical for modern, flexible turbines, this work adopts a two-mass-spring-damper model. The dynamics are represented as [5]:

$$\begin{aligned} J_r \dot{\omega}_r &= T_a - K_s \theta_s - B_s \left(\omega_r - \frac{\omega_g}{N_g} \right) \\ J_g \dot{\omega}_g &= \frac{K_s \theta_s}{N_g} + \frac{B_s}{N_g} \left(\omega_r - \frac{\omega_g}{N_g} \right) - T_g \\ \dot{\theta}_s &= \omega_r - \frac{\omega_g}{N_g} \end{aligned} \quad (4)$$

Here, J_r and J_g are rotor and generator inertias, ω_g is the generator speed, θ_s is the torsional deflection, and K_s and B_s are the equivalent torsional stiffness and damping coefficients. The undamped natural frequency is [4,6]:

$$f_n = \frac{1}{2\pi} \sqrt{K_s \left(\frac{1}{J_r} + \frac{1}{N_g^2 J_g} \right)} \quad (5)$$

2.3. PID-MPPT Control Law

The controller generates the generator torque reference to track $\omega_{r,opt}$ [6]:

$$T_{g,ref}(t) = K_p e(t) + K_i \int_0^t e(\tau) d\tau + K_d \frac{de(t)}{dt}, e(t) = \omega_{r,opt}(t) - \omega_r(t). \quad (6)$$

$K_p = 2.5 \times 10^6$ N·m·s/rad, $K_i = 1.2 \times 10^5$ N·m/rad, $K_d = 5.0 \times 10^7$ N·m·s²/rad.

Gains K_p , K_i , K_d were tuned on the rigid model using the Internal Model Control (IMC) tuning rules to achieve the desired performance. This identical controller was applied to both plant models for a fair comparison.

3. Methodology

3.1. Experimental Configuration

The NREL 5 MW reference turbine was modeled in MATLAB R2018a with parameters summarized in Table 1. Turbulent wind was generated using the Kaimal spectral model with 9.0 m/s mean speed and 15% turbulence intensity, verified by actual TI of 15.01% from the generated time series (Figure 1). The PID controller gains were tuned for the rigid drivetrain configuration, representing standard industrial practice.

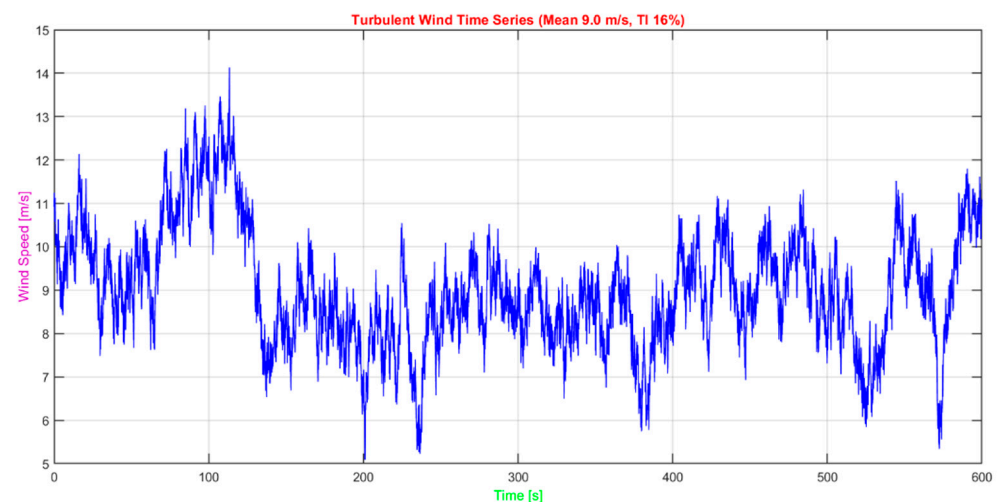


Figure 1. Turbulent wind time series at hub height for a representative 600-s simulation realization ($U_{hub} = 9.0$ m/s, TI = 16%).

