



## Full Length Article

## Dynamic response of a 3D-printed TPU non-pneumatic tyre under obstacle traversal and vibratory excitation

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## ABSTRACT

Non-pneumatic tyres are gaining interest for lightweight off-road mobility because they eliminate dependence on inflation pressure and enable mechanical behaviour to be tailored through structural design. However, controlled comparisons with conventional pneumatic tyres under impact and vibration conditions remain limited. This study evaluated the static and dynamic response of an additively manufactured thermoplastic-polyurethane (TPU) non-pneumatic tyre with a graded V-shaped spoke architecture and compared it with a conventional pneumatic tyre under equivalent loading conditions. Static compression tests were used to characterise the load-displacement response and to support qualitative interpretation using a finite element model. Dynamic performance was assessed through obstacle-traversal tests at 3 and 6 km h<sup>-1</sup> over semicircular obstacles with radii of 3, 6 and 9 mm, and through vibration tests using sinusoidal excitation at 4, 8 and 15 Hz and broadband pink-noise excitation from 50 to 250 Hz. The static tests showed force relaxation and a nonlinear increase in secant stiffness from 25.0 to 33.8 N mm<sup>-1</sup> between 2 and 8 mm displacement. Peak acceleration increased with obstacle size and speed, but no statistically significant difference was detected between the two tested tyre specimens under any matched traversal condition. The non-pneumatic tyre achieved greater positive vibration attenuation than the pneumatic tyre, reaching 14.22%, 25.80% and 49.48% at 4, 8 and 15 Hz, respectively, and 38.84% under broadband excitation. Under the tested lightweight conditions, the manufactured TPU tyre improved vibration isolation, although further testing with independently manufactured specimens is required to establish production repeatability and tyre-to-tyre variability.

## 1. Introduction

Wheel systems are critical components in lightweight mobile platforms, off-road vehicles, agricultural equipment, small trailers and unmanned ground vehicles, where the interaction between the wheel and the travelling surface affects shock transmission, vibration exposure, component durability and payload protection. In these applications, the wheel is not only a rolling element but also a compliant mechanical interface that attenuates irregular terrain inputs before they are transmitted to the chassis, sensors or transported load (Al-Baghdadi et al., 2021). This function becomes especially relevant in low-speed off-road operation, where local obstacles, repeated surface irregularities and broadband vibration can affect the reliability of mechanical and electronic components.

Conventional pneumatic wheels remain widely used because their air-filled structure provides compliance, load-bearing capacity and vibration filtering. However, their performance depends strongly on inflation pressure, tyre geometry, tread condition and operating load. Pressure losses, punctures or inappropriate inflation can modify the effective stiffness of the wheel, increase rolling resistance and reduce reliability in demanding environments (Bashford et al., 1993). These limitations have motivated growing interest in non-pneumatic wheel and tyre concepts, particularly for applications in which puncture resistance, low maintenance and repeatable mechanical response are relevant design requirements.

Non-pneumatic tyres and wheels replace the pressurised air cavity with a structural support architecture composed of spokes, cellular patterns, internal rings or graded lattices. Their mechanical behaviour is therefore governed by

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the combination of material properties and internal geometry. Different support structures have been proposed in the literature, including simple spokes, honeycomb and cellular architectures, anti-tetrachiral and auxetic geometries, gradient patterns and bio-inspired solutions aimed at improving load distribution, compliance and energy absorption (Jin et al., 2018; Liu and Xu, 2022; Montgomery-Liljeroth et al., 2023; Phromjan and Suvanjumrat, 2023; Vashi et al., 2021; Wu et al., 2021; Yang et al., 2024; Z. Zhang et al., 2021). Commercial developments such as the Michelin Tweel also illustrate the practical relevance of polymer-spoke non-pneumatic concepts for improving operational robustness and reducing dependence on inflation pressure (Sardinha et al., 2023; Żmuda and Jackowski, 2025).

Additive manufacturing has accelerated the development of non-pneumatic wheel systems by enabling the rapid fabrication of complex internal architectures that would be difficult or costly to manufacture using conventional methods. Material extrusion additive manufacturing (MEAM) is particularly attractive for prototyping compliant wheel structures because it allows geometrical design iterations, local stiffness tailoring and the use of flexible polymers. Among these materials, thermoplastic polyurethane (TPU) has become a relevant candidate due to its flexibility, high recoverable deformation, nonlinear viscoelastic/hyperelastic behaviour and compatibility with FFF processes (Desai et al., 2023; Xu et al., 2020). TPU-based structures can dissipate mechanical energy through material damping and structural deformation, which makes them suitable for vibration mitigation and impact-related applications (Gosar et al., 2023; Rizzo et al., 2021; Y. S. Zhang et al., 2025).

The mechanical response of additively manufactured TPU components depends not only on the base material formulation but also on printing parameters such as layer height, printing speed, toolpath orientation and interlayer bonding. These factors can introduce anisotropy and modify the effective stiffness, strength and damping response of printed structures (Martín-Sosa et al., 2025). Consequently, the performance of a TPU non-pneumatic wheel cannot be inferred solely from nominal material properties; it must be evaluated at the component level under representative mechanical excitations.

Finite element analysis has been widely used to support the design of non-pneumatic wheel structures by estimating deformation modes, stress distribution, contact response and load-bearing behaviour before physical manufacturing (Abraar et al., 2022; Fu et al., 2022; Genovese et al., 2021; Kim et al., 2024; Sun et al., 2024a,b). Computational modelling and structural analysis methods are essential tools for predicting the mechanical response of solid tyres. Whilst simplified analytical formulations can adequately capture in-plane stiffness in two-dimensional geometries (Gasmi et al., 2012), fully three-dimensional configurations with complex spoke topologies require finite element simulations. However, high-fidelity finite element analyses entail a high computational cost, particularly when evaluating multivariable design spaces. To optimise this computational effort, experimental design schemes such as the Taguchi method have been employed in the analysis of multiaxial stiffness and ground contact in cellular structures (Shuai et al., 2024; Zheng et al., 2022b). Similarly, the response surface methodology has enabled the evaluation of cornering stiffness (Zheng et al., 2022a), whilst staged numerical-experimental approaches have proven effective in validating rolling dynamics under service loads (Baranowski et al., 2016). Furthermore, analysis of variance (ANOVA) has been used to establish a relationship between elastomeric microstructural variables and the final mechanical properties (Nuñez Carrero et al., 2022).

Although previous studies have reported relevant advances in non-pneumatic wheel design, many contributions focus primarily on static stiffness, numerical optimisation or isolated structural analyses. Fewer studies provide controlled experimental comparisons between additively manufactured TPU non-pneumatic wheels and conventional pneumatic wheels under equivalent loading conditions. In particular, the distinction between two different dynamic situations remains insufficiently explored: short-duration obstacle-induced impacts and continuous or broadband vibratory excitation. These two loading cases are governed by different mechanical mechanisms.

Obstacle traversal is dominated by contact geometry, effective stiffness and inertia, whereas vibration isolation depends more strongly on transmissibility, damping and frequency-dependent structural response.

Therefore, the objective of this study is to experimentally analyse the static and dynamic behaviour of a TPU-based non-pneumatic wheel manufactured by material extrusion additive manufacturing and to compare its performance with a conventional pneumatic wheel under equivalent vertical load. The study combines static compression testing, finite element modelling, obstacle traversal tests and controlled vibration excitation. The specific contributions are:

- (i) A direct experimental comparison between a 3D-printed TPU non-pneumatic wheel and a conventional pneumatic wheel under identical loading and boundary conditions.
- (ii) A combined assessment of obstacle-induced peak acceleration and vibration attenuation under harmonic and broadband excitation.
- (iii) The use of finite element modelling to support the interpretation of deformation patterns and stress localisation in the additively manufactured wheel structure.
- (iv) Experimental evidence showing that vibration isolation and peak impact response should be treated as distinct design objectives in compliant non-pneumatic wheel systems.

The results are intended to support the engineering design of puncture-free compliant wheels for lightweight off-road mobility, including mobile robotic platforms, small transport systems and agricultural equipment where vibration protection and operational robustness are relevant.

