POLITECNICO MILANO 1863

SCUOLA DI INGEGNERIA INDUSTRIALE E DELL'INFORMAZIONE

Executive Summary of the Thesis

Developement of a Digital Twin of the MoLAS for Outdoor Tire Testing and Application to Driveline Dynamic Instability Mitigation

Laurea Magistrale in Mechanical Engineering - Ingegneria Meccanica

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1. Introduction

As the sole vehicle-road contact point, the tire is a critical non-linear safety system governing vehicle stability. Moving beyond traditional towed systems like the TASS semi-trailer [6], the MoLAS facility utilizes a dual-actuation powertrain (ICE and electromagnetic braking) to characterize longitudinal slip in both traction and braking. The system enables multi-axial testing via steering ($\pm 20°$), vertical loading (up to 20 kN), and camber ($\pm 5°$) control.

To bridge the gap between repeatable indoor testing and logistically intensive outdoor campaigns, this research introduces a Digital Twin (DT) as a virtual surrogate for the outdoor tire-testing setup, including beside the measuring unit a complete tractor-trailer model. Developed in MATLAB Simulink, the physics-based DT is experimentally validated against real track data, and optimizes track testing at Vizzola Ticino proving ground by predicting the influence of tractor and trailer compliance and kinematic variations on tire measurements. A key advancement is the investigation of severe driveline torsional instabilities observed during braking maneuvers that compromise measurement quality.

2. MoLAS Digital Twin

The MoLAS Digital Twin architecture follows a modular approach, partitioned into two primary configurations: indoor (developed by Semperboni L. [5]) and outdoor (see Figure 1). Although these models operate independently, the outdoor configuration effectively incorporates the indoor model almost in its entirety. Specifically, the measuring unit subsystems are fully integrated into the outdoor setup, serving as the functional core for tire characterization during track tests.

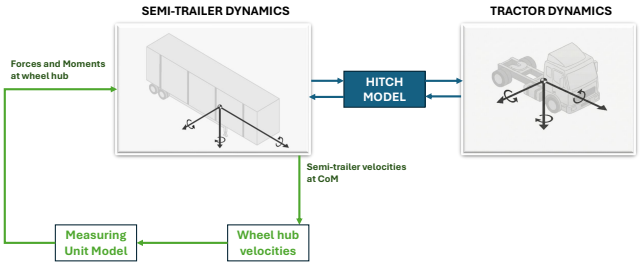

Figure 1: Scheme of the outdoor Digital Twin.

2.1. Measuring Unit Model

The model (schematized in Figure 2) is built around the driveline sub-system that acts as a backbone for the entire simulation. By solving the torque balance between actuation inputs

and tire-road interaction forces, this core governs the wheel rotational dynamics and angular velocity. The physical relationship between tire kinematics and the resulting hub force is handled by an integrated SWIFT (Short Wavelength Intermediate Frequency Tire) model [3], ensuring that the Digital Twin captures complex, high-frequency tire behaviours. The longitudinal and lateral dynamics are further defined using two sub-models that accounts for the structural compliance of the MoLAS frame, while the steering and camber actuation are modelled trough their dynamic motion equations, which are fundamental to correctly reproduce the tire characterization in all the testing scenarios.

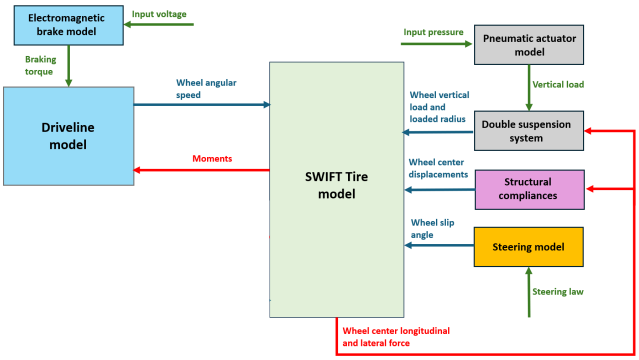

Figure 2: Scheme of the model of the measuring unit.

The vertical subsystem models the double-stage suspension guideways of the MoLAS measuring unit (Figure 3), capturing actuator stick-slip friction through a Karnopp-based formulation [2, 4] with Stribeck effect, coupled with longitudinal and lateral hub forces.