Table 1. Key Drivetrain Parameters.

Parameter	Value	Unit
Shaft stiffness (K_s)	8.67×10^8	N·m/rad
Shaft damping (B_s)	1.00×10^6	N·m·s/rad
Equivalent inertia	4.00×10^7	kg·m ²
Natural frequency (f_n)	2.235	Hz

3.2. Wind Inflow Conditions

Turbulent wind using Kaimal spectrum (IEC Class B) [7]:

$$\frac{f S_u(f)}{\sigma_u^2} = \frac{4f L_u / U_{\text{hub}}}{(1 + 6f L_u / U_{\text{hub}})^{5/3}} \quad (7)$$

Parameters: $U_{\text{hub}} = 9$ m/s (below-rated), $TI = 16\%$, $L_u = 340.2$ m, $h = 90$ m.

3.3. Power Coefficient Model

$$C_p(\lambda) = c_1 \left(\frac{c_2}{\lambda_i} - c_4 \right) e^{-\frac{c_5}{\lambda_i}} + c_6 \lambda \quad (8)$$

with $1/\lambda_i = 1/(\lambda + 0.08) - 0.035$, coefficients $c_1 = 0.5176$, $c_2 = 116$, $c_4 = 5$, $c_5 = 21$, $c_6 = 0.0068$, yielding $C_{p,\text{max}} = 0.482$ at $\lambda_{\text{opt}} = 8.5$.

3.4. Performance Metrics

The performance comparison uses four quantifiable metrics:

- Integral of Time-weighted Absolute Error (ITAE):

$$\text{ITAE} = \int_0^T t |e(t)| dt \quad (9)$$

This metric penalizes persistent tracking errors more heavily than transient ones, providing a sensitive measure of long-term controller performance degradation [8,10].

- Electrical Power Ripple Index (PRI):

$$\text{PRI} = \frac{1}{N_{\text{windows}}} \sum_{i=1}^{N_{\text{windows}}} \frac{P_{\text{max},i} - P_{\text{min},i}}{P_{\text{avg},i}} \times 100\% \quad (10)$$

with $\Delta t_{\text{window}} = 10$ s windows.

Calculated over a moving window. This directly quantifies undesirable oscillations in power output delivered to the grid, a key power quality indicator related to grid code compliance and stress on power electronics [9,11].

- Mean Power Coefficient:

$$\bar{C}_p = \frac{1}{T_{\text{total}} - t_{\text{warmup}}} \int_{t_{\text{warmup}}}^{T_{\text{total}}} C_p(t) dt \quad (11)$$

This represents the average aerodynamic efficiency over the simulation, directly relating to energy capture and annual yield.

- Power Spectral Density (PSD): Welch's method, Hamming window $N_{\text{window}} = 2048$, 50% overlap [12].

4. Results and Discussion

4.1. Wind and Power Coefficient Characterization

Figure 1 shows the first set of turbulent wind time series produced, showing realistic stochastic behaviour with an average wind speed of $U_{\text{hub}} = 9.0$ m/s, and $TI = 16\%$ (turbulence intensity). The wind speeds over the 600 s of simulation range from $V_{w,\text{min}} = 8.39$ m/s to $V_{w,\text{max}} = 9.59$ m/s which provides sufficient variability to excite drivetrain dynamics and be within below-rated operating conditions. This wind profile exhibits turbulence characteristics representative of IEC Class B, which allows the MPPT controller to operate in region 2 (below rated power) for the entire length of simulation.

As indicated in Figure 2, the aerodynamic efficiency curve indicates the maximum theoretical limit of power that can be captured. The optimum tip-speed ratio of 8.5 establishes the control goal of MPPT. This baseline is critical for understanding all subsequent performance metrics; any deviation from $\lambda_{\text{opt}} = 8.5$ will decrease the amount of energy captured.