## 2. Materials and methods

### 2.1. Tyre systems and experimental concept

Two tyre configurations were evaluated: an experimental non-pneumatic tyre (NPT) and a conventional pneumatic tyre (Fig. 1). The pneumatic configuration consisted of a conventional rubber tyre with an inner tube, external dimensions of  $165 \times 50$  mm and an inflation pressure of 30 psi. Both configurations used identical rim and bearing arrangements; the only difference was the replacement of the conventional pneumatic tyre with the non-pneumatic tyre. The mounting system incorporated two ball bearings and a 54 mm hub connection to ensure equivalent boundary conditions during testing.

The non-pneumatic wheel evaluated in this study was designed with a graded V-shaped spoke architecture. The internal structure was organized into two functional regions: an inner region with greater structural density and larger cross-section to provide load-bearing stiffness, and an outer region with lower structural density and reduced cross-section to promote deformability and energy absorption. This design rationale was intended to combine radial compliance in the outer region with sufficient stiffness near the hub, allowing the inclined spokes to deform mainly through bending-dominated mechanisms rather than purely compressive load transfer. Similar functionally graded concepts have been reported as effective strategies for balancing load capacity, stress distribution and compliance in non-pneumatic tyre structures (Phromjan and Suvanjumrat, 2023; Yang et al., 2024; Zhou et al., 2021).

The non-pneumatic tyre was manufactured from Creality CR-TPU filament (nominal diameter 1.75 mm; Shore hardness approximately 95A; density  $1.21 \text{ g cm}^{-3}$ ) using a Creality Ender-3 material-extrusion additive-manufacturing system. Before printing, the filament was dried for 10 h at  $50^\circ\text{C}$ . The tyre was printed using a 0.4 mm nozzle, a nozzle temperature of  $218^\circ\text{C}$ , a build-plate temperature of  $50^\circ\text{C}$ , a layer height of 0.2 mm and a nominal printing speed of  $25 \text{ mm s}^{-1}$ . The geometry was printed using four perimeter walls. The internal architecture was modelled explicitly; therefore, no slicer-generated infill was used. The printing angle was set to  $45^\circ$ , based on previous evidence indicating that layer orientation affects stiffness, strength

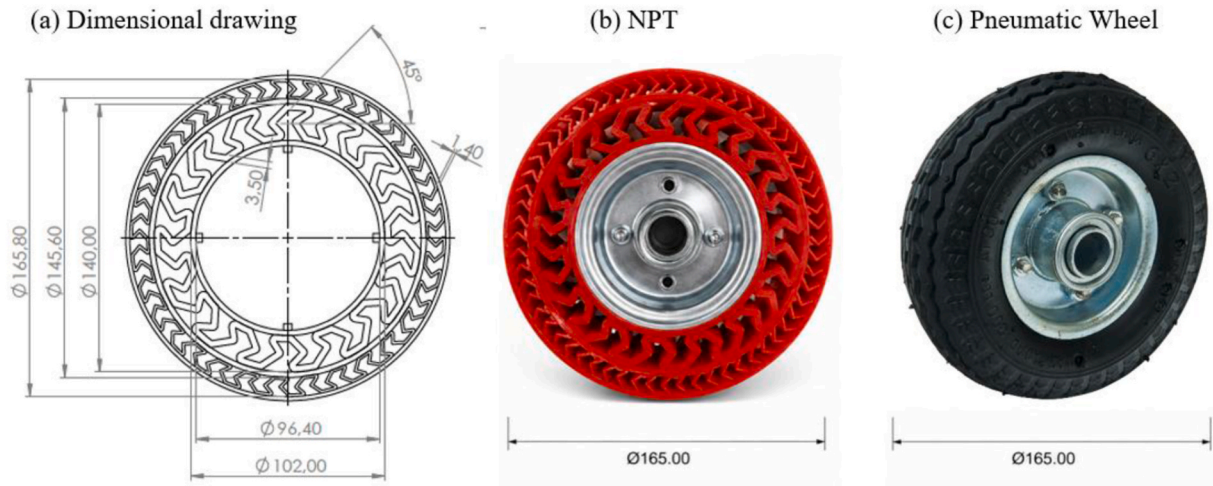


Fig. 1. NPT wheel and dimensions (a, b) and conventional wheel (c).

and anisotropy in printed TPU parts, and that intermediate orientations may provide favourable mechanical performance (Martín-Sosa et al., 2025). TPU was selected because of its flexible and viscoelastic behaviour, as well as its demonstrated potential for energy dissipation and vibration mitigation applications (Desai et al., 2023; Gosar et al., 2023; Rizzo et al., 2021; Y. S. Zhang et al., 2025). No supports or raft were used, but a 5 mm brim was applied to guarantee bed adhesion. After printing, no post-processing other than support removal was performed. These parameters are reported to enable reproduction of the manufactured architecture.

The experimental setup was conceived as a simplified wheel–axle system representative of lightweight off-road platforms such as small trailers, wheelbarrows or mobile robotic systems. The objective was not to reproduce all the complexities of real wheel–soil interaction, but to establish a controlled and repeatable methodology for comparing the structural dynamic response of both wheel systems under the same vertical load and excitation conditions.

A prototype test rig (Fig. 2) was built using a horizontal rotating arm with a length of 1.8 m. The arm rotated around an electric motor, allowing continuous circular motion at controlled forward speeds. A

rigid box mounted on the arm was used to accommodate adjustable mass and impose a constant vertical load of 100 N during all experimental tests. This load was selected to evaluate the wheel response under lightweight operating conditions. The analysis focused on the vertical component because it is directly related to transmitted shock, axle vibration and payload excitation during obstacle contact. Longitudinal and lateral forces were not considered in this study, as they are mainly associated with traction, braking and steering effects, which were outside the scope of the present controlled comparison.

At the distal end of the arm, a dedicated mounting structure housed the wheel axle. An onboard wireless MEMS accelerometer (LLC X200-5, Gulf Coast Data Concepts, USA;  $\pm 200$  g range) was mounted on the axle structure at a height of 25 cm to measure acceleration signals during motion. The vertical acceleration component was used to evaluate the dynamic response of the wheel–axle system during obstacle traversal.

## 2.2. Static compression characterisation and finite element model

The static compression response of the TPU non-pneumatic tyre was

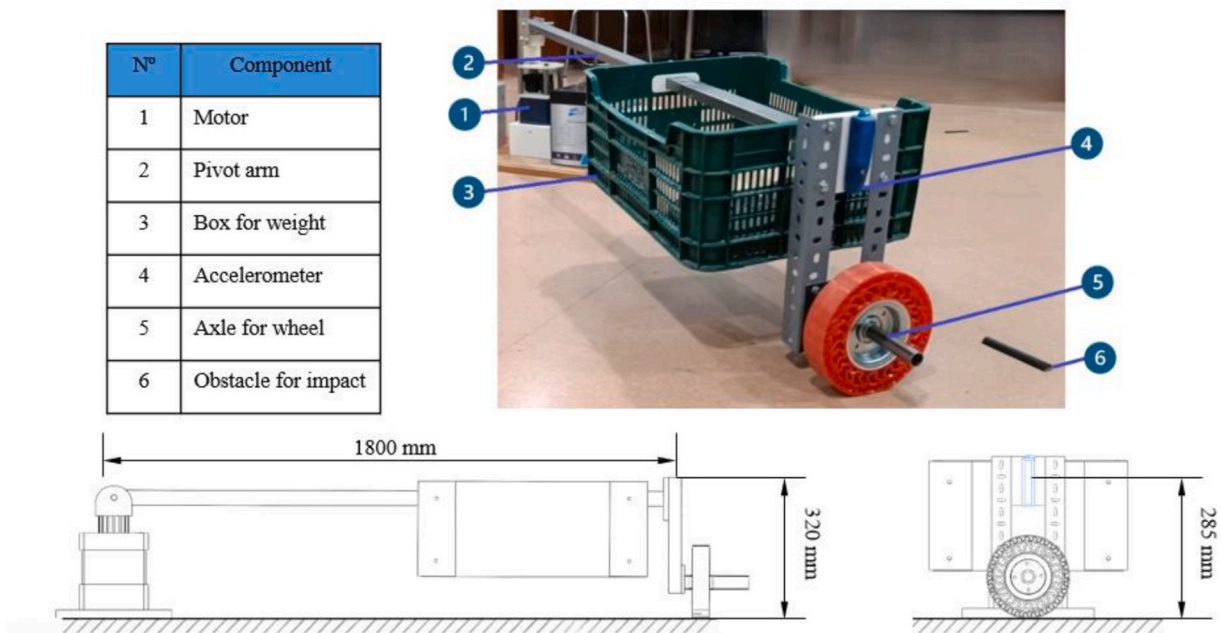


Fig. 2. Test rig configuration.

measured using a custom-built compression rig operating in displacement-control mode (Fig. 3). Vertical displacement was applied from 2 to 10 mm in increments of 2 mm. The reaction force was recorded using the integrated load cell and a dynamometer (PCE-PFG 2K, Albacete Spain) with a resolution of 0.5 N. At each displacement level, a 60 s holding period was introduced to allow force relaxation and obtain a quasi-steady force measurement. Photographic documentation of the deformed structure was collected at the different displacement levels to support comparison with the numerical deformation patterns.