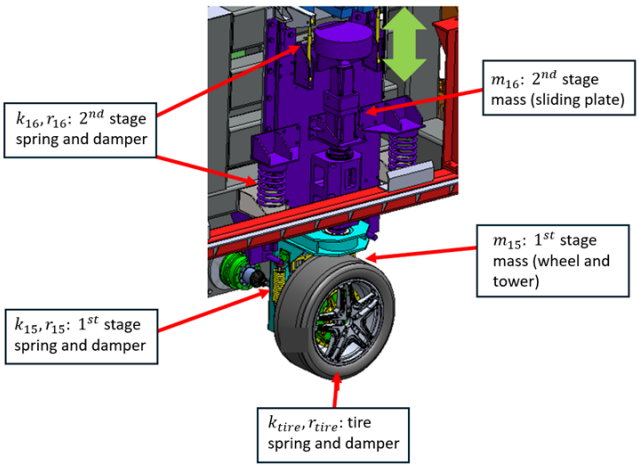

Figure 3: Vertical subsystem principal components.

The developement and validation workflow is illustrated in Figure 4.

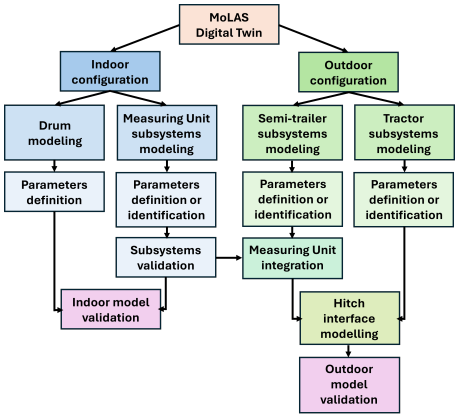

Figure 4: Developement and validation workflow.

2.2. Trailer Model

The trailer model architecture is based on a complex multi-body model subdivided into three main components (as reported in Figure 5). The trailer body 6DOF block implements a rigid-body representation of a two-axle trailer to calculate its kinematics and dynamics. The trailer body inertia tensor has been computed using a sophisticated CAD model designed during the development process of the MoLAS semi-trailer. The wheel subsystem utilizes the Magic Formula 6.2 [3] to characterize tire-road interactions, leveraging a 295/75 R22.5 truck tire dataset as a physical baseline. Built on a 2DOF "Combined Slip" architecture, the model tracks wheel rotation and vertical travel while computing the full six-component array of forces and moments ($F_{x,y,z}$ and $M_{x,y,z}$).
To capture the unique mechanical response of the MoLAS trailer, the suspension subsystem was modelled as a solid axle featuring a pneumatic air spring model, a kinematic damper and an anti-roll bar.

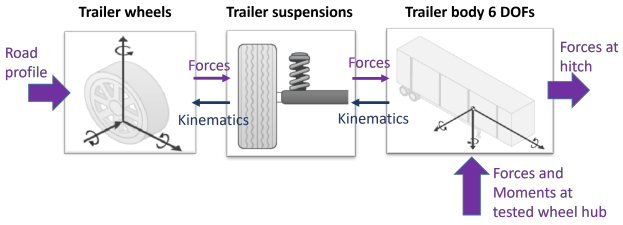

Figure 5: Trailer model subsystems.

The Digital Twin incorporates three specialized sub-models designed to handle the non-linearities of the physical suspensions (schematized in Figure 5):

- Pneumatic Air Springs: an isentropic relationship is used to calculate restorative

forces based on internal pressure and volume gradients;

- Kinematic Dampers: a custom block was authored to account for the circular trajectory of the axle relative to the chassis, resolving damping forces in both the vertical and horizontal planes;
- Anti-Roll Bar: a structural torsional model was integrated to modulate roll stiffness without compromising pure vertical compliance:

$$F_{antiroll} = \frac{k_{t,bar} \cdot \Delta\phi_{rel}}{L_{arm}}. \tag{1}$$

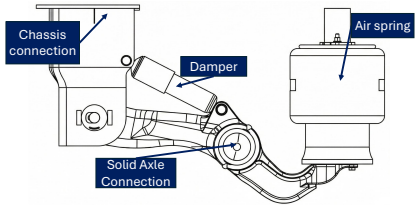

Figure 6: Suspension system overview.

The model reliability is ensured by experimental tests performed to characterize some crucial parameters, such as the air spring effective area or the damping coefficients.

2.3. Tractor Model

The tractor unit is modelled as a 6-DOF rigid body and a Volvo FH16 780 4x2 was considered as a reference. The model features three subsystems: body, wheels and suspensions. To simulate realistic directional control, the steering input is managed by a Stanley lateral controller [1].