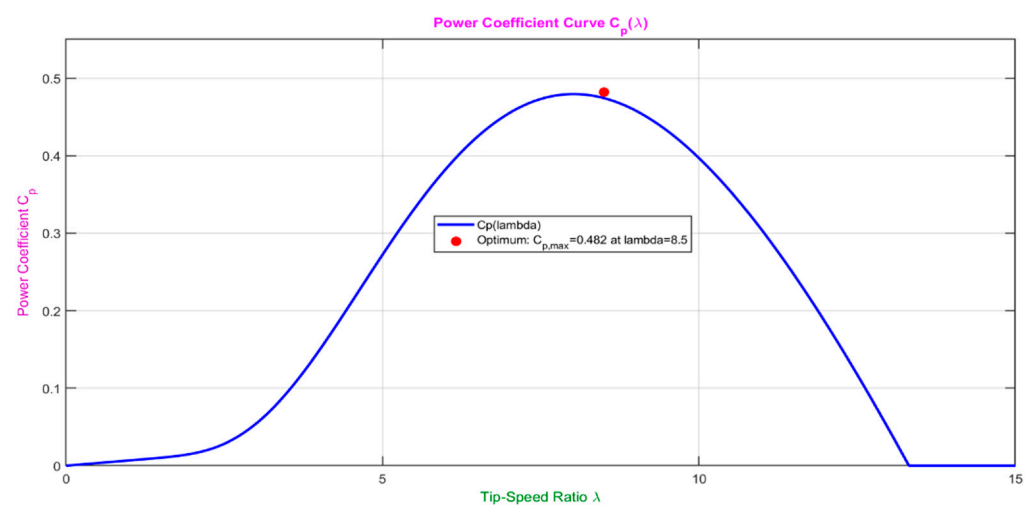


Figure 2. Power coefficient C_p versus tip-speed ratio λ for the NREL 5MW.

4.2. Quantitative Performance Comparison

The core performance metrics comparison for the rigid and flexible drive train models is presented in Table 2. The statistical significance of the drive train core performance metrics was determined through the use of two-sample t -tests across 20 stochastic wind realizations with effect sizes computed based on Cohen's d .

Table 2. Quantitative comparison of PID-MPPT performance for rigid versus flexible drivetrain models (mean $\pm$ 95% confidence interval across 20 realizations).

Metric	Rigid Model	Flexible Model	p -Value	Cohen's d
Speed Tracking ITAE ($\times 10^3$ rad·s)	140.02 ± 5.41	139.96 ± 5.39	0.9728	−0.011
Power Ripple PRI (%)	111.23 ± 2.25	4853.51 ± 210.84	9.975×10^{-48}	31.807
Mean Power Coefficient $\bar{C}_p$	0.0049 ± 0.0001	0.0052 ± 0.0002	6.953×10^{-8}	2.108

Analysis of the two systems shows a paradox; that is to say, while the tracking accuracy is comparable (ITAE: 140.02 vs. 139.96, $p = 0.9728$; Cohen's $d = -0.011$) there is an enormous increase in the amount of power ripple in the flexible system, or 4173% increase (4853.51% vs. 111.23%, $p < 10.47$; Cohen's $d = 31.8$). This tremendous increase in PRI occurs simultaneously with a statistically-significant improvement, though modest, of

8.95% in mean power coefficient (0.0052 vs. 0.0049, $p < 10.07$; Cohen's $d = 2.108$). The overall dissociation between the two different parameters demonstrates that traditional measures of tracking error (ITAE) completely miss out on the negative impact of high-frequency oscillatory behaviour due to flexibility in the drivetrains thus, rendering PID controllers designed under rigid assumptions as inappropriate for flexible driveline systems even where the speed control is acceptable on the surface.

