

Received 22 June 2026, accepted 5 July 2026, date of publication 10 July 2026, date of current version 17 July 2026.

Digital Object Identifier 10.1109/ACCESS.2026.3712011

RESEARCH ARTICLE

Novel Holonomic Driving Wheel Mechanism for Omnidirectional Motion in Two-Wheeled Robots

JEEHO WON¹, CHANJU LEE², INHA PARK¹, MYEONGJIN CHOI¹, SANGHYUN CHO², CHEOLGI LYOO², DONGWOO SUH², SANGWOO HWANG³, GUNWOO NOH^{1,2}, (Member, IEEE), AND TAEWON SEO¹, (Senior Member, IEEE)

¹Department of Mechanical Engineering, Hanyang University, Seoul 04763, Republic of Korea

²Department of Mechanical Engineering, Korea University, Seoul 02841, Republic of Korea

³Advanced Chassis Development Team, Hyundai Motor Group, Hwasung, Gyeonggi-do 18278, Republic of Korea

Corresponding authors: Gunwoo Noh (gunwoonoh@korea.ac.kr) and Taewon Seo (taewonso@hanyang.ac.kr)

This research was supported by the “Regional Innovation System & Education (RISE)” through the Seoul RISE Center, funded by the Ministry of Education (MOE) and the Seoul Metropolitan Government (2026-RISE-01-027-04) and Hyundai Motor Group.

Jeeho Won and Chanju Lee contributed equally to this work.

ABSTRACT This work presents a robotic implementation of a holonomic wheel mechanism that achieves omnidirectional motion through a compact dual-input differential drive. A single-sided differential gear assembly combined with a compliant toroidal tire is introduced as the core mechanical configuration. The tire functions as a deformable, coreless structure actuated through a roller-pinion interface that preserves the internal actuation path under tire deformation. The kinematic relationships governing the dual-input mechanism are formulated, and a velocity-control-based coupled response model is introduced to describe the motor-side behavior under commanded rolling, lateral, and diagonal motion. A prototype two-wheeled platform, consisting of one active and one passive omnidirectional wheel, is evaluated through trajectory-level motion tests and motor-side velocity/current measurements. The results demonstrate the feasibility of the proposed mechanism and characterize the coupled response of the two input axes without claiming transmission efficiency or traction superiority over other active omnidirectional wheel mechanisms.

INDEX TERMS Holonomic drive, active omnidirectional wheel, differential mechanism, compliant torus tire, omnidirectional robot.

I. INTRODUCTION

Robotics has increasingly advanced in the areas of mobility and maneuverability, particularly within omnidirectional motion environments [1], [2], [3]. Traditional mechanisms, including Mecanum and omni wheels, allow a wide range of planar motion through passive rollers or slanted treads [4], [5]. Although these mechanisms are widely used, their motion capability depends strongly on the roller arrangement, contact condition, load distribution, and the way in which lateral force is generated [6].

Numerous surveys have assessed the characteristics and limitations of omnidirectional wheel systems [7], [8], [9], [10]. Traditional three- or four-wheel platforms equipped with passive omnidirectional wheels (POWs) have been reported to suffer from transmission loss, roller wear, limited payload capacity, and instability under varying load

conditions, depending on their roller geometry and contact configuration. These limitations are especially relevant when lateral motion is produced through passive rolling or frictional interaction rather than by directly actuating the lateral direction within each wheel unit. Active omnidirectional wheels (AOWs) were developed to address this structural limitation by introducing active longitudinal and lateral actuation within a wheel unit [11], [12], [13], [14], [15], [16], [17]. Such mechanisms increase the available actuation authority for compact holonomic platforms, but they also introduce mechanical packaging, transmission, and coupled-control challenges. Industrial prototypes such as Honda's omnidirectional personal mobility device further demonstrate the practical interest in AOW mechanisms, while also highlighting the importance of compact and mechanically robust wheel integration [18].

Nevertheless, many dual-input AOW mechanisms require actuation or transmission elements on both sides of the wheel, which increases the radial footprint and mechanical

The associate editor coordinating the review of this manuscript and approving it for publication was Giulio Reina¹.