The longitudinal actuation framework is governed by a powertrain model implemented through a series of interpolated performance curves (reported in Figure 7). This subsystem processes vehicle speed as an input to deliver wheel torque, accounting for discrete gear ratios, the final drive ratio, and overall mechanical efficiencies across the driveline.

2.4. Hitch Model

The coupling between the tractor and trailer is managed by a hitch subsystem implemented as a 3-DOF translational spring-damper assembly. By regulating relative displacements and velocities along the longitudinal (x), lateral (y), and vertical (z) axes, the model ensures accurate force transmission during dynamic manoeuvrers.

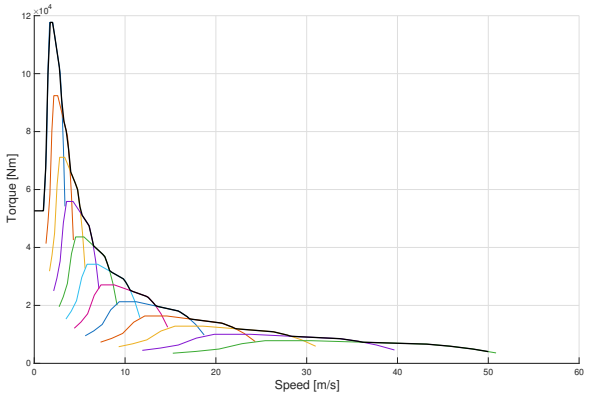

Figure 7: Interpolated performance curve of the ICE engine.

2.5. Inter-systems State Mapping

Owing to the complexity of the outdoor multibody system, several kinematic and dynamic transformations are required to align the trailer global motion with the local input requirements of the SWIFT tire model. The steering and the camber inputs are affected respectively by trailer side-slip angle (as shown in Figure 8) and roll motion, arising the necessity to compute the new imposed values to the tested tire as:

$$\psi_{outdoor} = \psi_{imposed} + \beta_{MoLAS} \tag{2a}$$

$$\gamma_{outdoor} = \gamma_{imposed} + \theta_{MoLAS} \tag{2b}$$

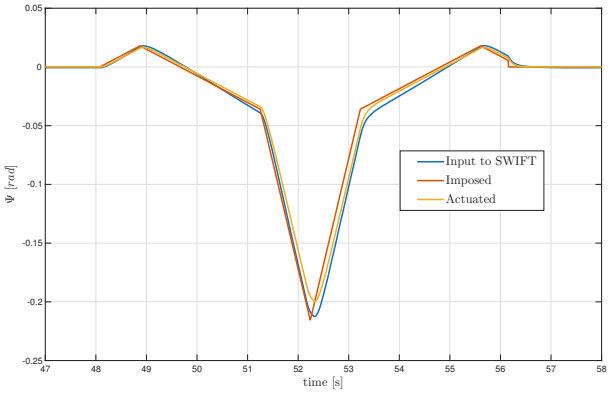

Figure 8: Effect of the actuation and of the trailer on steering angle.

The most critical point is the velocity and force mapping between the tested tire and the trailer CoM. Three rotation matrices account for reference frame differences, and the transport theorem is applied between the tire hub and the MoLAS CoM.

3. Results and Model Applications

The Digital Twin was deployed across four complementary applications: the experimental validation of the outdoor model, pre-test scheduling for the Vizzola outdoor campaign, a comparative indoor-vs.-outdoor analysis to quantify trailer dynamic coupling on tire measurements, and the investigation and mitigation of driveline torsional instabilities during braking.

3.1. Outdoor Model Validation

The outdoor model was experimentally validated against hub force data collected during real outdoor track tests at 25 km/h. The simulation closely reproduces the characteristic shape of the measured force channels throughout both lateral and longitudinal maneuvers. As shown in Figures 9 and 10, simulated hub forces demonstrate a high correlation with the experimental track measurements, confirming the representativeness of the Digital Twin for outdoor tire characterization.

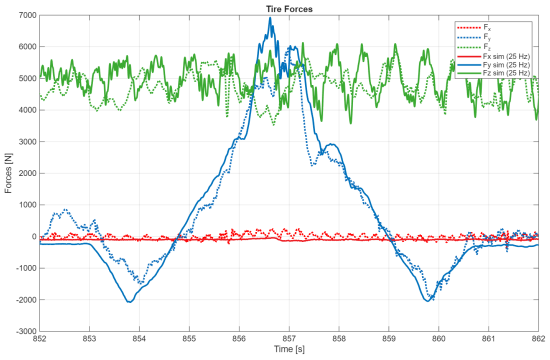

Figure 9: Hub forces during the lateral validation test: measured (dotted) vs. simulated (solid).