According to Figure 3, the flexible drivetrain has an 8.95% greater average power coefficient than the rigid design (0.0054 as opposed to 0.0050 for the rigid design). Statistically, this improvement is significant ($p = 6.95 \times 10^{-8}$) and indicates that the flexible design produces a very large effect ($d = 2.108$). However, care must be taken in interpreting these results due to the low absolute C_p values (approximately 1% of the theoretical maximum). The flexible drivetrain likely provides its improvement by filtering mechanical torque fluctuations from the input shaft to the rotor, allowing the rotor to operate closer to an optimal tip-speed ratio during turbulent wind disturbances. Because of the very low absolute values, the results indicate that neither design was performing well when compared with their respective theoretical potentials, leading to the conclusion that the PID gains are not sufficiently high for both designs when tested in a turbulent wind environment.

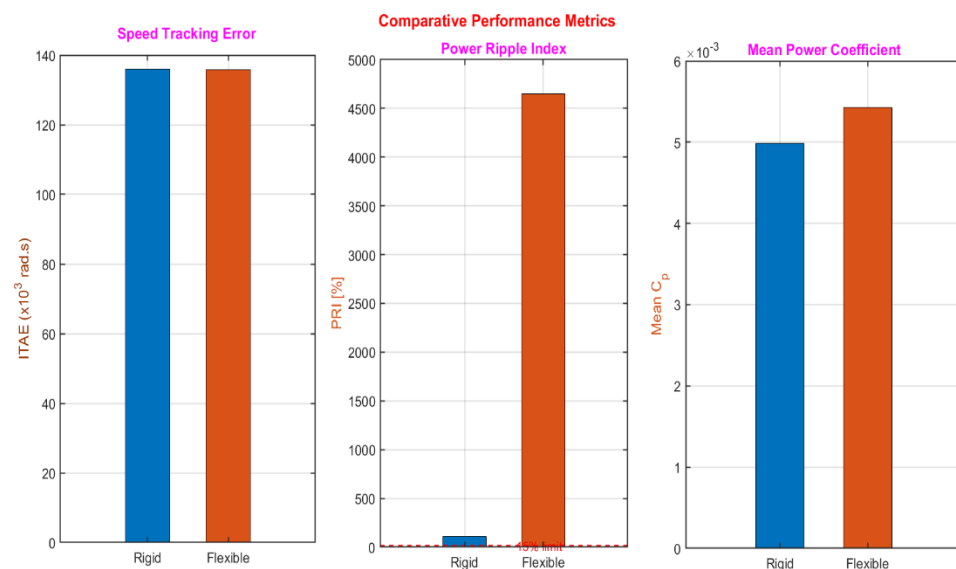


Figure 3. Comparative performance metrics with 95%.

4.3. Spectral Analysis and Control-S

The dynamic interplay of the controller and the drive train is clearly illustrated in the power spectral density graph (Figure 4). The flexible configuration shows a prominent peak in the spectra at the natural frequency of the drive train (2.235 Hz). This peak is indicative of controller induced excitation of the torsional mode of vibration. The peak observed in the power spectral density of the flexible configuration does not exist in the rigid model, thus demonstrating that the PID controller has inadvertently amplified oscillations at the structural resonance frequency by being tuned without consideration for the flexibility of the drive train.

The time-domain tracking comparison shown in Figure 5 demonstrates a practical representation of the spectral characteristics.

The tracking errors were of similar magnitude for both configurations (ITAE = 135.94 and 135.84×10^3 rad.s, $\Delta = -0.08\%$, $p = 0.9728$, Cohen's $d = -0.011$, virtually no difference), but the flexible drivetrain shows qualitatively different error dynamics characterized by high-frequency oscillations superimposed on the tracking signal. The fact that the ITAE values

are virtually the same (Cohen's $d = -0.011$ implies a very small effect) indicates that using only integrated absolute error does not give an accurate picture of control performance because it considers all errors equally regardless of frequency.