The TPU material was represented using a second-order, five-parameter Mooney–Rivlin hyperelastic model (Rivlin, 1948). For an incompressible material, the strain-energy density function is expressed as:

$$W_{(Mooney-Rivlin)} = C_{10} (\bar{I}_1 - 3) + C_{01} (\bar{I}_2 - 3) + C_{11} (\bar{I}_1 - 3)(\bar{I}_2 - 3) + C_{20} (\bar{I}_1 - 3)^2 + C_{02} (\bar{I}_2 - 3)^2 \quad (1)$$

where  $\bar{I}_1$  and  $\bar{I}_2$  are the first and second modified strain invariants, respectively, and  $C_{10}$ ,  $C_{01}$ ,  $C_{20}$ ,  $C_{11}$  and  $C_{02}$  are material coefficients expressed in MPa. The coefficients were obtained from the experimental tensile dataset reported by Martín-Sosa et al. (2025), comprising five TPU specimens printed at a 45° raster orientation. Fig. 4 shows the experimental tensile data and the corresponding Mooney–Rivlin fit, while the fitted coefficients are reported in Table 1.

Using the adopted material properties, a finite element model of the TPU non-pneumatic wheel was developed in ANSYS Workbench 2023 R1 to reproduce the static compression test and support the interpretation of the deformation mechanisms. The deformation of the inner rigid hub was ignored because the inner rigid hub is generally made of aluminium alloy, which is much harder than TPU. The deformation of the ground was ignored and wheel–ground contact was defined as frictional, with a friction coefficient of 0.8, while bonded contacts were applied at the internal interfaces (Fig. 5). These contact assumptions are consistent with modelling approaches used in previous NPT studies (Fu et al., 2022; Genovese et al.,

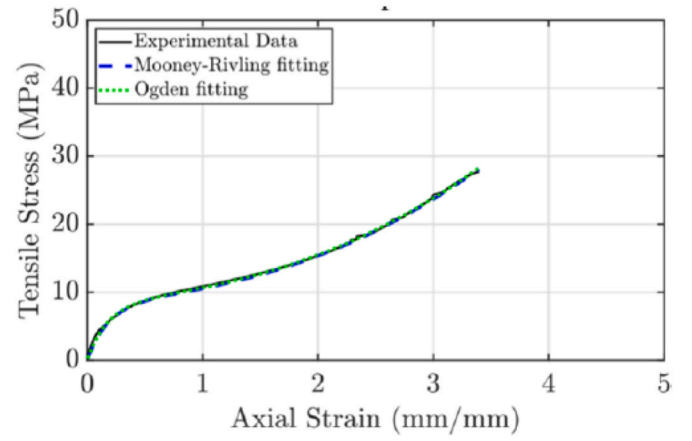


Fig. 4. Comparison of the experimental TPU data with Mooney–Rivlin model (Martín-Sosa et al., 2025).

Table 1

Mooney–Rivlin coefficients used for the TPU material model.

| Parameters   | $C_{10}$ | $C_{01}$ | $C_{20}$ | $C_{11}$ | $C_{02}$ |
|--------------|----------|----------|----------|----------|----------|
| Values (MPa) | −7.725   | 15.849   | 0.209    | 0.013    | 1.83     |

2021; Kim et al., 2024). The applied displacement restrictions were modelled using a reference point coupled to the inner rigid hub, coinciding with the experimental displacement. As the NPT structure is complex, a tetrahedral mesh (SOLID187) was selected to divide the entire NPT (Fig. 5), following common practice in comparable NPT analyses (Fu et al., 2022; Genovese et al., 2021). A mesh-convergence study was conducted to select an appropriate element size. Table 2 shows the effect of the number of elements on the maximum stress under an applied displacement of 4 mm. The maximum stress fluctuated when the number of elements was less than 80,000, indicating that the number of elements influences the numerical

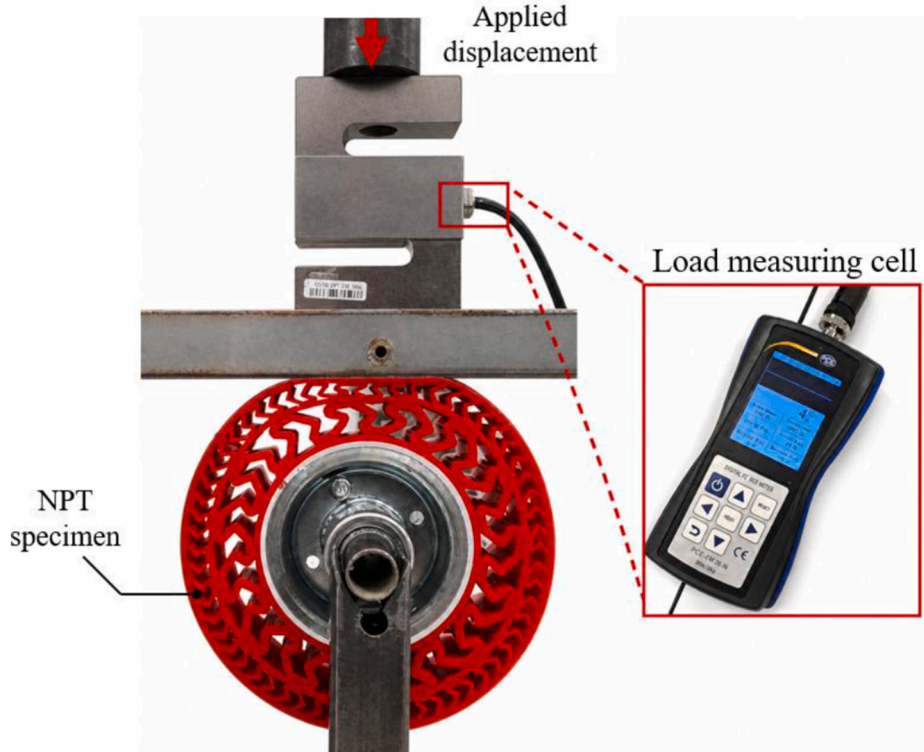


Fig. 3. Static compression of the non-pneumatic tyre using a custom-built displacement-controlled compression rig fitted with a digital force cell.

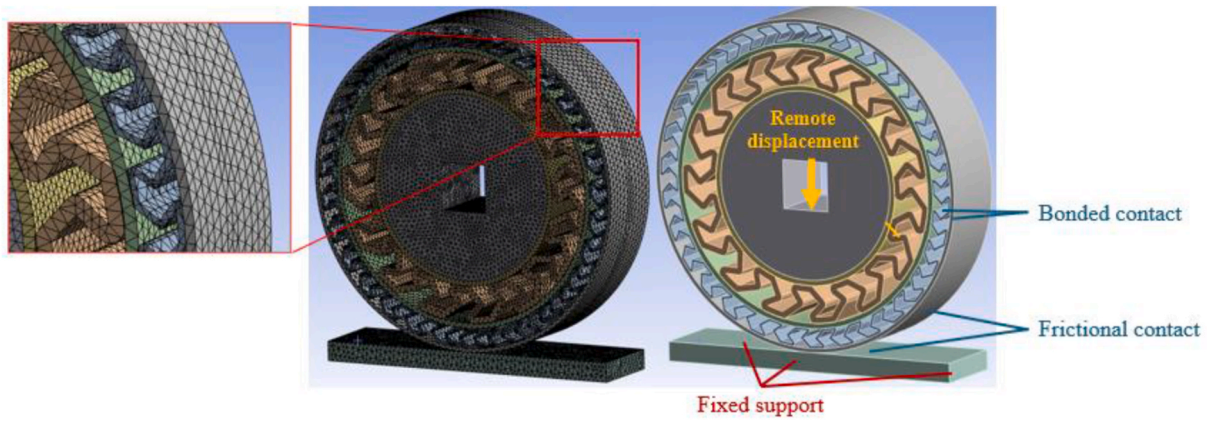


Fig. 5. FE model used for static characterisation.

results. Considering the accuracy and simulation time, a mesh size of 5 mm was selected. Different compression levels were simulated by applying remote displacement, and the resulting deformation modes, strain distribution and stress localisation were compared qualitatively with the experimental observations.