3.2. Test-day Simulation

The Digital Twin is initially used for pre-test scheduling at the Vizzola proving ground. Straightaway lengths were evaluated to ensure sufficient time for braking and steering actions from steady-state conditions, while leaving adequate stopping distance. U-turn simulations (Figure 11) verified the tractor-trailer's capability to maneuver within track boundaries.

Three test scenarios were defined, including warm-up procedures to maintain tire temperature, alongside lateral and longitudinal maneuvers across varying loads, pressures, and camber angles.

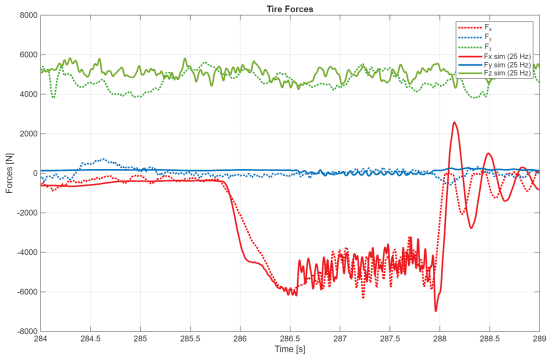

Figure 10: Hub forces during the longitudinal validation test: measured (dotted) vs. simulated (solid).

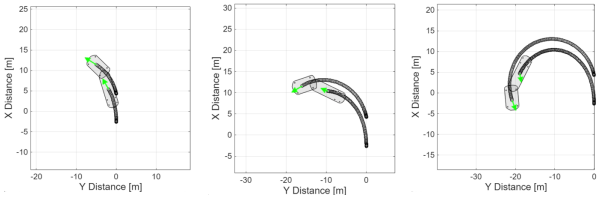

Figure 11: U-turn test simulation.

Following the experimental validation, a comparative analysis was performed to assess the predicted influence of the dynamics of the trailer on the parameters fundamental for the correct tire characterization. The effects of the trailer Euler angles (reported in Figure 12) have been analysed, focusing on their effects on the input quantities of the SWIFT tire model: the results show a relation between the yaw angles imposed by the trailer and the input steering angle given to the tire, as well as a variation in the camber angle caused by the roll action of the MoLAS during the tests.

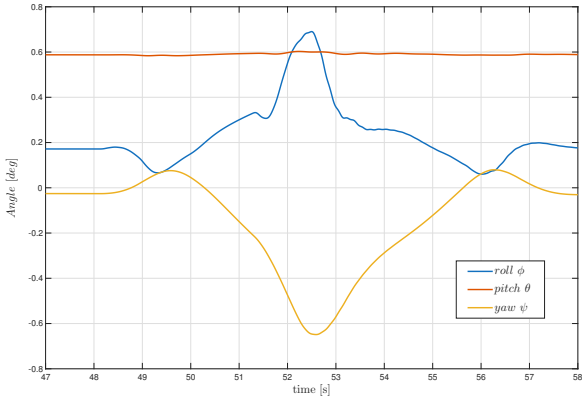

Figure 12: Trailer angles during lateral test at tire nominal load $F_z = 5000\,\mathrm{N}$.

4. Driveline Instability and Active Control

4.1. Instability Characterization

During longitudinal braking, when the tire transitions into the post-peak saturated friction region (negative slope of F_x vs. κ), the negative damping introduced by the tire contact patch drives severe self-excited torsional oscillations in the driveline. These instabilities corrupt hub force measurements and hinder tire parameter identification. The DT was employed to investigate such instabilities and evaluate mitigation strategies. Spectral analysis of experimental and simulated data isolates two dominant resonant modes:

- **13 Hz**: primary torsional mode of the driveline, visible in wheel angular velocity ω;
- **40 Hz**: secondary structural mode, dominating longitudinal displacement of the hub and degrading the quality of F_x measurement.

The DT accurately replicates onset timing and modal frequencies (Figure 13), qualifying it as the investigation platform for mitigation strategies. The Peak Power Spectral Density (PSD) at each resonance is employed to evaluate the effectiveness of the different mitigation strategies.