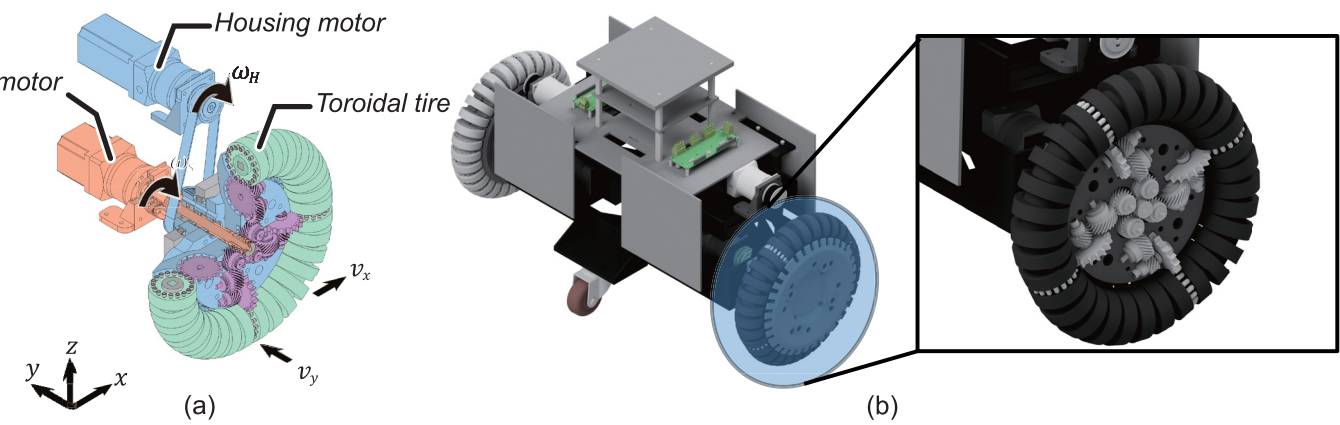


FIGURE 1. (a) Sketch of the proposed AOW. (b) Prototype two-wheeled robot platform with one AOW and one POW.

TABLE 1. Qualitative comparison of representative active omnidirectional wheel mechanisms (“+” = clearly present, “-” = weak / absent).

	Komori-type differential AOW [11]	Terakawa seamless AOW [12]	Liddiard wheel [19]	Honda HOT (U3-X type) [18]	OmBURo AOW [13]	Proposed wheel (this work)
Representative image						
Direct lateral drive	+	-	-	-	+	+
Compliant / deformable tire	-	-	+	-	-	+
Single-sided input	+	-	+	-	-	+
Simple drivetrain	-	+	-	-	-	-
Low internal complexity	-	-	+	-	+	-

complexity for compact integration. To reduce this footprint, Liddiard designed a single-sided AOW with an internal motor mounted on the wheel hub [19]. This design provides a compact layout, but its lateral actuation is generated through frictional engagement inside the drive mechanism. As a result, the lateral response can be affected by internal drive slip as well as by the actual tire-ground contact condition, which can reduce motion accuracy under changing surface conditions. Its rigid tire geometry also limits compliance and shock absorption when compared with deformable wheel structures.

In light of these restrictions, this study introduces a single-sided dual-input holonomic wheel that combines a differential gear transmission and a compliant toroidal tire. The concept is based on two recent patent disclosures by Hwang [20], [21], where a compact one-sided dual-input wheel architecture and a toroidal airless tire structure were proposed for omnidirectional motion. Building on this

concept, the present implementation introduces a roller-pinion interface to preserve a mechanically constrained actuation path between the internal drive and the deformable tire. Both input drives are arranged on one side of the hub, while the outer wheel geometry remains compatible with a compact wheel-mount configuration. Although the added belt-pulley and differential transmission increase drivetrain complexity, the layout allows the wheel to be integrated into a pre-existing mobile platform without requiring a symmetric two-sided actuation structure.