### 2.3. Obstacle traversal response

The obstacle traversal tests were performed using the rotating arm test rig shown in Fig. 2. Motion was controlled by a frequency inverter, and two forward speeds were evaluated: 3 and 6 km h<sup>-1</sup>. These values are representative of low-speed operation in lightweight off-road platforms and unmanned ground vehicles. Three semicircular obstacles with radii of 3, 6 and 9 mm were installed along the circular trajectory, with a spacing of 1 m between consecutive obstacles.

Semicircular obstacles were selected to provide simple, controllable and repeatable contact conditions. This geometry produces a gradual wheel–obstacle interaction and avoids excessively sharp impacts, while still representing rounded irregularities commonly encountered in off-road surfaces. The 1 m spacing was adopted to minimise overlapping effects between consecutive impacts and to allow each obstacle interaction to be analysed independently.

For each combination of tyre type, forward speed and obstacle size, 25 repeated traversal events were recorded using the same tyre specimen. These repeated measurements were used to characterise within-specimen test variability and measurement dispersion; they should not be interpreted as independent manufactured-tyre replicates. Acceleration was sampled at 400 Hz, and the peak vertical acceleration associated with each obstacle-traversal event was extracted.

### 2.4. Vibration excitation tests: isolation performance

Vibration tests were conducted using an electromagnetic shaker (LDS V406, Nærum, Denmark) connected to a power amplifier to quantify acceleration transmission from an excited base to the wheel axle (Fig. 6). The test was designed to compare the isolation capability of the pneumatic wheel and the TPU non-pneumatic wheel under controlled excitation

conditions. Acceleration signals were acquired using a multichannel dynamic analyser (OR36 Mobi-Pack, OROS, France). A uniaxial accelerometer (PCB 352C33, Depew, New York, NY, USA) was mounted on the base plate to measure the input acceleration, while another accelerometer was mounted on the axle to measure the transmitted output acceleration.

Vibration attenuation was evaluated using the amplitude attenuation ratio  $\eta$ , calculated from the RMS acceleration at the input base and the RMS acceleration at the wheel axle, as defined in Equation (2). According to this definition, positive values indicate amplification, whereas negative values indicate attenuation.

$$\eta = 100 \frac{A_{\text{base-plate}} - A_{\text{wheel}}}{A_{\text{base-plate}}} \quad (2)$$

Sinusoidal excitations at 4, 8 and 15 Hz were applied to represent low-frequency ranges relevant to vehicle and implement motion. In addition, broadband pink-noise excitation from 50 to 250 Hz was applied for 20 s to evaluate vibration isolation under random excitation. Pink-noise excitation was selected to provide broadband input with greater energy at lower frequencies, following approaches used in transport-vibration testing (Fernando et al., 2020). Five repeated vibration records were acquired for each tyre and excitation condition using the same physical specimen.

Signal processing was performed in MATLAB (MathWorks, USA). Statistical analysis was carried out using IBM SPSS Statistics 25. the normality test. Non-parametric tests were applied when the data failed either the Shapiro–Wilk normality test or Levene's test for homogeneity of variance. Kruskal–Wallis tests were used for multi-group comparisons, while Mann–Whitney U tests with Holm correction were used for pairwise comparisons. For vibration attenuation comparisons, parametric tests were applied when assumptions were met, using Student's t-test with Holm correction. The significance level was set at  $\alpha = 0.05$ . When no statistically significant differences were detected between wheel types, results are reported as adjusted  $p > 0.05$ .

## 3. Results

### 3.1. Static compression response and numerical deformation patterns

Static compression tests revealed a consistent transient response in the TPU non-pneumatic wheel (Fig. 8). When the prescribed displacement was reached, the measured force showed an initial peak followed by relaxation towards a quasi-steady value. At 2 mm displacement, the force decreased from approximately 60 N to 50 N. At 4 mm, it decreased from 130 N to 110 N. At 6 mm, it decreased from 230 N to 190 N, and at 8 mm, it decreased from 330 N to 270 N. This peak-then-relaxation behaviour is consistent with the viscoelastic response of TPU and with progressive stress redistribution within the deformable spoke structure, Fig. 7.

The quasi-steady force values indicate a nonlinear increase in vertical stiffness with displacement. Based on the stabilised force values, the

Table 2

Mesh quality with different number of elements.

| Number of elements | Number of nodes | Mesh metric average (mm) | Number of errors | Maximum stress (Mpa) |
|--------------------|-----------------|--------------------------|------------------|----------------------|
| 34520              | 74841           | 7.048                    | 0                | 3.38                 |
| 45371              | 97135           | 6.15                     | 0                | 4.478                |
| 47774              | 99947           | 5.98                     | 0                | 4.26                 |
| 81808              | 157800          | 5.00                     | 0                | 4.31                 |
| 96263              | 186372          | 4.95                     | 0                | 4.29                 |
| 272236             | 465597          | 3.43                     | 0                | 4.27                 |

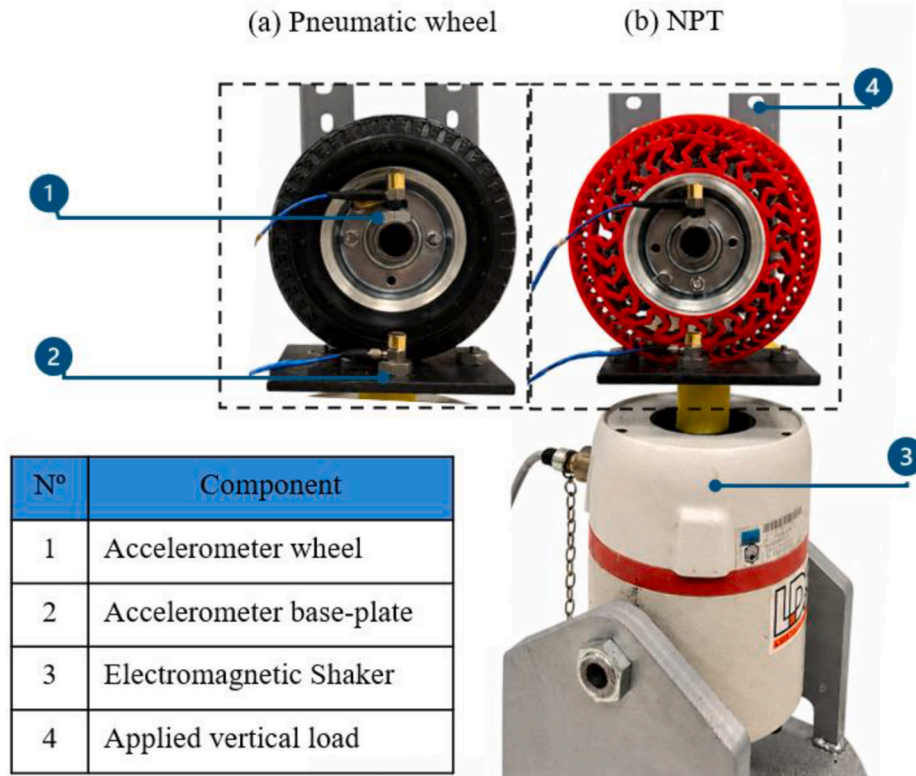


Fig. 6. Vibration excitation test using an electromagnetic shaker.

corresponding secant stiffness values were approximately  $25.0 \text{ N mm}^{-1}$  at 2 mm,  $27.5 \text{ N mm}^{-1}$  at 4 mm,  $31.7 \text{ N mm}^{-1}$  at 6 mm and  $33.8 \text{ N mm}^{-1}$  at 8 mm (Fig. 7). This progressive increase suggests a strain-stiffening response at the component level, which is expected in compliant wheel architecture where increasing deformation progressively engages a larger fraction of the internal structure.