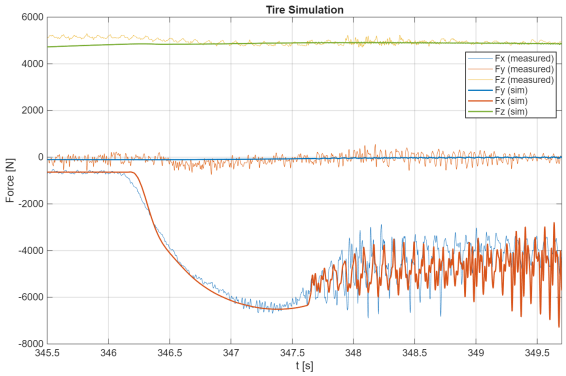

Figure 13: Baseline hub forces during braking: severe oscillations at 13 Hz and 40 Hz degrade measurement quality.

4.2. Passive Solutions

Four candidate mounting positions along the transmission shaft (A, B, C, D in Figure 14) were evaluated under packaging and accessibility constraints. Position C was identified as optimal for the hydrodynamic coupling, while position D was selected for the flexible joint and active disk brake.

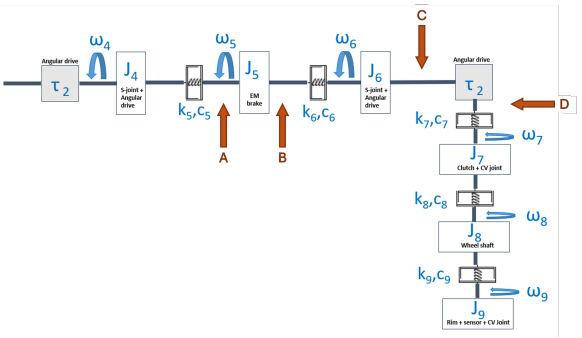

Figure 14: Alternative mounting positions (A, B, C, D) along the transmission shaft evaluated for mitigation countermeasures.

Flexible elastic couplings (position D), modeled as a parallel torsional spring-damper with DIN 740-2 viscous damping, achieve only 50% attenuation of the 13 Hz PSD peak and 7.6% at 40 Hz, insufficient for practical measurement quality.

Hydrodynamic fluid couplings (position C), modeled via slip-factor lookup tables with a first-order fluid transport delay ($\tau_{\text{fluid}} \approx 0.032$ s, cutting off torque fluctuations above 5 Hz), deliver 99.7% / 90.7% attenuation at 13 Hz / 40 Hz. However, only the largest available frame size (FA 590) transmits sufficient torque to drive the system into the tire's saturation zone, and its large external diameter poses significant mounting constraints within the MoLAS driveline geometry.

4.3. Active Disk Brake Control

An active friction disk brake caliper at position D superimposes a corrective torque onto the electromagnetic retarder baseline, combining gain-scheduled PI slip tracking ($e_\kappa(t) = \kappa_{\text{ref}} - \kappa(t)$) with proportional damping on drive shaft velocity differential ($e_\omega(t) = \omega_{\text{trans}}(t) - \omega_{\text{wheel}}(t)$):

$$T_{\text{active}}(t) = K_{p,\kappa} e_\kappa(t) + K_{i,\kappa} \int_0^t e_\kappa(\tau)\, d\tau + K_{p,\omega} e_\omega(t) \tag{3}$$

Controller gains are tuned via Surrogate Model Optimization (SMO) across braking segments (Figure 15).

4.4. Results and Sensitivity Analysis

The active system achieves ≈100% and 98.3% PSD attenuation in wheel speed at 13 Hz and

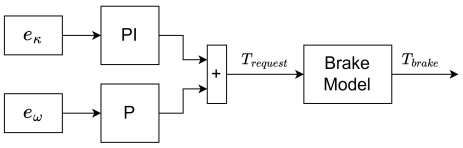

Figure 15: Active control architecture: gain-scheduled PI slip tracking superimposed with proportional torsional damper.

40 Hz, and 99.8% / 98.5% in hub force, outperforming all passive solutions. Figures 16 and 17 show the stabilized wheel angular velocity and suppressed hub force fluctuations, respectively, eliminating self-excited oscillations and restoring clean slip tracking into the tire saturation zone.