Table 1 summarizes representative AOW mechanisms compared with the proposed wheel, where “+” indicates that the feature is clearly present or easy to achieve, and “-” indicates that it is absent, weak, or difficult to obtain. Existing AOWs with rigid rollers or module arrays can provide active omnidirectional motion, but they differ in terms of compliance, lateral-drive interface, packaging, and drivetrain complexity. In contrast, the proposed wheel combines a

compliant toroidal tire, a mechanically constrained lateral-drive interface, and a single-sided input arrangement. This combination is obtained at the expense of a more complex internal drivetrain and stronger coupling between the two input axes, which motivates the velocity-control-based modeling and motor-side response characterization in this study.

The proposed wheel and the two-wheeled prototype platform are shown in Fig. 1. To evaluate this configuration, the mechanism is analyzed from its kinematic velocity relationship and then modeled as a coupled motor-side velocity-control system. Because the wheel is driven by a differential mechanism, rolling and lateral motion are generated by the common and relative velocity components of the two input axes. Therefore, the revised model focuses on commanded motor velocity, measured motor velocity, and q-axis current response rather than direct transmission-efficiency or traction-force estimation. The concept is examined through prototype construction, velocity-control simulation, motor-side response experiments, and trajectory-level tests performed on a two-wheeled platform consisting of one AOW and one POW.

II. HOLONOMIC WHEEL MECHANISM

The proposed AOW consists of two mechanically coupled subsystems: a dual-input differential transmission that generates orthogonal velocity components and a compliant toroidal tire that converts them into planar motion. A schematic of the overall structure is shown in Fig. 2, illustrating the torque and motion flow from the two motor inputs to the tire-ground interface. The single-sided configuration was intentionally selected to ensure compatibility with conventional wheel-mount platforms. In contrast to symmetric dual-input AOWs that require motors protruding from both sides, the proposed design holds both drives within a single side of the hub and has the same mounting interface as an ordinary wheel. Although it requires an additional belt-pulley stage, the layout reduces the radial footprint of the wheel and allows retrofitting to a pre-existing platform without structural modification.

For clarity, the two input actuators are hereafter referred to as the *housing drive* and *sun drive*, which correspond to the outer wheel housing-connected and inner sun gear-connected shafts of the differential, respectively. Their angular velocities and torques are denoted by ω_H and ω_S , and τ_H and τ_S , respectively. The resulting wheel motions along the longitudinal and lateral directions are termed the *roll channel* and *lateral channel*, with corresponding velocities v_x and v_y , respectively.

A. STRUCTURE AND TRANSMISSION PATH

The two brushless direct current (BLDC) motors provide input angular velocities ω_H and ω_S from one side of the wheel. The housing drive is associated with the main wheel rotation and generates the roll-channel motion through the hub-side transmission path. The sun drive is coupled to the internal differential and roller-pinion path, such that its

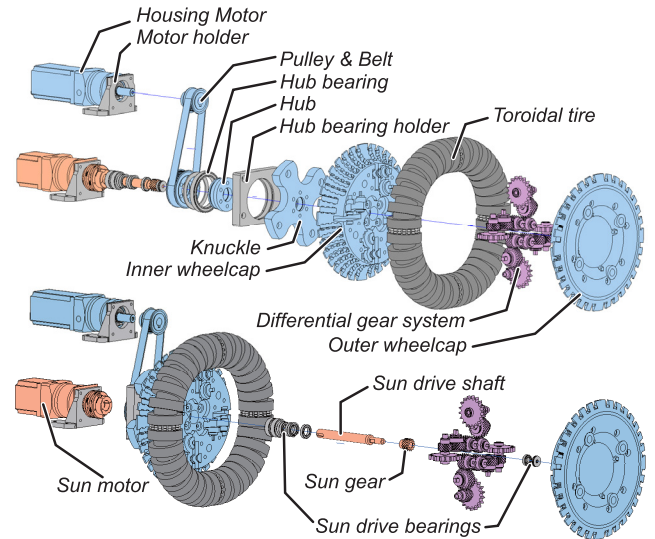


FIGURE 2. Expanded view of the proposed AOW. The single-sided configuration integrates two BLDC motors, a belt-pulley reduction, 45° helical differential gear set, and compliant toroidal tire. The layout shows how the sun and housing drives are positioned within one side of the hub to minimize the radial footprint while enabling independent rolling and lateral actuation through differential and roller-pinion interfaces.

motion relative to the housing-side input produces lateral-channel actuation. Both inputs are coupled through a compact differential stage to enable bidirectional, independent control of the rolling and lateral movement.