Fig. 8 compares the experimentally observed deformation of the physical tyre with the deformation and stress distributions predicted by the finite element model. The numerical model reproduced the main deformation pattern observed during static compression, particularly at displacements of 6–8 mm, where pronounced asymmetric bending of the lower curved spokes was observed. This behaviour is consistent with a local buckling-type deformation mechanism influenced by the geometry of the

internal spoke architecture. The numerical results also identified regions of elevated stress near the spoke junctions and the nodes of the inner ring, which may represent critical locations for future structural optimisation.

The comparison presented in Fig. 8 is primarily qualitative. Although the simulated deformation pattern was consistent with the experimental observations and with the progressive stiffening measured during the compression tests, no predefined error metric or independent validation dataset was used to establish a quantitative correlation between the numerical and experimental results. Accordingly, the finite element model is used here as a structural interpretation tool rather than as a fully validated predictive model. The experimental force-relaxation response reflects the time-dependent viscoelastic behaviour of TPU, whereas the static hyperelastic model principally captures the geometric nonlinearity, spoke bending and stress redistribution associated with the tyre architecture.

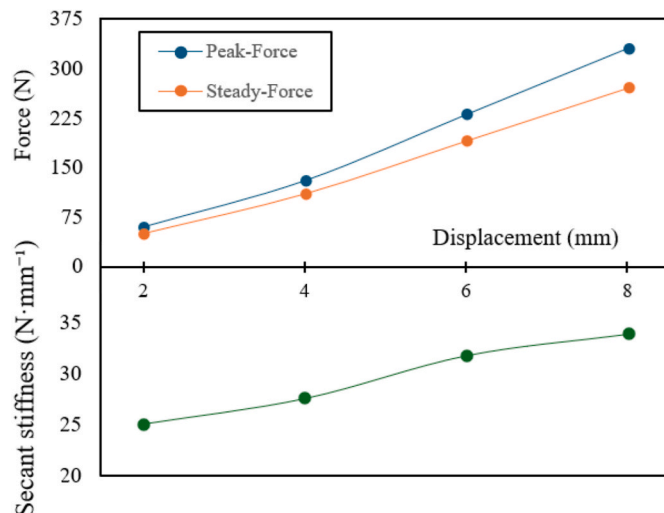


Fig. 7. Displacement vs secant stiffness and force.

### 3.2. Obstacle traversal response

Representative vertical acceleration signals during obstacle traversal showed discrete peaks associated with wheel–obstacle interaction (Fig. 9). These peaks were clearly distinguishable from the background vibration of the rotating system. The time required to complete the analysed trajectory decreased from approximately 8 s at  $3 \text{ km h}^{-1}$  to approximately 4 s at  $6 \text{ km h}^{-1}$ . A constant acceleration offset was observed in the signals due to the MEMS accelerometer configuration and gravitational component.

The mean peak accelerations increased with obstacle size and forward speed for both wheel types (Fig. 10). At  $3 \text{ km h}^{-1}$ , the mean peak accelerations were  $27.42 \pm 4.86 \text{ m s}^{-2}$  for the TPU non-pneumatic wheel and  $22.50 \pm 2.83 \text{ m s}^{-2}$  for the pneumatic wheel when crossing the 3 mm obstacle. For the 6 mm obstacle, the values increased to  $42.74 \pm 4.09 \text{ m s}^{-2}$  and  $36.57 \pm 6.35 \text{ m s}^{-2}$ , respectively. For the 9 mm obstacle, the peak accelerations were almost identical, with  $53.75 \pm 8.41 \text{ m s}^{-2}$  for the TPU non-pneumatic wheel and  $53.70 \pm 14.12 \text{ m s}^{-2}$  for the pneumatic wheel. After Holm correction, no statistically significant differences were found between wheel types for a given obstacle size at  $3 \text{ km h}^{-1}$  (adjusted  $p > 0.05$ ).

At  $6 \text{ km h}^{-1}$ , peak accelerations increased for all obstacle sizes. For the

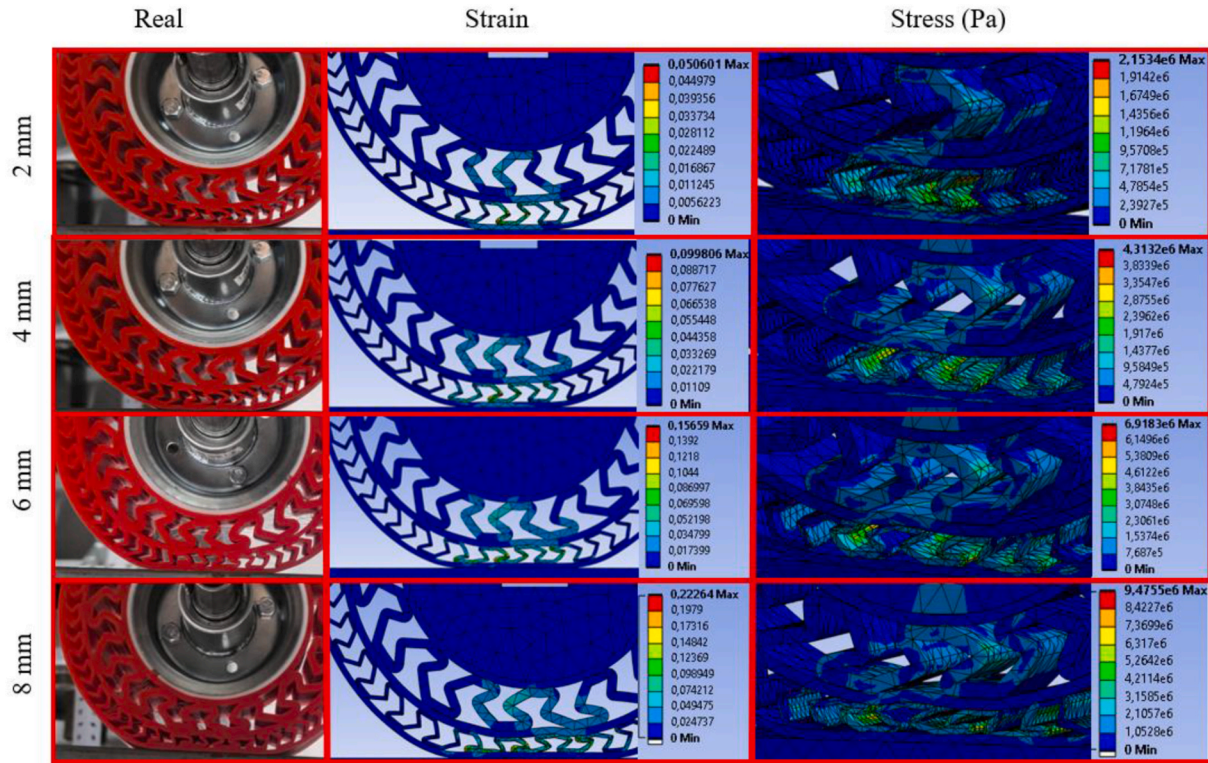


Fig. 8. Comparison between static deformation testing and its corresponding computational simulation.