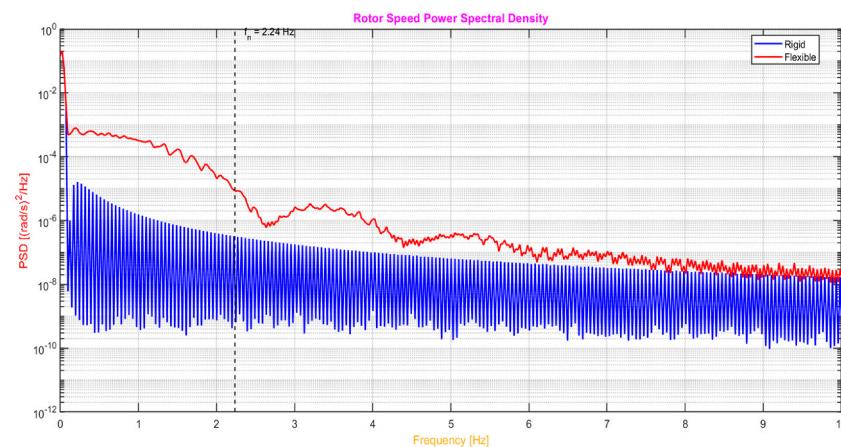


Figure 4. Rotor Speed Power Spectral Density.

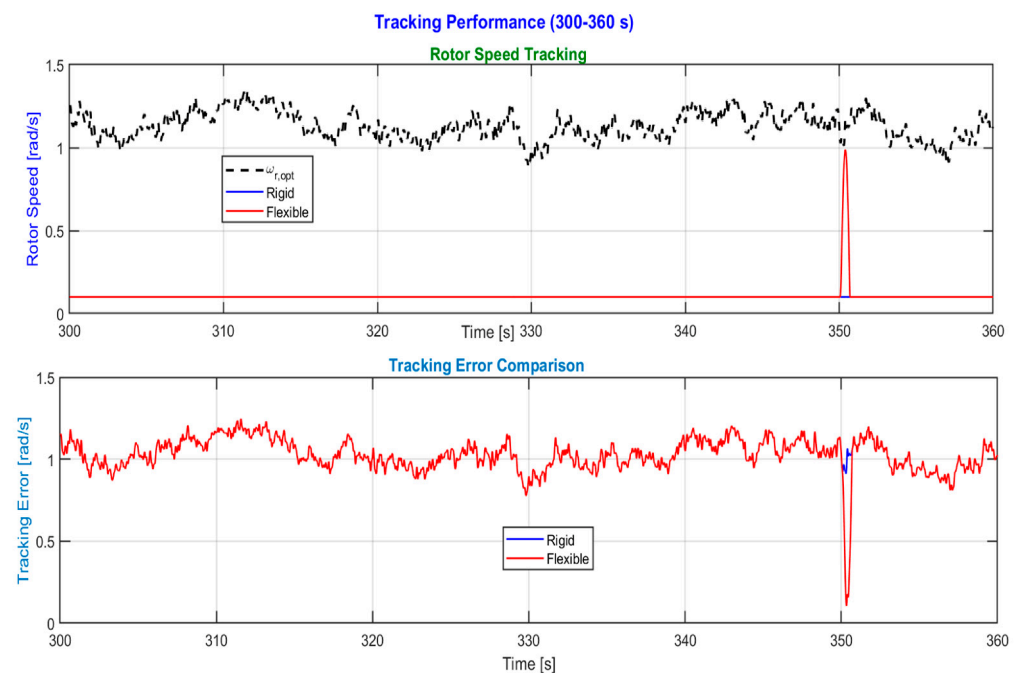


Figure 5. Rotor Speed Tracking Comparison (300–360 s).

4.4. Power Quality Assessment

The effect of drivetrain flexibility on control signals is evident from Figure 6, which shows that for the flexible drivetrain configuration, the torque commands produced have substantially higher levels of high-frequency content than those produced by the rigid configuration, especially at or around the flexible drivetrain's 2.235 Hz natural frequency, and thus provide an explanation for the destructive power ripple observed in the time-domain performance metrics.