The internal gear set consists of 45° helical gears arranged in a single plane. The helical configuration provides smoother meshing than spur gears and enables the axial redirection of the torque within the limited gear hub space. This gear setup also introduces the geometric twist necessary to connect the differential mechanism with the lateral roller-pinion interface inside the tire, ensuring direct transmission while maintaining compactness.

The helical gears are S45C steel, with the shafts being SUS304 stainless steel. The wheel cap/gear housing was 3D printed using polylactic acid (PLA) plastic, as well as the pinion gear for the roller-pinion interface.

The relative geometry of the gears, pulleys, and shafts was determined to satisfy spatial constraints and minimize inertia while maintaining manufacturability.

The key transmission parameters of the final mechanism are listed in Table 2.

The roll- and lateral-channel inertia values were obtained from CAD mass-property evaluation of the rotating wheel components and are expressed in the rotational channel coordinates used for the response simulation. The fixed external motor reductions are omitted from the kinematic notation and are reflected in the effective motor-side parameters used in the simulation.

B. KINEMATIC ANALYSIS

The model uses three coordinate spaces: the workspace velocity, the rotational equivalent workspace velocity, and the

TABLE 2. Main design parameters of the differential mechanism.

Parameter	Symbol	Value
Rolling radius	R	150 mm
Lateral effective radius	r	30 mm
Helix angle	β	45°
Internal lateral gear ratio	n	15/13
Total mass	m	2.7 kg
Roll-channel inertia	J_{roll}	$3.559 \times 10^{-2} \text{ kg m}^2$
Lateral-channel inertia	J_{lat}	$9.96 \times 10^{-4} \text{ kg m}^2$
Housing-side effective K_t	$K_{t,H}^{\text{eff}}$	0.4032 N m/A
Sun-side effective K_t	$K_{t,S}^{\text{eff}}$	0.4877 N m/A

modeled input-axis velocity. These coordinates are defined as

$$\mathbf{v} = \begin{bmatrix} v_x \\ v_y \end{bmatrix} \quad \omega_{\text{ch}} = \begin{bmatrix} \omega_{\text{roll}} \\ \omega_{\text{lat}} \end{bmatrix} \quad \omega = \begin{bmatrix} \omega_H \\ \omega_S \end{bmatrix} \quad (1)$$

The radius conversion from rotational channel velocity to workspace velocity is written as

$$\mathbf{v} = D_r \omega_{\text{ch}}, \quad D_r = \begin{bmatrix} R & 0 \\ 0 & r \end{bmatrix} \quad (2)$$

The differential mechanism maps the input-axis velocities to the roll and lateral rotational channels as

$$\omega_{\text{ch}} = A\omega, \quad A = \begin{bmatrix} 1 & 0 \\ n & -n \end{bmatrix} \quad (3)$$

or equivalently,

$$\begin{bmatrix} \omega_{\text{roll}} \\ \omega_{\text{lat}} \end{bmatrix} = \begin{bmatrix} 1 & 0 \\ n & -n \end{bmatrix} \begin{bmatrix} \omega_H \\ \omega_S \end{bmatrix} \quad (4)$$

Therefore, the full kinematic relationship governing the wheel motion is expressed as

$$\mathbf{v} = D_r A \omega = G\omega, \quad (5)$$

where

$$G = D_r A = \begin{bmatrix} R & 0 \\ nr & -nr \end{bmatrix} \quad (6)$$

Thus,

$$\begin{bmatrix} v_x \\ v_y \end{bmatrix} = \begin{bmatrix} R & 0 \\ nr & -nr \end{bmatrix} \begin{bmatrix} \omega_H \\ \omega_S \end{bmatrix} \quad (7)$$

This mapping is used to convert the desired platform velocity into the motor velocity references used in the velocity-control model.

