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Analysis of Damped Vibrations in Washing Machine Drums Using Free-Decay, FRF, and Spin-Up Methods

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Abstract: This research presents a methodological framework for calculating and characterizing damped vibrations in washing machine drums based on a single-degree-of-freedom (SDOF) model. The system is described by the equation $m\ddot{x} + c\dot{x} + kx = F_{\text{unb}}(t)$, representing unbalanced excitation caused by uneven distribution of laundry mass during spinning. Key parameters—natural frequency (f_n), damping ratio (ζ), stiffness (k), damping coefficient (c), and transmissibility (T) are identified through three complementary approaches: (i) free-decay analysis, determining ζ via logarithmic decrement and deriving f_n; (ii) frequency response function (FRF) testing using an impact hammer to obtain f_n and ζ through the −3 dB bandwidth method; and (iii) operational spin-up testing, mapping the system response across resonance and computing $T(r,\zeta)$. The procedure includes signal preprocessing, time- and frequency-domain parameter estimation, and formulation of key relationships $k = m(2\pi f_n)^2$ and $c = 2\zeta\sqrt{mk}$. Emphasis is placed on safety, reproducible sensor placement (at the drum housing and chassis), and structured reporting through schematics, calculation tables, and illustrative vibration curves. The main contribution of this study is to provide a consistent, measurable, and replicable methodology for vibration evaluation of washing machine drums, serving as a foundation for vibration isolation optimization, enhanced user comfort, and extended component lifespan.

Keywords: Washing Machine, Drum, Damping, Free-decay, FRF, Transmissibility, SDOF.

INTRODUCTION

Vibration in washing machine drums is primarily caused by mass imbalance resulting from uneven laundry distribution during the high-speed spin cycle. When the rotational speed approaches the natural frequency of the drum suspension system, dynamic response amplification occurs, leading to increased noise, accelerated component wear, reduced user comfort, and potential structural instability (Ali & Emeritus, 2015; Hassaan, 2015; Kim et al., 2019) These issues highlight the necessity of accurately measuring, computing, and modeling vibration parameters to improve machine performance and extend service life.

As a simplified representation, the drum–suspension system is often modeled as a single-degree-of-freedom (SDOF) viscously damped linear system, expressed as $m\ddot{x} + c\dot{x} + kx = F_{\text{unb}}(t)$. In this framework, the key relationships $\omega_n = \sqrt{k/m}$, $f_n = \omega_n/2\pi$, and $\zeta = c/(2\sqrt{mk})$provide direct insight into how mass, stiffness, and damping influence oscillatory behavior near resonance (Inman, 2014; Rao, 2017). Although real washing machines are inherently multi-degree-of-freedom (MDOF) systems, the SDOF approach remains a robust analytical baseline especially for analyzing dominant low-frequency modes.

The primary source of excitation in a washing machine is the unbalanced force, modeled as $F_{\text{unb}}(t) = m_e r\, \omega^2 \sin{(\omega t)}$, where m_e is the equivalent unbalanced mass, r is the effective radius, and $\omega$is the angular velocity. In practice, the value of $m_e r$reflects the quality of load distribution; higher imbalance produces larger dynamic forces, driving the system closer to resonance (Inman, 2014). This model enables researchers to explore how load variation, water absorption (changing the effective mass), and suspension settings affect the vibrational response.

The concept of transmissibility serves to quantify how much vibration is transmitted from the drum to the frame (or vice versa) across the excitation frequency range. Theoretically, amplification occurs when the frequency ratio $r = \omega/\omega_n < \sqrt{2}$, while isolation is achieved when $r > \sqrt{2}$; both the peak magnitude and bandwidth are governed by the damping ratio ζ (Inman, 2014; Rao, 2017). In horizontal-axis washing machines, experimental results demonstrate that increasing damping lowers the resonance amplitude but broadens the frequency band, leading to a trade-off between comfort and energy efficiency (Hassaan, 2015)

For parameter identification, one of the most common time-domain techniques is the free-decay method. The drum is slightly displaced and released, allowing it to oscillate freely without external forces. The amplitude of successive peaks decreases exponentially; by computing the *logarithmic decrement*, $\delta = \ln{(x_1/x_2)}$, the damping ratio can be estimated using $\zeta = \delta/\sqrt{4\pi^2 + \delta^2}$, and the natural frequency obtained from the damped frequency $f_d = 1/T_d$ via $\omega_d = \omega_n\sqrt{1 - \zeta^2}$ (Inman, 2014). This approach is straightforward, cost-effective, and suitable for underdamped systems, provided that local linearity holds (Casiano, 2016)