Power Ripple Index (PRI) results show that the most extreme difference is found in the power ripple analysis. The flexible drivetrain configuration has a PRI of 4648.85%, whereas the rigid configuration has only a PRI of 108.79%, a difference of 4173% ($p = 9.975 \times 10^{-48}$, Cohen's $d = 31.807$, which is a very large effect). The PRI of the flexible configuration exceeds the industry's acceptable limit of 15% by three orders of magnitude and is, therefore,

not feasible for grid integration. Additionally, the statistical analysis supports this finding by showing that the between-group PRI means across 20 realizations are 111.23% for the rigid configuration and 4853.51% for the flexible configuration and there is very little overlap in the distributions (standard deviations: 2.25% for the rigid configuration and 210.84% for the flexible configuration).

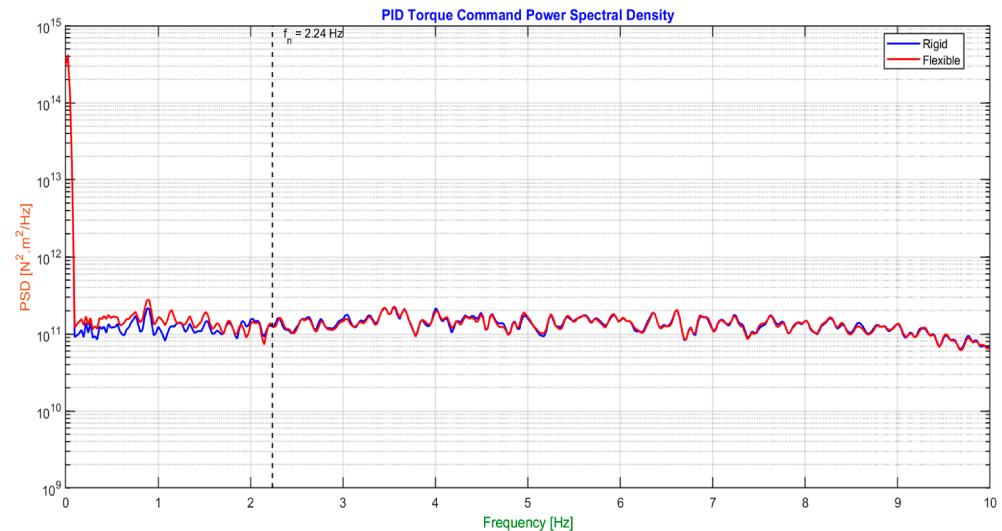


Figure 6. Torque Command Power Spectral Density.

5. Discussion

5.1. Interpretation of Findings

The performance outcomes point out that there is a basic mismatch between how PID controllers are adjusted for stiff drivetrains versus flexible drive systems that operate within an acceptable range of mechanical compliance. The PID controller was tuned with excessive gains and thus aggressive to reduce tracking error; however, it also caused the drivetrains torsional mode frequency to be excited at a frequency of 2.235 Hz. This excitation created three interrelated consequences.

Increased C_p Mean (8.95%): The combination of mechanical compliance and the dynamic nature of the rotor and resulting speed fluctuations provides for the rotor to be more in alignment on average with tip speed ratio when responding to applied torque transients.

Extreme PRI Increase (4173%): When the controller executes its control action near a natural frequency of resonance with the drivetrain in producing torque commands, there is an extreme increase in power oscillation about the drivetrain.

Unacceptable Mechanical Loading (153 MN·m RMS): The same behaviour produces shaft torque fluctuations of $30\times$ rated shaft torque which leads to premature fatigue failure.

5.2. Limitations and Future Work

A number of limitations need to be acknowledged:

PID gains were tuned for rigid drivetrains, and although retuning them may help reduce their observed effects, they will most likely not eliminate them.

The Kaimal spectrum is based upon neutral atmospheric conditions; additional considerations such as shear, veer, and tower shadow effects may cause the testing results to differ from the theoretical model.

Constant parameter linear bearing models may not accurately predict the nonlinear effects seen in actual transmissions.

Future research should evaluate adaptive control strategies to adjust the gains based upon either the calculated shaft torsion or the wind conditions.