When $\omega_H = \omega_S$, the wheel performs pure rolling ($v_y = 0$); lateral motion is generated by the differential velocity component between the two inputs. The inverse kinematics are expressed as

$$\omega = G^{-1}\mathbf{v}, \quad (8)$$

or

$$\begin{bmatrix} \omega_H \\ \omega_S \end{bmatrix} = \begin{bmatrix} \frac{1}{R} & 0 \\ \frac{1}{R} & -\frac{1}{nr} \end{bmatrix} \begin{bmatrix} v_x \\ v_y \end{bmatrix} \quad (9)$$

Only the internal gear ratio n is retained in this mapping. The external pulley or motor gearboxes are omitted for clarity because their effects are reflected in the effective parameters used in the response model. The corresponding roll and lateral channel definitions are illustrated in Fig. 3.

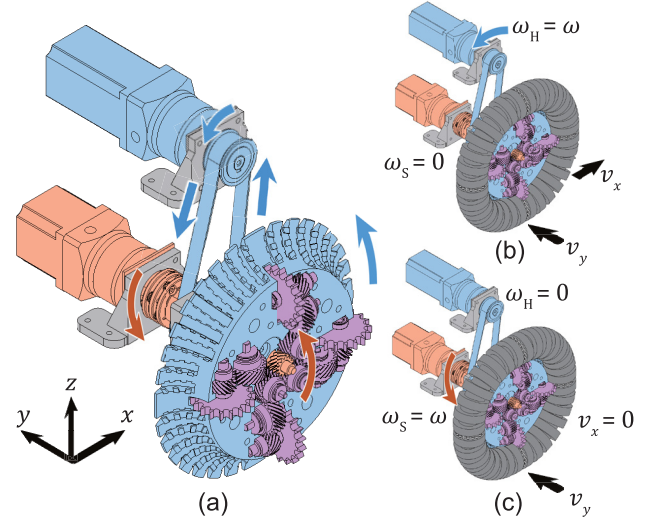


FIGURE 3. Kinematic representation of the proposed holonomic wheel. The roll channel and lateral channel are defined by the rotational equivalent coordinates ω_{roll} and ω_{lat} , while ω_H and ω_S denote the housing- and sun-side input-axis velocities.

C. VELOCITY-CONTROL-BASED COUPLED RESPONSE MODEL

The wheel is driven through a differential mechanism; therefore, the roll and lateral channels are commanded by the common and relative velocity components of the two input axes. In the prototype, the two motors are not commanded by direct torque input but by motor velocity references. The model used in this study is therefore centered on motor-side velocity response, q-axis current response, and disturbance loads acting on the commanded velocity loop.

1) DIFFERENTIAL VELOCITY COORDINATES

Using the transform A in (3), the roll and lateral rotational components are defined as

$$\begin{bmatrix} \omega_{\text{roll}} \\ \omega_{\text{lat}} \end{bmatrix} = A \begin{bmatrix} \omega_H \\ \omega_S \end{bmatrix} = \begin{bmatrix} \omega_H \\ n(\omega_H - \omega_S) \end{bmatrix} \quad (10)$$

Here, ω_{roll} denotes the rolling component, and ω_{lat} denotes the lateral rotational component that produces lateral actuation, i.e., the rotational equivalent of the lateral velocities.

2) EMBEDDED VELOCITY CONTROLLER REPRESENTATION

For each motor axis, the velocity tracking error is defined as

$$e_i(t) = \omega_i^{\text{ref}}(t) - \omega_i(t), \quad i \in \{H, S\}, \quad (11)$$

where ω_i^{ref} and ω_i are the commanded and measured input-axis angular velocities, respectively. The embedded drive

controller is represented by a nominal PID-type velocity loop,

$$\dot{i}_{q,i}^{\text{ref}}(t) = K_{p,i}e_i(t) + K_{i,i} \int e_i(t) dt + K_{d,i}\dot{e}_i(t), \quad i \in \{H, S\}. \quad (12)$$

The gains in (12) are treated as effective controller parameters corresponding to the configured drive behavior. If derivative action is not used in the simulation, $K_{d,i}$ is set to zero. The current loop is assumed to be faster than the velocity response considered here, and the q-axis current is used as the measured motor-side effort signal. The motor torque is written in vector form as

$$\tau_m = K_t^{\text{eff}} \mathbf{i}_q, \quad (13)$$

where

$$\tau_m = \begin{bmatrix} \tau_{m,H} \\ \tau_{m,S} \end{bmatrix} \quad \mathbf{i}_q = \begin{bmatrix} i_{q,H} \\ i_{q,S} \end{bmatrix} \quad K_t^{\text{eff}} = \begin{bmatrix} K_{t,H}^{\text{eff}} & 0 \\ 0 & K_{t,S}^{\text{eff}} \end{bmatrix} \quad (14)$$

Here, K_t^{eff} is the effective torque-constant matrix reflected to the modeled input-axis coordinate.