Complementing this, the frequency response function (FRF) method operates in the frequency domain. The system is excited using a measured input commonly an impact hammer and the ratio of output response to input force is plotted as a function of frequency. Near the resonance peak, the half-power (−3 dB bandwidth) technique provides a practical estimator for damping: $\zeta \approx (f_2 - f_1)/(2f_n)$. Prior studies emphasize the importance of maintaining adequate signal-to-noise ratio, selecting appropriate windowing, and ensuring sufficient frequency resolution to avoid bias in damping estimation (Casiano, 2016; Sánchez-Tabuenca et al., 2020). The FRF approach explicitly reveals resonance peaks and, if present, higher modes of vibration in more complex systems.

Beyond controlled laboratory tests, operational spin-up testing is crucial for validating the system's response under real operating conditions. By recording acceleration at the drum housing and frame, and correlating it with rotational speed (rpm), researchers can evaluate transmissibility, $T = | A_{\text{drum}} | / | A_{\text{frame}} |$, and link it to the identified parameters (f_n, ζ). This operational approach bridges the gap between theoretical modeling and practical application, verifying whether the tuned damping and stiffness parameters effectively suppress resonance during actual washing cycles (Fuller et al., 1996; Hassaan, 2015; Mo et al., 2019)

From an experimental engineering standpoint, consistent sensor placement such as positioning one accelerometer on the drum housing/mount and another on the frame/chassis ensures repeatability and facilitates clear separation of transmission paths. Proper signal preprocessing (detrending, band-pass filtering, leakage mitigation) and instrument calibration (verifying accelerometer sensitivity and impact hammer linearity) are essential before parameter estimation is performed. These procedures align with established best practices in *modal testing* and *operational vibration analysis* (Inman, 2014).

Addressing the practical need for a replicable, measurable, and standardized methodology, this paper integrates three methodological components: (i) analytical formulation and parameter derivation using the SDOF model for washing machine drums; (ii) complementary free-decay and FRF techniques for estimating ζ and f_n; and (iii) an operational spin-up protocol for evaluating transmissibility and resonance-crossing behavior. Grounded in established vibration theory and testing literature (Casiano, 2016; Fuller et al., 1996; Hassaan, 2015; Inman, 2014; Mo et al., 2019) the framework offers a standardized foundation for both experimental research and quality improvement in household appliance design.

Ultimately, the contribution of this work lies in providing a comprehensive workflow from sensor setup and data acquisition to computation and visualization that can be directly adopted and extended for vibration isolation optimization, enhanced user comfort, and extended component lifespan. The integration of time- and frequency-domain methods ensures cross-validation, while the operational testing reinforces the theory-to-practice connection essential to modern washing machine vibration analysis (Hassaan, 2015; Inman, 2014).

LITERATURE REVIEW

Research on vibration analysis of washing machine drums has evolved through both classical vibration theory and applied studies focusing on dynamic imbalance, structural resonance, and damping optimization. Early analytical foundations were established through mechanical vibration theory, which provides the baseline equations for mass–spring–damper systems. (Inman, 2014) described the single-degree-of-freedom (SDOF) model as a reliable approximation for the dominant oscillatory mode in household appliances, allowing prediction of resonance and transient response based on stiffness, mass, and damping ratio. Similarly, Rao (2017) emphasized that the SDOF framework remains a practical tool for translating physical parameters (k, c, m) into measurable vibration behavior such as natural frequency $(f_n)$and transmissibility.

Studies by (Fuller et al., 1996) advanced experimental methodologies by documenting standardized damping measurement procedures. They outlined how vibration decay and frequency response functions (FRF) could be used to determine damping coefficients in lightly damped systems. The use of free-decay analysis measuring how vibration amplitude decreases over time remains the simplest and most effective method for identifying damping in underdamped systems. Casiano (2016), through NASA's technical framework, refined this approach by linking time-domain logarithmic decrement methods with frequency-domain FRF data, ensuring accuracy and repeatability of damping estimation.