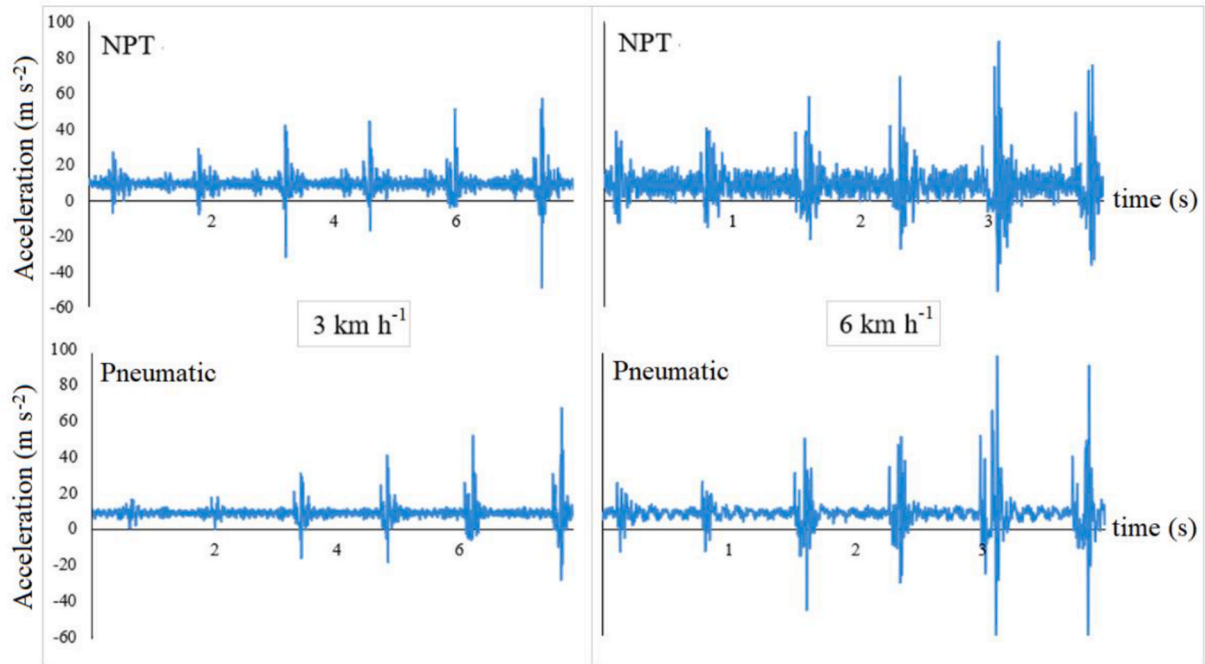


Fig. 9. Examples of accelerations reported during the obstacle traversal tests.

3 mm obstacle, the mean peak acceleration was  $36.56 \pm 5.99 \text{ m s}^{-2}$  for the TPU non-pneumatic wheel and  $28.06 \pm 3.28 \text{ m s}^{-2}$  for the pneumatic wheel. For the 6 mm obstacle, values were  $61.67 \pm 10.72 \text{ m s}^{-2}$  and  $51.67 \pm 9.93 \text{ m s}^{-2}$ , respectively. For the 9 mm obstacle, peak accelerations were again very similar between both configurations, with  $93.44 \pm 16.40 \text{ m s}^{-2}$  for the TPU non-pneumatic wheel and  $92.88 \pm 16.43 \text{ m s}^{-2}$  for the pneumatic wheel. As observed at the lower speed, no statistically significant differences were detected between

wheel types for a given obstacle size at  $6 \text{ km h}^{-1}$  after Holm correction (adjusted  $p > 0.05$ ).

Within each wheel type, obstacle size significantly affected peak acceleration at both speeds, indicating that impact severity was mainly governed by obstacle geometry and forward speed. The largest obstacle showed the strongest dependence on speed, with coefficients of determination of  $R^2 = 0.79$  for the pneumatic wheel and  $R^2 = 0.84$  for the TPU non-pneumatic wheel. Moderate correlations were observed for the 6 mm

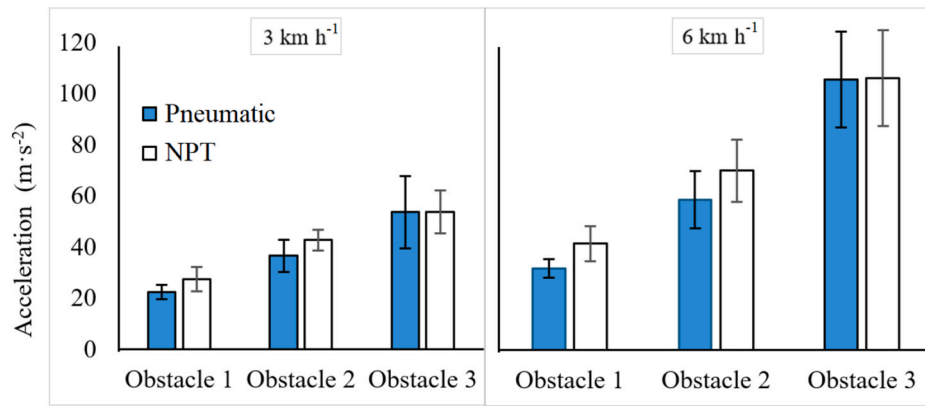


Fig. 10. Acceleration reported during the obstacle traversal tests (mean and standard deviation of 25 tests).

obstacle, with  $R^2 = 0.68$  for the pneumatic wheel and  $R^2 = 0.77$  for the TPU non-pneumatic wheel, while the 3 mm obstacle showed the weakest correlations.

Overall, the obstacle traversal results indicate that the TPU non-pneumatic wheel did not significantly reduce the maximum obstacle-induced acceleration compared with the pneumatic wheel under the tested conditions. However, it also did not produce significantly higher peak accelerations, even at the largest obstacle and highest speed. This suggests that, for the specific geometry and load evaluated, the TPU non-pneumatic wheel provided an impact response comparable to that of the conventional pneumatic solution.

### 3.3. Vibration attenuation response

Both wheel systems attenuated the acceleration transmitted from the excited base to the axle, and attenuation increased as excitation frequency increased (Fig. 11). However, the TPU non-pneumatic wheel consistently achieved greater attenuation than the pneumatic wheel under all tested excitation conditions.

At 4 Hz, the attenuation ratio was  $14.22 \pm 0.24\%$  for the TPU non-pneumatic wheel and  $4.96 \pm 0.29\%$  for the pneumatic wheel. At 8 Hz, attenuation increased to  $25.80 \pm 0.45\%$  for the TPU non-pneumatic wheel and  $18.24 \pm 2.12\%$  for the pneumatic wheel. At 15 Hz, the strongest attenuation was obtained, with  $49.48 \pm 1.53\%$  for the TPU non-pneumatic wheel and  $39.42 \pm 5.48\%$  for the pneumatic wheel. Differences between wheel types were statistically significant after Holm correction for all sinusoidal excitation frequencies ( $p < 0.05$ ).

Under broadband pink-noise excitation from 50 to 250 Hz, the TPU non-pneumatic wheel also showed higher attenuation than the pneumatic wheel. The attenuation ratio was  $38.84 \pm 0.78\%$  for the TPU non-pneumatic wheel and  $26.97 \pm 3.80\%$  for the pneumatic wheel, with statistically significant differences between configurations ( $p < 0.05$ ).

The improvement achieved by the TPU non-pneumatic wheel relative to the pneumatic wheel was consistent across the excitation range. The difference in attenuation was approximately 9.26 percentage points at 4 Hz, 7.56 percentage points at 8 Hz, 10.06 percentage points at 15 Hz and 11.87 percentage points under random excitation. These results indicate that the TPU non-pneumatic wheel behaved as a more effective vibration isolation element than the pneumatic wheel, particularly under medium-frequency and broadband excitation conditions.

Representative time-domain responses and spectral results under random excitation showed lower transmitted acceleration levels at the axle for the TPU non-pneumatic wheel (Fig. 12). The resulting spectral curves show lower transmitted amplitudes at high frequencies, with significantly lower transmitted amplitudes for the TPU solid tyre. Under broadband excitation with an input level of  $1.0 \text{ m s}^{-2}$  RMS, a clear and broad vibration isolation bandwidth is observed for the NPT, starting at 100 Hz and extending up to 250 Hz. In contrast, the conventional

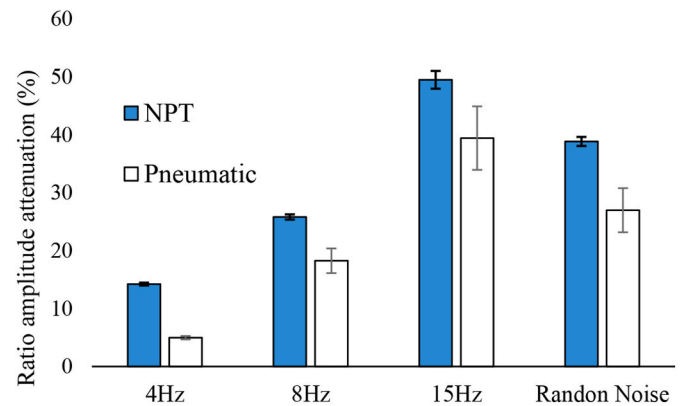


Fig. 11. Acceleration attenuation measured under harmonic and broadband excitation using the electromagnetic shaker.