3) MOTOR-SIDE DYNAMICS AND DISTURBANCE LOADS

The CAD-derived channel inertias are written as

$$J_{\text{ch}} = \begin{bmatrix} J_{\text{roll}} & 0 \\ 0 & J_{\text{lat}} \end{bmatrix} \quad (15)$$

Because the response model is written in the input-axis space, the corresponding inertia matrix is obtained by the power-consistent coordinate reflection

$$J_m = A^T J_{\text{ch}} A. \quad (16)$$

Equivalently,

$$J_m = \begin{bmatrix} J_{\text{roll}} + n^2 J_{\text{lat}} & -n^2 J_{\text{lat}} \\ -n^2 J_{\text{lat}} & n^2 J_{\text{lat}} \end{bmatrix} \quad (17)$$

The motor-side dynamics are written as

$$J_m \dot{\omega} = K_t^{\text{eff}} \mathbf{i}_q - B_m \omega - \tau_d, \quad (18)$$

where J_m and B_m are the effective input-axis inertia and residual motor-side viscous coefficient matrices, and τ_d is the disturbance load reflected to the input axes. The term $B_m \omega$ represents residual input-axis losses, whereas the roll and lateral resistance terms below represent load resistance expressed in the channel coordinates. These coefficients are treated as lumped effective parameters and are not separated into independently measured physical components in this study.

The disturbance terms are defined in the roll/lateral channel coordinates and then mapped back to the housing and sun axes. The channel disturbance vector is written as

$$\tau_{\text{ch}} = \begin{bmatrix} \tau_{\text{roll}} \\ \tau_{\text{lat}} \end{bmatrix} \quad (19)$$

The roll-channel resistance is modeled as

$$\tau_{\text{roll}} = B_{\text{roll}} \omega_{\text{roll}} + \tau_{c,\text{roll}} \tanh\left(\frac{\omega_{\text{roll}}}{\epsilon_{\text{roll}}}\right), \quad (20)$$

and the lateral-channel resistance is modeled as

$$\tau_{\text{lat}} = B_{\text{lat}} \omega_{\text{lat}} + \tau_{c,\text{lat}} \tanh\left(\frac{\omega_{\text{lat}}}{\epsilon_{\text{lat}}}\right) + \tau_p(t). \quad (21)$$

The Coulomb component is first represented conceptually by a sign function,

$$\tau_c(\omega) = \tau_{c,0} \text{sgn}(\omega). \quad (22)$$

For numerical simulation, the discontinuous sign function is replaced by a smooth hyperbolic tangent approximation,

$$\text{sgn}(\omega) \approx \tanh\left(\frac{\omega}{\epsilon}\right), \quad (23)$$

where ϵ is a small smoothing parameter near zero velocity.

The periodic component is defined as a relative-position-periodic lateral disturbance. Its phase is modeled from the relative motion between the housing and sun input axes as

$$\dot{\theta}_p(t) = 2\pi \frac{|\omega_H^{\text{ref}}(t) - \omega_S^{\text{ref}}(t)|}{N_p}, \quad \theta_p(0) = 2\pi \phi_p. \quad (24)$$

The periodic resistance term is then modeled as

$$\tau_p(t) = A_p \sin \theta_p(t), \quad (25)$$

where A_p , N_p , and ϕ_p are fitted disturbance parameters representing the amplitude, relative input-axis displacement corresponding to one disturbance period, and phase of the periodic resistance term, respectively. This term represents the repeated resistance fluctuation observed from the segmented tire and roller-pinion contact pattern in the off-ground experimental setup. Because it depends on the relative motion between the two input axes rather than time directly, its apparent temporal frequency changes with the commanded relative velocity. The resistance and periodic disturbance coefficients are calibrated as effective prototype-level values for reproducing the off-ground motor-side response; they are not interpreted as independently identified tire-force constants or as a general tire-force model.