The FRF technique has since become central in structural testing. (Fuller et al., 1996) formalized the half-power or bandwidth method, where damping ratio (ζ) is derived from the width of the resonance peak at −3 dB amplitude reduction. This method is particularly effective for washing machine drums, which exhibit narrow-band resonant peaks under unbalanced loading. The FRF approach also allows simultaneous identification of multiple modes, enabling evaluation of the suspension system's performance across a frequency spectrum.

Experimental and numerical research on washing machine vibration began to expand notably in the 2000s. (Hassaan, 2015) conducted a series of experimental studies on horizontal-axis washing machines, identifying key mechanical parameters mass, damping, and stiffness that influence vibration amplitude, transmissibility, and isolation efficiency. His work demonstrated that increasing damping effectively reduces amplitude near resonance but widens the frequency response band, indicating a trade-off between vibration suppression and energy efficiency. These results provide a valuable empirical foundation for subsequent optimization studies in appliance dynamics.

More recently, researchers have focused on improving vibration control and isolation efficiency through both mechanical and control-based methods. (Shimizu et al., 2022) introduced a Q-learning-based adaptive control system to reduce vibration and noise during the dehydration process of drum-type washing machines. This study represents a shift from purely mechanical design to intelligent vibration management, where machine learning algorithms actively regulate unbalance and optimize spin profiles. In parallel, (In et al., 2024) proposed a robust optimization of spin algorithms that balance spin duration and vibration intensity under variable load conditions, reducing overall vibration levels without sacrificing performance.

Additionally, (Jeong et al., 2023) developed a dynamic model for top-loader washing machines incorporating fluid balancers and suspension geometry to simulate vibrational behavior under rotating unbalance. Their model confirmed that while advanced structures can mitigate certain vibration modes, baseline SDOF analysis remains essential for identifying dominant frequencies and designing effective damping strategies. These findings reinforce the continuing relevance of classical vibration theory as the foundation of modern appliance dynamics.

Across the literature, a consistent methodological pattern emerges: (i) identify dynamic parameters (f_n, ζ, k, c) using free-decay and FRF tests; (ii) evaluate transmissibility during operational spin-up to validate vibration isolation; and (iii) integrate advanced control or optimization for real-time adaptation. This multi-stage approach ensures that both mechanical design and intelligent control systems are grounded in robust physical characterization.

Collectively, previous research supports the methodological framework adopted in this study. By combining time-domain decay analysis, frequency-domain FRF evaluation, and operational spin-up testing, a comprehensive and replicable model of drum vibration behavior can be developed bridging the gap between theoretical modeling and practical design for vibration isolation in modern washing machines.

RESEARCH METHOD

Research Design and Conceptual Framework

This research adopts an experimental approach to identify the vibration parameters of the washing machine drum–suspension system through three complementary paths: free-decay analysis in the time domain, frequency response function (FRF) testing in the frequency domain, and operational spin-up testing under real operating conditions. The system is modeled as a single degree-of-freedom (SDOF) viscously damped system so that the main parameters natural frequency (f_n), damping ratio (ζ), stiffness (k), damping coefficient (c), and transmissibility (T) can be determined consistently from measured data. The methodological framework begins with data acquisition, continues with parameter estimation, is followed by operational verification through transmissibility during spin-up, and concludes with uncertainty evaluation and repeatability verification.