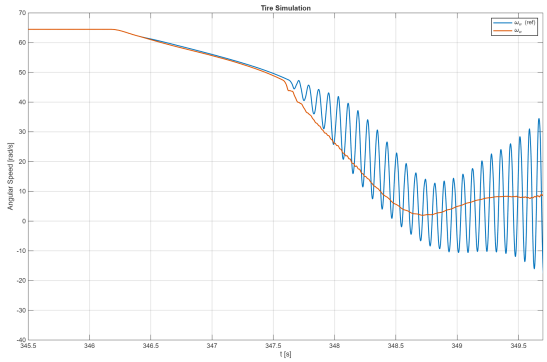

Figure 16: Active disk brake indoor response: wheel speed stabilization.

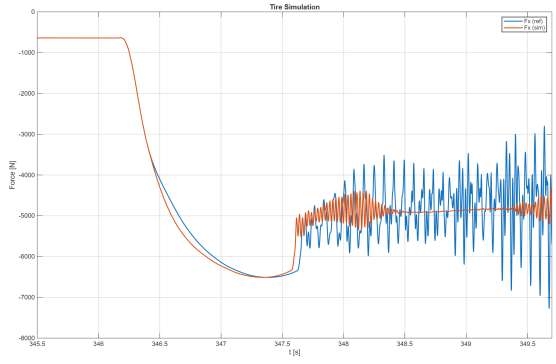

Figure 17: Active disk brake indoor response: tri-axial hub forces attenuation.

Table 1 synthesizes the full comparative assessment. Solutions are reported as percentage PSD attenuation relative to the unmitigated baseline; sensitivity cases show percentage variation relative to the nominal active brake.

A sensitivity analysis across vertical load (1500-7200 N), forward velocity (60-100 km/h), and tire shape factor $PCX1$ ($\pm 10\%$) confirms controller robustness without gain recalibration. Although high load excites the 40 Hz mode (+633%), the critical 13 Hz resonance remains suppressed (-18.6%), while lower loads and higher speeds attenuate both modes. In all conditions, limit cycles are eliminated and slip tracking is preserved.

Finally, the active brake was validated in the outdoor DT on an ISO 8608 rough road profile at 77 km/h. Despite stochastic road excitation, the 13 Hz torsional oscillations remain strongly suppressed and tri-axial hub forces (F_x, F_y, F_z) stay well within bounds for tire characterization.

Configuration	**Atten. 13 Hz**		**Atten. 40 Hz**	
	ΔF_x %	$\Delta\omega$ %	ΔF_x %	$\Delta\omega$ %
Solutions vs. unmitigated baseline				
Flexible Joint (D)	-41.0	-50.2	-9.6	-7.6
Hydrodyn. Joint (C)	-98.5	-99.7	-88.1	-90.7
Active Disk Brake (D)	-99.8	≈ -100	-98.5	-98.3
Active brake sensitivity vs. nominal active				
$F_z = 1500$ N	-55.6	$+18.9$	-99.9	≈ -100
$F_z = 7200$ N	$+66.3$	-18.6	$+789$	$+633$
$PCX1 - 10\%$	-97.2	-81.4	-99.6	-99.7
$PCX1 + 10\%$	$+462$	$+86.1$	$+296$	$+177$
$v = 60$ km/h	-11.7	$+9.3$	$+926$	$+897$
$v = 100$ km/h	$+2005$	$+3567^\dagger$	-99.9	-99.9

†Artifact of near-zero nominal baseline; in absolute terms the peak ω PSD remains at 0.341 $(\text{rad/s})^2/\text{Hz}$, well below the unmitigated value of 23.3 $(\text{rad/s})^2/\text{Hz}$. All sensitivity cases use nominal active brake gains without recalibration.

Table 1: PSD attenuation summary: solutions (vs. unmitigated baseline) and active brake sensitivity analysis (vs. nominal active brake).

5. Conclusions

This thesis developed and validated an outdoor Digital Twin for the MoLAS tire testing platform. Main contributions include:

- A full tractor-trailer multi-body Simulink DT featuring custom suspension models (air springs, kinematic dampers, anti-roll bar) validated against experimental data, extending the indoor model to outdoor tracks;
- A pre-test simulation tool for the Vizzola track, enabling test scheduling, U-turn feasibility checks, and quantification of trailer dynamic effects on tire characterization accuracy;
- The systematic investigation of driveline torsional instabilities (13 Hz and 40 Hz), followed by the design of an active disk brake controller achieving near-total suppression, outperforming passive solutions while ensuring robust-

ness across varying conditions.

Future work involves on-track integration of the active brake hardware, and adding thermal dynamics to the MF-SWIFT model to capture temperature-dependent grip.

References

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