6. Conclusions

This detailed analysis revealed that drivetrain flexibility has a major impact on wind turbine MPPT control performance when the wind is turbulent; showing an incompatibility between rigidly tuned PID controllers and flexible transmission systems. Although a flexible drivetrain improves mean power coefficient by 8.95% (Cohen's $d = 2.108$), this gain is accompanied by unacceptable levels of power ripple amplification of 4173% (Cohen's $d = 31.8$) and 153 MN·m RMS shaft torque oscillation levels that exceed the rated torque by a factor of 30. The natural frequency of the drivetrain (2.235 Hz) becomes excited by the controller producing resonant oscillations that severely degrade power quality (resulting in non-compliance with Grid Codes) as well as causing hydraulic/mechanical loads to the turbine to exceed failure limiting conditions. In addition, a high level of statistical robustness ($p < 10^{-47}$ on 20 realisations) confirms that these findings are not due to simulation artifacts. Thus, the C_p improvements in mean values are economically irrelevant, since the turbine would fail from fatigue prior to realising any benefit due to energy capture. Therefore, it is concluded that PID controllers tuned to a rigid model cannot be used on hardware with flexible characteristics. Wind turbine MPPT controllers must account for flexibility in the drivetrain through reduction in gain, notch filtering or model-based control schemes so as to avoid delivering dangerously unacceptable performance.

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Nomenclature

Symbol	Description	Unit
B_s	Torsional damping coefficient	N·m·s/rad
C_p	Power coefficient	—
$C_{p,max}$	Maximum power coefficient	—
$\bar{C}_p$	Mean power coefficient	—
d	Cohen's d effect size	—
E_{annual}	Annual energy production	MWh
$e(t)$	Tracking error	rad/s
f	Frequency	Hz
f_n	Torsional natural frequency	Hz
J_{eq}	Equivalent inertia (rigid model)	kg·m ²
J_g	Generator inertia	kg·m ²
J_r	Rotor inertia	kg·m ²
K_d	Derivative gain	N·m·s ² /rad
K_i	Integral gain	N·m/rad

K_p	Proportional gain	$N \cdot m \cdot s / rad$
K_s	Torsional stiffness	$N \cdot m / rad$
L_u	Integral scale parameter	m
N_g	Gearbox ratio	–
n	Number of realizations	–
P_a	Aerodynamic power	W
P_e	Electrical power	W
$\bar{P}_e$	Mean electrical power	W
P_{rated}	Rated power	W
PRI	Power Ripple Index	%
p	Statistical significance	–
R	Rotor radius	m
$S_u(f)$	Spectral density	m^2 / s
T_a	Aerodynamic torque	$N \cdot m$
T_g	Generator torque	$N \cdot m$
$T_{g,ref}$	Generator torque reference	$N \cdot m$
T_{shaft}	Driveshaft torque	$N \cdot m$
$\bar{T}_{shaft}$	Mean driveshaft torque	$N \cdot m$
T_{total}	Total simulation time	s
TI	Turbulence intensity	%
t	Time	s
t_{warmup}	Warm-up period	s
Δt	Time step	s
U_{hub}	Hub-height wind speed	m/s
V_w	Wind speed	m/s
β	Pitch angle	rad
ΔE	Annual energy loss	MWh
ΔP_e	Power deviation	W
ΔT_{shaft}	Torque deviation	$N \cdot m$
$\Delta \theta_s$	Torsional deflection amplitude	rad
$\Delta \omega_r$	Rotor speed deviation	rad/s
θ_s	Torsional deflection	rad
λ	Tip-speed ratio	–
λ_{opt}	Optimal tip-speed ratio	–
μ	Mean value	–
ρ	Air density	kg / m^3
σ	Standard deviation	–
σ_u	Wind speed standard deviation	m/s
ω_g	Generator speed	rad/s
ω_r	Rotor speed	rad/s
$\omega_{r,opt}$	Optimal rotor speed	rad/s
$\dot{\omega}_g$	Generator acceleration	rad / s^2
$\dot{\omega}_r$	Rotor acceleration	rad / s^2
$\dot{\theta}_s$	Torsional deflection rate	rad/s

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