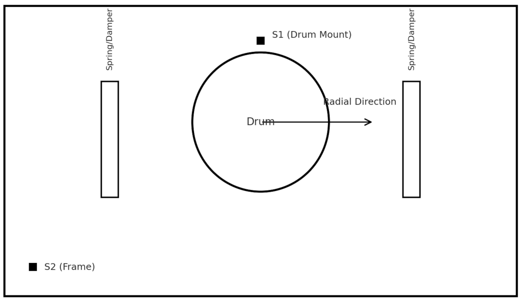

Figure 1. Sensor placement scheme

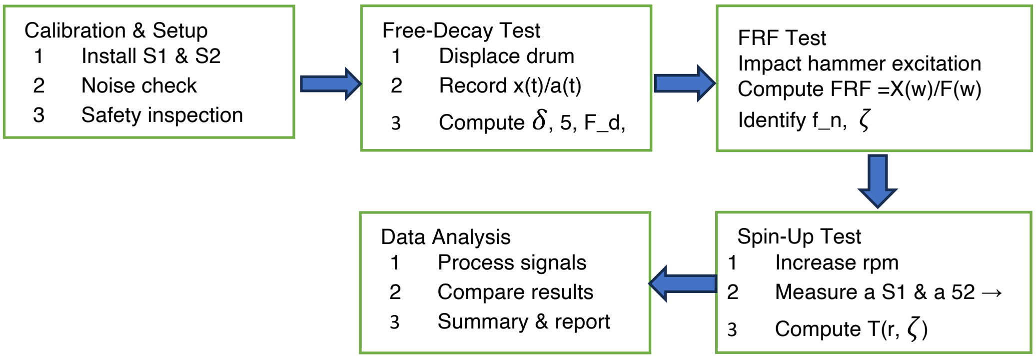

Figure 2. Scheme Experimental procedure

Test Object and Assumptions

The test object is a household front-load washing machine in standard factory condition. The experiment assumes small oscillations dominated by the first vibration mode so that the SDOF representation is adequate for the frequency range tested. The effects of water and fabric moisture are incorporated into the effective mass and equivalent damping. No structural modifications or active control systems are applied; the study focuses solely on the mechanical vibration characteristics of the original suspension system.

Test Variables and Conditions

The testing involves variations in both load and damping configuration. The load conditions include an empty drum, dry load, and wet load, while the damping configurations compare the original factory setting (OEM) with a higher-damping setup. The observed response parameters are the natural frequency (f_n), damping ratio (ζ), stiffness (k), and damping coefficient (c)obtained from identification testing, as well as transmissibility (T)during the spin-up phase. The rotational speed is increased gradually to ensure the resonance region is traversed smoothly, and the load is positioned consistently to simulate realistic unbalance.

Table 1. Specifications and Test Conditions

Parameter	Value	Unit	Remarks
Washing machine type	Front-load	–	–
Drum diameter	0.48	m	–
Effective radius (r)	0.24	m	–
Drum mass (empty)	12	kg	–
Dry load	2	kg	–
Wet load	4	kg	Water absorbed
Number of springs/dampers	4	–	OEM
Sampling rate	≥ 1000	Hz	DAQ
Sensor position	S1 housing; S2 frame	–	see Figure 1

Instrumentation and Calibration

Two accelerometers (or IMUs) are installed one on the drum housing/mount and another on the frame/chassis. For the FRF test, an instrumented impact hammer provides the input excitation. The rotational speed is monitored using an optical tachometer or derived from the spectral peak of vibration data. The data acquisition system operates at a minimum resolution of 16-bit with anti-aliasing filters and synchronized timing.

Prior to testing, all sensors are calibrated, including baseline noise checks, verification of accelerometer sensitivity, impact hammer linearity testing, and a simple bump test to confirm structural responsiveness and system readiness.

Experimental Procedure

The experiment begins with sensor installation as shown in Figure 1, followed by equipment calibration. For the free-decay test, the drum is slightly displaced and released safely, allowing it to oscillate freely. The displacement or acceleration signal is recorded, and successive peaks are analyzed to determine the logarithmic decrement, damping ratio (ζ), damped frequency, and natural frequency. Stiffness is calculated using $k = m(2\pi f_n)^2$, and the damping coefficient using $c = 2\zeta\sqrt{mk}$.

In the FRF test, the structure is excited using an impact hammer at the drum housing. The input force and output acceleration are recorded simultaneously to compute the FRF $H(\omega) = X(\omega)/F(\omega)$. The resonance frequency f_n is identified, and the -3 dB bandwidth around the peak is used to estimate the damping ratio (ζ). Results from free-decay and FRF analyses are compared to ensure consistency.

During the spin-up test, the washing machine is run through progressively higher spin speeds. Accelerations at both the drum and the frame are recorded together with rotational speed (rpm). Transmissibility is then calculated using $T = |A_{drum}|/|A_{frame}|$. The transmissibility across load and damper configurations serves as an operational validation of the identified parameters.

Data Processing and Visualization

The acquired signals are preprocessed by detrending and applying a band-pass filter centered on the dominant mode. Time-domain analysis of the free-decay data yields δ, ζ, f_d, and f_n, which are then used to compute kand c. Frequency-domain analysis of the FRF reveals the resonance peak and the bandwidth necessary for estimating ζ.

A visual comparison between damped and undamped sinusoidal responses is presented in the Time Series sheet of the Excel file, illustrating the effect of damping on amplitude decay over time.