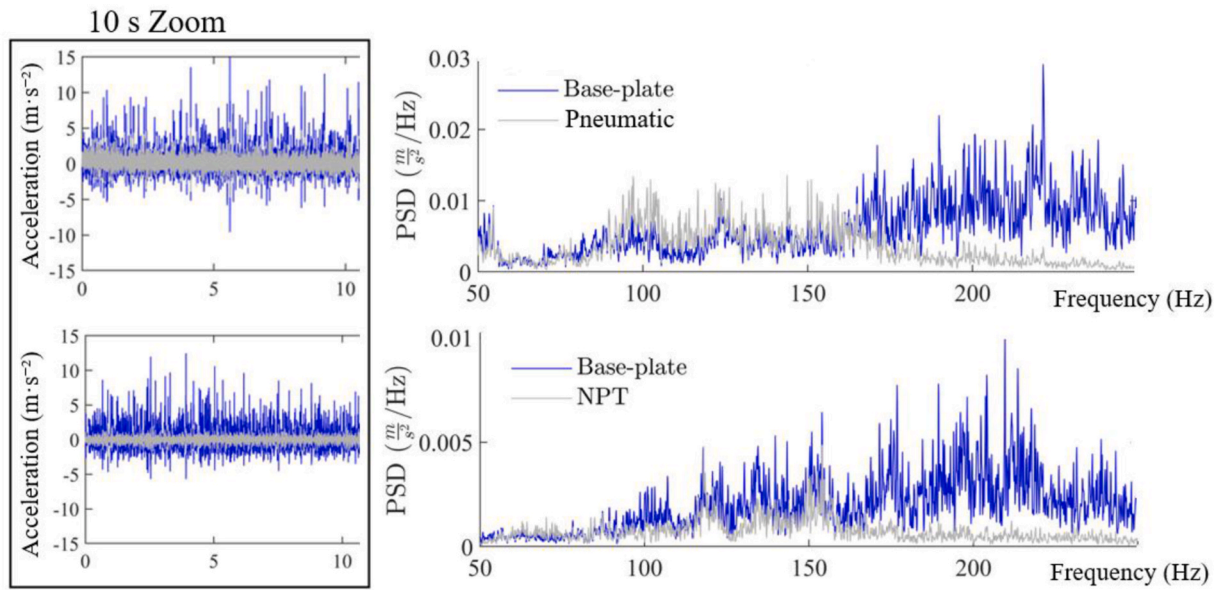
pneumatic tyre exhibits a delayed filtering response, with continuous attenuation becoming evident above approximately 170 Hz. This confirms that the improved attenuation was not limited to single-frequency sinusoidal excitation, but was also present under broadband excitation, where multiple frequency components were simultaneously applied to the wheel system.

## 4. Discussion

The results show that the TPU non-pneumatic wheel and the pneumatic wheel exhibited comparable peak acceleration responses during discrete obstacle traversal, but clearly different behaviour under continuous and broadband vibratory excitation. This distinction is important because obstacle-induced impacts and vibration isolation are governed by different dynamic mechanisms. Peak impact response is mainly affected by contact geometry, forward speed, global effective stiffness and inertia, whereas vibration attenuation depends more strongly on damping, transmissibility and frequency-dependent structural deformation.

### 4.1. Static compression behaviour and structural deformation

The static compression response of the TPU non-pneumatic wheel showed characteristic peak-then-relaxation behaviour. This is consistent with the viscoelastic nature of TPU and with time-dependent stress redistribution within the internal spoke architecture. The stabilised force increased nonlinearly with displacement, producing a progressive increase in secant stiffness from approximately  $25.0 \text{ N mm}^{-1}$  at 2 mm to  $33.8 \text{ N mm}^{-1}$  at 8 mm. This behaviour suggests that the wheel becomes



**Fig. 12.** Representative input and output acceleration responses under broadband pink-noise excitation. (For interpretation of the references to colour in this figure legend, the reader is referred to the Web version of this article.)

progressively stiffer as deformation increases, which may help maintain load-bearing capacity under larger vertical displacements.

The observed deformation pattern was also consistent with the intended design rationale of the graded V-shaped architecture. The outer region promoted bending and local deformation to optimise compliance, whereas the denser inner region contributed to load support near the hub, a mechanism that aligns with the radial density gradients recently validated by Liu et al. (2024). Furthermore, the finite element results showed stress concentration near spoke junctions and inner ring nodes. This behaviour agrees with previous studies indicating that spoke–ring transitions are critical regions in non-pneumatic tyre architectures (Genovese et al., 2021; Jin et al., 2018; Kim et al., 2024), which are also highly susceptible to localised cracks and structural fatigue under operational loads as highlighted by Deng et al. (2023). These critical zones should therefore be considered in future design iterations, for example through fillets, smoother thickness transitions, or modified spoke curvature to reduce local stress concentration.

To accurately reproduce the non-linear elasticity and large deformations of additively manufactured TPU, the five-parameter Mooney–Rivlin constitutive model was implemented, in line with established hyperelastic characterisation methodologies for material extrusion (Torrente-Prato et al., 2026). Whilst alternative higher-order formulations, such as the Yeoh model, have been shown to effectively approximate the strain energy density under specific load histories (Shuai et al., 2024), the polynomial structure of the second-order Mooney–Rivlin model ensures high numerical stability and reproduces the characteristic double curvature under moderate multiaxial deformation conditions. With regard to spatial discretisation, mesh convergence showed that the maximum equivalent stresses stabilise asymptotically from 80,000 elements onwards, with variations of less than 1.4% compared with finer meshes (Table 2). This threshold for mesh independence is fully consistent with established practice by Gao et al. (2024), who observed that geometries with curved radii require fine discretisation exceeding 80,000 elements to eliminate mesh bias in the stress fields. In contrast, whilst studies typically employ element sizes of around 6 mm with 15,000 elements to reduce computation time (Shuai et al., 2024), our choice of an average size of 5 mm (81,808 elements, SOLID187) achieves an optimal compromise.

Although the finite element model was not used here as a fully validated predictive model, it provided useful support for interpreting the

experimentally observed deformation modes. The qualitative agreement between experimental images and numerical deformation patterns indicates that the model captured the main structural response of the wheel under static compression.

#### 4.2. Obstacle traversal and impact response

The obstacle traversal tests showed that peak vertical acceleration increased with both obstacle size and forward speed. This trend was expected because larger obstacles impose greater vertical displacement demands, while higher speeds increase the dynamic severity of the contact event. For both wheel types, the highest acceleration levels were recorded at 6 km h<sup>-1</sup> when crossing the 9 mm obstacle.

No statistically significant differences were found between the TPU non-pneumatic wheel and the pneumatic wheel for a given obstacle and speed. This result indicates that, under the tested conditions, the compliant TPU structure did not significantly reduce the maximum acceleration transmitted during short-duration obstacle contact. However, it also indicates that replacing the pneumatic wheel with the TPU non-pneumatic wheel did not significantly worsen the peak impact response.

This finding can be explained by the short time scale of obstacle-induced impacts. During a rapid obstacle contact event, the response is dominated by the instantaneous deformation of the wheel, the effective stiffness of the wheel–axle system, and the geometry of the obstacle. While the internal geometry helps guide the deformation path, the viscoelastic damping mechanisms of TPU may not have sufficient time to fully dissipate enough energy to substantially reduce the peak acceleration during localized high-velocity transients. This mechanical behavior relates to the steady-state dynamic rolling evaluations conducted by Deng et al. (2024), who noted that under continuous high-speed rotation, homogeneous cellular configurations become highly sensitive to contact patch constraints and localized structural friction. Conversely, our graded V-shaped core architecture leverages radial geometry variations similar to the design principles evaluated by Liu et al. (2024) to smoothly distribute these impact boundaries across the spoke array. This geometric distribution prevents the rapid stiffness collapse or dramatic frequency disruptions traditionally caused by localized spoke failures under heavy impact testing, as warned by Deng et al. (2023). Consequently, even when obstacle size increases and both wheels enter a deformation regime

dominated by macro-geometric constraints rather than material damping, the TPU non-pneumatic tyre maintains a highly stable and predictable transient response.

The results therefore suggest that vibration isolation and peak impact mitigation should not be assumed to improve simultaneously in compliant non-pneumatic wheels. A structure that provides effective attenuation under repeated or harmonic excitation may still exhibit impact peaks comparable to a pneumatic wheel during discrete obstacle traversal. This distinction is relevant for wheel design because it implies that geometry, stiffness distribution and damping should be optimised differently depending on whether the design objective is vibration isolation, impact reduction or both.