Using the velocity-coordinate transform in (10), the disturbance loads are mapped to the input axes as

$$\tau_d = A^T \tau_{\text{ch}}. \quad (26)$$

That is,

$$\begin{bmatrix} \tau_{d,H} \\ \tau_{d,S} \end{bmatrix} = \begin{bmatrix} 1 & n \\ 0 & -n \end{bmatrix} \begin{bmatrix} \tau_{\text{roll}} \\ \tau_{\text{lat}} \end{bmatrix} \quad (27)$$

This mapping describes why resistance in the relative/lateral channel can appear in both input axes, even when the kinematic command is assigned to one channel.

To illustrate the effect of these disturbance terms, Fig. 4 compares the simulated response of the same lateral velocity command with and without the disturbance model. The disturbance-free case represents the nominal embedded velocity-loop response, while the disturbance-included case

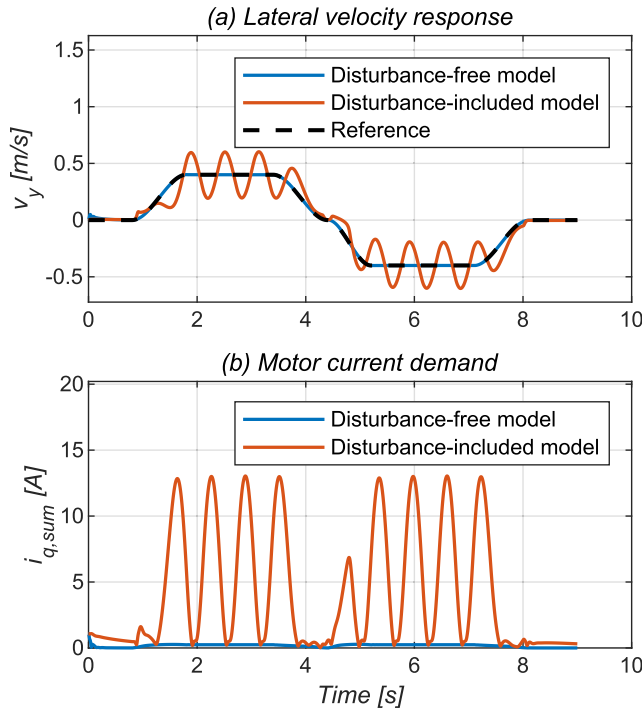


FIGURE 4. Effect of disturbance terms in the velocity-control-based response model. The disturbance-free model represents the nominal motor-side velocity response, whereas the disturbance-included model adds viscous resistance, smoothed Coulomb friction, and a relative-position-periodic lateral disturbance term. The comparison illustrates how the added disturbance terms affect lateral velocity tracking and motor current demand.

applies the roll and lateral-channel resistance terms defined above.

As shown in Fig. 4, the disturbance-included model remains bounded under the commanded lateral velocity but produces periodic tracking deviation and increased motor current demand. Therefore, the disturbance terms are not introduced as an independent tire-force model, but as a compact representation of the load effects that influence the motor-side velocity and current response of the prototype.

III. HOLONOMIC TIRE SYSTEM

A. NONPNEUMATIC HOLONOMIC TIRE DESIGN

A novel nonpneumatic tire was developed to accommodate the proposed holonomic mechanism. The tire design aims to address four concurrent requirements: (1) sufficient structural compliance to allow bending during assembly onto the holonomic wheel, (2) adequate stiffness and traction to sustain payload-induced compression, (3) compatibility with the roller-pinion drive interface, and (4) transmission of actuation forces from the roller-pinion interface throughout the tire.

To satisfy these requirements, a segmented toroidal cellular architecture constructed from Bambu Lab TPU95A HF was adopted. The proposed tire for the holonomic platform consists of four segments, as illustrated in Fig. 5. Each segment comprises three types of cells designated as Cell A, Cell B, and Cell C, where Cells A and B were alternately

arranged between the terminal Cell C structures at both ends of each segment.