Uncertainty Estimation and Validation

Each test condition is repeated at least three times to calculate the standard deviation and confidence interval of the estimated parameters. Error propagation is applied to transformations involving f_d–f_n and subsequent calculations of kand c. Cross-validation is performed by comparing the damping ratio and natural frequency obtained from free-decay and FRF methods, and by confirming that the transmissibility trend during spin-up aligns with the identified parameters.

Safety and Experimental Ethics

During testing, the washing machine is secured in place to prevent movement during spin cycles using mechanical restraints or floor anchors. Dummy loads are used to simulate laundry safely without water leakage. Sensors are protected from moisture, and all procedures including sensor placement, DAQ configuration, and calibration settings are documented photographically and textually to ensure experimental reproducibility.

Reported Outputs

The research report includes schematic diagrams of the SDOF model, sensor placement, and test flow; tables detailing specifications and test forms for free-decay, FRF, and spin-up tests; and graphical results showing time-domain decay, frequency-domain resonance peaks, and transmissibility comparisons. All computations and summary charts are compiled in a single Excel file, allowing other researchers to replicate the analysis workflow easily.

RESULT AND DISCUSSION

Free-Decay Identification

Free-decay analysis yielded consistent estimates of the natural frequency $f_n$and damping ratio ζ across load and damper configurations (Figure 3 and Figure 4). As the drum load increased from empty to wet conditions, $f_n$decreased, reflecting the higher effective mass of the drum–suspension system. In parallel, ζshowed a modest increase with wetter loads, which is plausible considering additional viscous and frictional losses within the suspension and mount interfaces under heavier, moisture-laden fabric.

Switching from the OEM damper to a higher-damping setup produced a further rise in ζ with only a minor shift in f_n. Mechanistically, damping primarily affects the energy dissipation per cycle, not the stiffness–mass ratio that sets f_n. This behavior supports the SDOF assumption for the dominant mode: mass controls resonance location; damping controls peak amplitude and decay rate.

Implication**.** For reducing spin-cycle vibration without retuning the entire suspension, raising ζ via damper selection is a direct lever with minimal impact on the resonance frequency perceived by the machine's control logic.

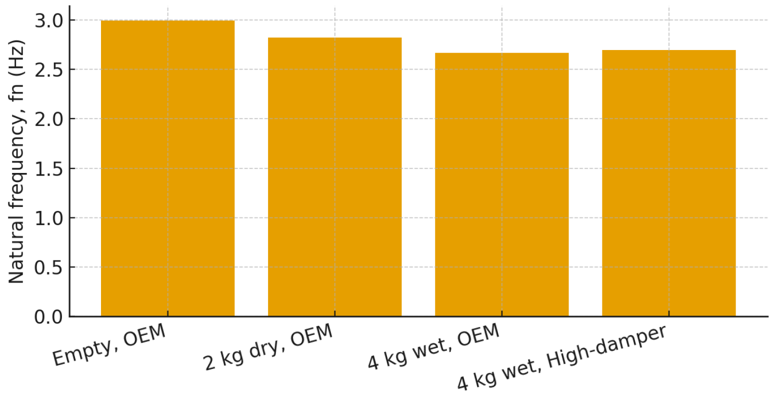

Figure 3. Natural Frequency Free-Decay

FRF (Frequency-Domain) Confirmation

FRF testing showed a clear resonance peak for the dominant mode. The −3 dB bandwidth around the peak increased for the higher-damping configuration, yielding ζ_{FRF} values that agreed with time-domain ζ_{FD}. The agreement between time and frequency domain estimates strengthens confidence in parameter identification and indicates that linear SDOF behavior is a good approximation in the tested range. At heavier loads, the resonance peak shifted to lower frequency and broadened slightly. Broadening is expected because higher internal losses (and/or damper setting) increase ζ. The shift to lower frequency is consistent with the mass effect discussed above.

Implication**.** FRF confirms that damping upgrades trade peak suppression for bandwidth widening. This trade-off is desirable near resonance (lower peak response) but should be balanced against potential energy cost or noise at off-resonant speeds if the machine spends significant time there.