#### 4.3. Vibration isolation performance

In contrast to the obstacle traversal results, vibration excitation tests showed a clear advantage of the TPU non-pneumatic wheel. The TPU wheel consistently achieved greater attenuation than the pneumatic wheel at 4, 8 and 15 Hz, as well as under broadband random excitation. The difference was especially relevant at 15 Hz and under random excitation, where the TPU non-pneumatic wheel reduced the transmitted RMS acceleration by approximately 49% and 39%, respectively.

This behaviour supports the interpretation of the TPU non-pneumatic wheel as a compliant dynamic filter. The combination of material damping and structural deformation in the V-shaped spoke architecture likely contributed to energy dissipation and reduced acceleration transmission from the excited base to the axle. TPU is known to exhibit viscoelastic behaviour, and its damping capacity can be enhanced when the material is arranged in deformable cellular or spoke-like architectures (Desai et al., 2023; Gosar et al., 2023; Rizzo et al., 2021; Xu et al., 2020; Y. S. Zhang et al., 2025). In the present design, the inclined spokes and graded structure may have promoted bending-dominated deformation, increasing the capacity of the wheel to attenuate vibratory inputs.

The higher attenuation observed at increasing frequency is also consistent with the frequency-dependent nature of vibration transmissibility in compliant systems. At low frequency, the wheel follows the base motion more closely and attenuation is limited. As excitation frequency increases, the structural compliance and damping of the wheel reduce the acceleration transmitted to the axle more effectively. The random excitation results confirm that this effect was not restricted to a single harmonic input but also occurred under broadband excitation.

These findings are relevant for lightweight off-road and mobile robotic applications, where vibration transmission can affect sensor stability, electronic components, structural fatigue and transported payloads. The results suggest that additively manufactured TPU non-pneumatic wheels may be useful when vibration protection and puncture-free operation are prioritised. However, the results should not be interpreted as evidence of improved traction, rolling resistance or soil performance, since these aspects were not evaluated in the present study (Angelucci et al., 2025).

#### 4.4. Engineering implications

From an engineering perspective, the main contribution of this study is the controlled comparison between a 3D-printed TPU non-pneumatic wheel and a conventional pneumatic wheel under identical loading and excitation conditions. The results indicate that the TPU non-pneumatic wheel can provide greater vibration isolation while maintaining a peak obstacle-induced acceleration response comparable to that of a pneumatic wheel.

This has practical relevance for lightweight systems operating in environments where punctures, maintenance requirements and vibration exposure are problematic. Potential applications include small off-road transport systems, agricultural robotic platforms, inspection vehicles, sensorised mobile units and lightweight trailers. In such systems, reducing transmitted vibration may help protect electronics, improve sensor measurements and reduce mechanical fatigue. At the same time,

eliminating inflation-pressure dependence may improve operational robustness.

The study also highlights the value of additive manufacturing for wheel prototyping. The internal architecture can be modified to tune stiffness, compliance and damping without changing the external wheel dimensions. Future designs could use the present results as a baseline to modify spoke thickness, spoke curvature, density gradients or tread integration. In particular, the stress concentration zones identified near spoke junctions should be considered in future geometry optimisation.

#### 4.5. Limitations and future work

The study has several limitations that should be considered when interpreting the results. First, only one vertical load level, two forward speeds and three obstacle sizes were evaluated. Therefore, the conclusions are limited to the tested lightweight operating conditions. Second, the obstacle traversal tests were performed over rigid semicircular obstacles rather than deformable soils. This approach improved repeatability and allowed the structural response of the wheels to be isolated, but it did not reproduce soil sinkage, soil deformation, traction or rolling resistance. Third, the TPU non-pneumatic wheel did not include a dedicated tread layer. In real applications, the tread plays an important role in contact mechanics, wear, grip and pressure distribution. The absence of a differentiated tread region may affect durability and field performance. Fourth, fatigue, long-term cyclic loading, thermal effects, creep and wear were not evaluated, even though these are critical issues for non-pneumatic wheel designs (Sardinha et al., 2023; Sun et al., 2024a,b; Żmuda and Jackowski, 2025). Finally, the finite element model was used to support interpretation of deformation patterns under static compression, but dynamic numerical modelling and quantitative validation were outside the scope of this study.

Future work should evaluate the non-pneumatic tyre under a broader range of loads, speeds and representative road or off-road conditions, as well as during longer-term cyclic tests. Rolling resistance, contact-patch evolution, tread integration, wear, thermal behaviour and structural durability should also be investigated. Since only one specimen of each tyre configuration was tested, the repeated measurements quantify within-specimen variability but not manufacturing reproducibility. The present results should therefore be interpreted as a controlled proof-of-concept comparison. Future campaigns should test at least three nominally identical tyres manufactured in independent printing batches to assess tyre-to-tyre and batch-to-batch variability. Material compression-set tests and an appropriately scaled plunger-energy procedure could also be used to characterise permanent deformation, energy absorption and structural failure. Any modification of ASTM F414 should be reported as an adapted protocol rather than as direct compliance with the standard.

The numerical model is also limited to static analysis using a hyperelastic formulation and cannot reproduce the time- and frequency-dependent viscoelastic response of TPU. Future work should incorporate experimentally calibrated viscoelastic properties into dynamic simulations and validate them against measured transmissibility and impact responses. These models could then support optimisation of the spoke architecture to balance vibration isolation, impact mitigation and structural durability.

## 5. Conclusions

This study presented a controlled experimental and numerical assessment of a 3D-printed TPU non-pneumatic wheel under static compression, obstacle traversal and controlled vibratory excitation. The wheel was compared with a conventional pneumatic wheel under equivalent vertical load and boundary conditions.

The static compression tests showed a time-dependent response with force relaxation after peak loading, consistent with the viscoelastic behaviour of TPU and progressive stress redistribution within the internal spoke structure. Stabilised force values indicated a nonlinear increase in secant stiffness with displacement, from approximately  $25.0 \text{ N mm}^{-1}$  at

2 mm to 33.8 N mm<sup>-1</sup> at 8 mm. Finite element modelling supported the interpretation of the observed deformation modes and identified stress concentration zones near spoke junctions and inner ring nodes.

Obstacle traversal tests showed that peak vertical acceleration increased with obstacle size and forward speed for both wheel types. However, no statistically significant differences were found between the TPU non-pneumatic wheel and the pneumatic wheel for a given obstacle size and speed under the tested conditions. This indicates that the TPU non-pneumatic wheel provided an impact response comparable to that of the pneumatic wheel during discrete obstacle traversal.

In contrast, vibration excitation tests showed that the TPU non-pneumatic wheel consistently achieved higher vibration attenuation than the pneumatic wheel. Attenuation values for the TPU wheel were 14.22%, 25.80% and 49.48% at 4, 8 and 15 Hz, respectively, compared with 4.96%, 18.24% and 39.42% for the pneumatic wheel. Under broadband random excitation from 50 to 250 Hz, attenuation reached 38.84% for the TPU non-pneumatic wheel and 26.97% for the pneumatic wheel.

Overall, the tested TPU non-pneumatic tyre achieved greater vibration attenuation than the pneumatic reference specimen under the evaluated harmonic and broadband excitation conditions. No statistically significant difference in peak obstacle-induced acceleration was detected between the two specimens during obstacle traversal. These findings support the potential of the proposed architecture for vibration isolation in lightweight off-road, robotic and agricultural applications. However, independently manufactured specimens, durability testing, tread integration, rolling-resistance measurements and experiments under representative soil conditions are required before the results can be generalised to other tyres or operating conditions.

## CRediT authorship contribution statement

**Pedro Sánchez-Cachinero:** Writing – original draft, Formal analysis. **Matteo Sbaglia:** Investigation. **Bruno Bernardi:** Writing – review & editing. **Rafael R. Sola-Guirado:** Writing – review & editing, Project administration, Methodology, Conceptualization.

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## Declaration of competing interest

The authors declare the following financial interests/personal relationships which may be considered as potential competing interests: Rafael R. Sola Guirado reports financial support was provided by University of Cordoba. If there are other authors, they declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

## Data availability

Data will be made available on request.

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