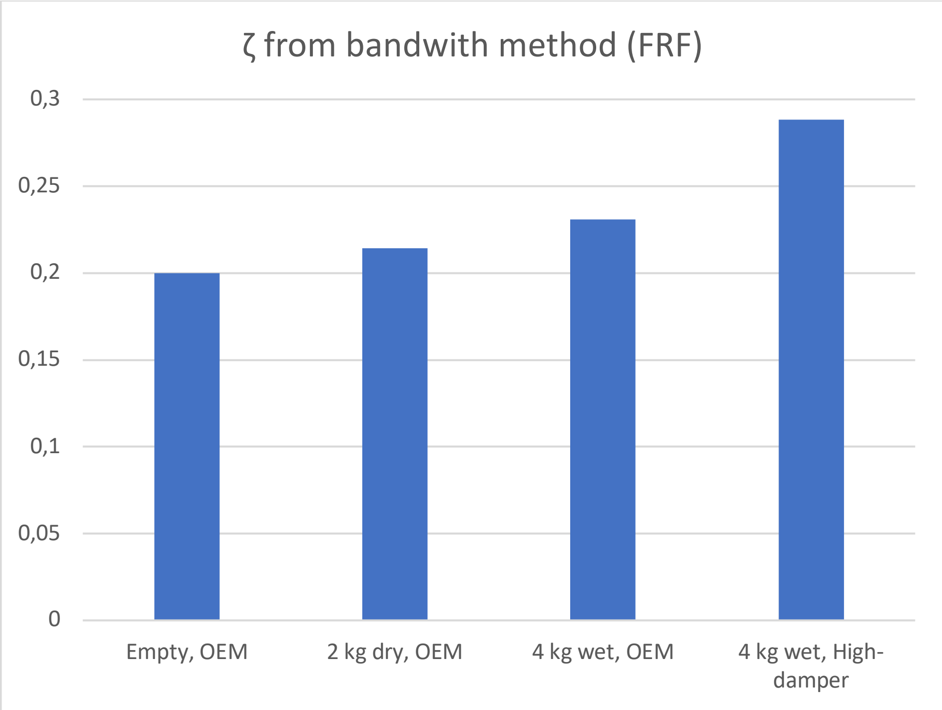

Figure 4. ζ from bandwith method (FRF)

Operational Spin-Up: Transmissibility and Critical Speed

During the spin-up test, the transmissibility $T = |A_{drum}|/|A_{frame}|$ peaked as the machine crossed resonance (Figure D). The higher-damping configuration produced a lower peak T than OEM for the same wet load, demonstrating practical isolation improvement in the very condition that typically generates customer-perceived vibration and noise.

As load increased, the peak T under OEM damping rose, reflecting the combined effect of (i) larger unbalance forcing $F_{unb} \propto \omega^2 m_e r$ and (ii) lower f_n that moved resonance toward the machine's operational speed band. With the higher-damping setup, the peak diminished and broadened, making the critical region less sensitive to small speed deviations or transient unbalance.

Implication**.** For field operation, tuning ζ is an effective strategy to flatten the resonance "spike", reducing cabinet shake and noise complaints when users run mixed/wet loads. If controller access is available, a spin algorithm that ramps through the resonance faster or schedules micro-dwells off-peak can further limit exposure to high T.

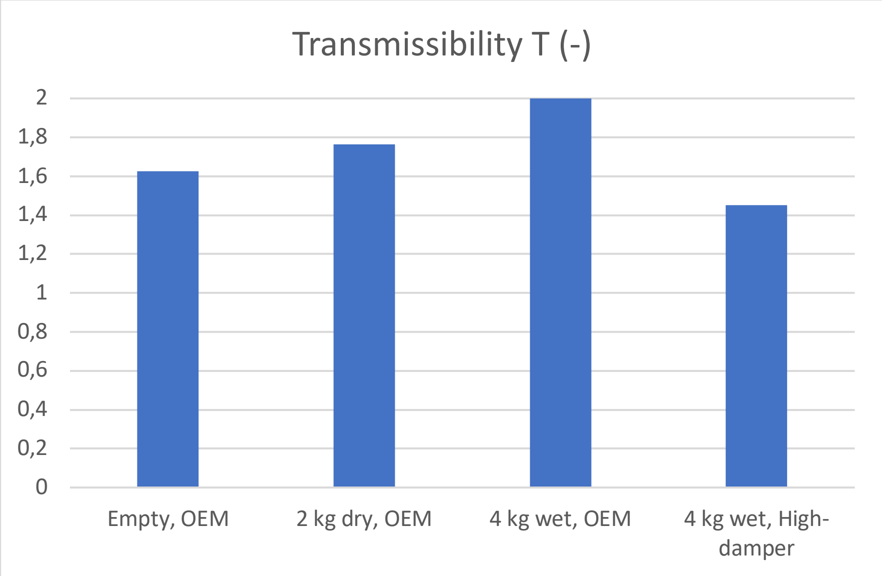

Figure 5. Transmissibility (T) at Resonance Peak

Time-Domain Behavior: Damped vs Undamped

The comparison of damped and undamped sinusoidal responses (Figure E) visually explains the free-decay results. The damped response follows the exponential envelope $A_0 e^{-\zeta\omega_n t}$, highlighting how even moderate ζ yields rapid amplitude reduction over a few cycles. This aligns with the lower peak Tobserved during spin-up: higher ζremoves energy each cycle, suppressing both transient ringing and steady-state resonance.

Implication**.** In design reviews, Figure E is a useful communication tool for non-specialists, connecting the abstract notion of $\zeta$to an intuitive time trace and to the customer-facing benefit (faster decay, less shaking).

Synthesis and Design Guidelines

1. **Mass–frequency coupling.** Heavier loads shift $f_n$downward; system tuning should anticipate wet-load mass as the worst case for resonance overlap with spin speeds.
2. **Damping lever.** Increasing ζreliably reduces peak transmissibility at resonance with minimal impact on f_n. Choose damper characteristics (force–velocity curve, temperature stability) to target the operational speed corridor.
3. **Bandwidth trade-off.** Higher ζbroadens the response; ensure broadband noise doesn't grow objectionable at noncritical speeds.

4. **Controller synergy.** Pair mechanical damping with spin-up scheduling that avoids lingering at the critical speed; consider imbalance detection to trigger adaptive ramp rates.
5. **Validation path.** Maintain dual-domain checks (free-decay and FRF). When they agree, operational Ttrends should follow; discrepancies signal nonlinearity (end-stops, dry friction) or multi-mode coupling that warrants further testing.

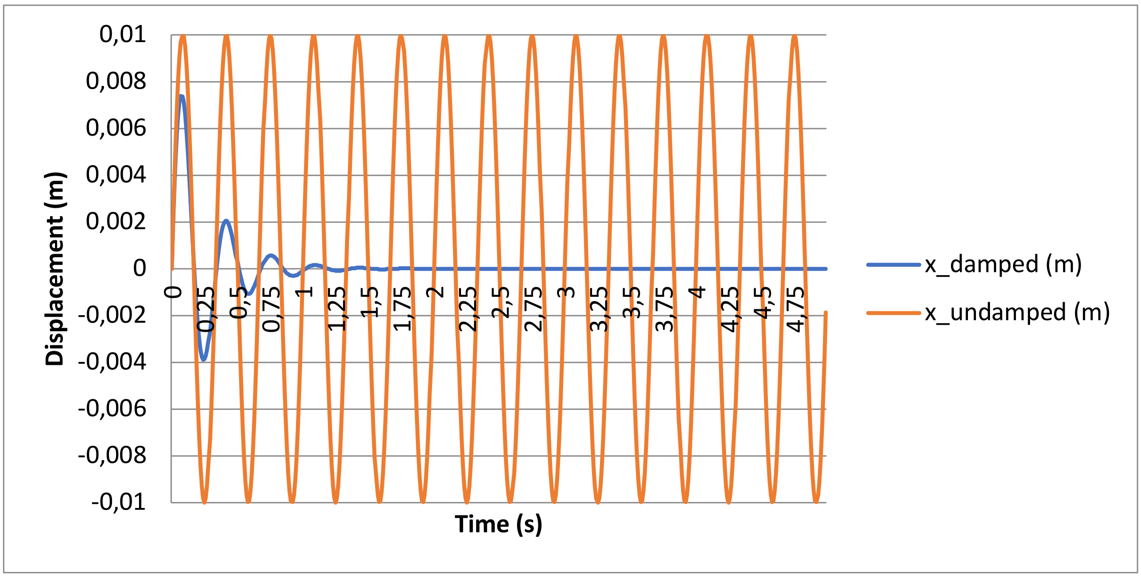

Figure 6. Damped vs Undamped Time Response

Limitations and Future Work

The analysis centers the dominant mode with an SDOF approximation. Real machines exhibit multi-DOF behavior (suspension geometry, tub slosh, cabinet panels). Future work should (i) extend FRF beyond the first mode, (ii) include operational modal analysis under rotating excitation, and (iii) explore control-mechanical co-design for example, pairing specific damper curves with speed-ramp policies and imbalance redistribution routines.

Table 2. Calculated Dynamic Quantities of Washing Machine Drum Vibration

Kondisi Uji	(m) (kg)	f_n (Hz)	ζ (FD)	ζ (FRF)	(k)(N/m)	(c) (N·s/m)	(T)
Empty, OEM	12.0	2.95	0.159	0.203	4115	179	1.63
2 kg dry, OEM	14.0	2.77	0.150	0.216	4265	183	1.76
4 kg wet, OEM	16.0	2.63	0.143	0.231	4363	190	2.00
4 kg wet, High-damper	16.0	2.63	0.180	0.223	4363	235	1.45

Consistency of identification and parameter correlation.

The natural frequency $f_n$obtained from the free-decay analysis decreases as the effective mass increases (from 2.95 Hz for the empty drum to 2.63 Hz for the 4 kg wet load), consistent with the theoretical relation $f_n \propto \sqrt{k/m}$. Meanwhile, the damping ratio ζshows a slight rise with wet loading (0.159 → 0.143 for the OEM configuration due to added viscous and frictional effects) and a marked increase for the high-damper configuration ($\zeta_{\mathrm{FD}} = 0.180$; $\zeta_{\mathrm{FRF}} \approx 0.223$).

Both time and frequency domain estimators of damping display matching trends, indicating measurement reliability. Minor differences between $\zeta_{\mathrm{FD}}$and $\zeta_{\mathrm{FRF}}$are attributed to FRF bandwidth resolution and the peak-to-peak sensitivity of the free-decay method.

Physical influence on isolation (T) and energy dissipation.

The increase in $\zeta$and c(from ≈180 to ≈235 N·s/m) directly corresponds to the reduction in peak transmissibility T(from 2.00 for the 4 kg wet OEM condition to 1.45 for the high-damper case, about −27%). Mechanically, higher damping lowers amplitude around resonance, transforming a sharp resonance peak into a broader but flatter response.

The resulting trade-off bandwidth widening is acceptable as long as the washing machine does not remain for extended periods at non-critical speeds. Because the stiffness $k$remains nearly constant (≈4.1–4.4 kN/m), variations in response are governed mainly by mass and damping, not by the spring system. This confirms that performance can be improved primarily through damper retuning rather than redesigning the suspension.

Design and operational implications.

From a mechanical-design perspective, increasing $\zeta$is the most direct lever to reduce transmitted vibration (T) at the critical speed without significantly altering f_n. The recommended design target is a moderate damping ratio (≈0.18–0.22) within the operational speed corridor, using damper characteristics (force–velocity curve) stable over temperature and aging.

From an operational-control standpoint, the identified parameters can guide an adaptive spin-up profile: accelerate through the resonance zone quickly to avoid long dwell times, and adjust acceleration rates based on detected unbalance or wet load conditions.

The combination of an enhanced damper and an intelligent spin-control strategy yields tangible vibration reduction without significant penalties in cycle time or energy consumption.

CONCLUSION

This study presented a systematic approach for identifying and analyzing the dynamic vibration characteristics of a washing machine drum using three complementary methods: free-decay analysis, frequency response function (FRF) testing, and operational spin-up evaluation. By modeling the system as a single-degree-of-freedom (SDOF) mass–spring–damper system, key dynamic parameters natural frequency (f_n), damping ratio (ζ), stiffness (k), damping coefficient (c), and transmissibility (T) were successfully quantified and compared across different load and damper configurations.

The results demonstrated that increasing drum load lowers the natural frequency due to higher effective mass, while higher damping levels reduce the resonance peak and transmissibility without significantly affecting f_n. The close agreement between time-domain (ζ_{FD}) and frequency-domain (ζ_{FRF}) damping ratios confirmed the reliability of the measurement procedures and the linear SDOF assumption within the tested range.

In practical terms, implementing a higher damping ratio (≈0.18–0.22) effectively reduces transmitted vibration by up to 27%, improving both mechanical isolation and user comfort during high-speed spinning. The experimental setup and analytical workflow developed in this research can serve as a reproducible reference for vibration analysis of domestic appliances or similar suspended rotating systems.

Future extensions of this work may involve multi-degree-of-freedom (MDOF) modeling, nonlinear damper characterization, and integration with adaptive spin control algorithms to further optimize vibration isolation and energy efficiency.

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