Cell A employed a strut-core structure to provide a continuous actuation path in both the longitudinal and lateral directions while maintaining flexural compliance. Cell B features a Bezier curve-based profile derived from [22] to provide compressive load support and stable ground contact. Cell C incorporated circumferentially arranged roller pockets for roller coupling, and central spigot-socket joints for inter-segmental connection and axial alignment. Cell C adopts the same Bezier-based profile as Cell B and features an outer arc curvature designed to achieve circular continuity upon assembly.

This design and its parameters were determined through computational design and multi-objective optimization, and a detailed tire-focused study is being conducted separately.

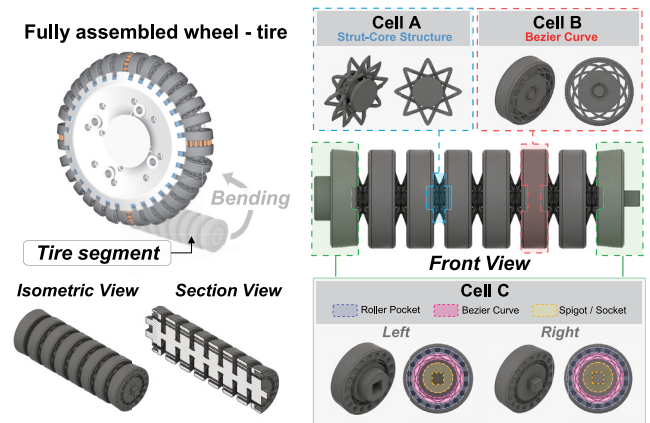


FIGURE 5. Segmented toroidal cellular tire architecture. Each segment consists of three cell types: Cell A with a strut-core structure (blue highlight), Cell B with a Bezier curve-based profile (red highlight), and Cell C featuring roller pockets, spigot-socket joints, and a Bezier curve-based profile (green highlight).

B. TIRE-WHEEL COUPLING MECHANISM

The tire was assembled by bending along the curved outer frame of the wheel, with mechanical coupling achieved through two types of rollers, as illustrated in Fig. 6. Drive rollers, positioned at both ends of each segment, are inserted into the roller pockets of Cell C (Fig. 6(a)). These rollers meshed with pinion gears mounted inside the wheel frame (Fig. 6(b)), enabling torque transmission from the wheel to the tire. This interface is designed to maintain positive mechanical engagement under local tire deformation. Unlike friction-only drive contact, the roller-pinion geometry provides a constrained actuation path without introducing a rigid tire core. With this design, the tire retains compliant, shock-absorbing characteristics while avoiding any rigid mesh or core design within the tire.

Support rollers were distributed along the remaining circumference (Fig. 6(c)), maintaining continuous contact with the wheel-facing surface of the tire. They rotate freely to reduce sliding resistance and do not transmit active torque. Instead, they provide structural support and transfer vertical

loads to the wheel. With the drive and support rollers, the tire has the potential to deform easily and still be driven both longitudinally and laterally.

To transfer the lateral torque from the internal differential to the compliant tire, a roller-pinion interface was used. This series of small rollers was located in the lateral direction of the toroidal tire and contacted the pinion gear driven through the differential lateral actuation path. The cores of the rollers are stainless steel, and each roller uses a self-lubricating polymer plain bearing (Igus iglidur[®] series) to reduce friction and wear at the roller shafts. The pinion gear uses an epicycloidal profile and was 3D printed in PLA plastic. This configuration was selected to distribute contact through rolling elements and to preserve the tire's deformability while providing a positive lateral actuation path.

A simplified roller-pinion feasibility analysis was performed to check whether the commanded pinion rotation produces comparable lateral displacement under representative normal loads. The simulation environment is shown in Fig. 7(a), where a single roller-pinion contact pair is loaded with a given force. The pinion gear is rotated by 90°, and the lateral travel of the roller counterpart is recorded relative to the nominal geometric displacement. The results in Fig. 7(b) show that the realized lateral travel remained close to the nominal geometric value under the tested load range, supporting the geometric compatibility of the roller-pinion interface with the compliant tire.

The present manuscript uses this analysis to support the structural compatibility of the roller-pinion actuation path with the compliant tire. Full tire-ground contact behavior, TPU viscoelastic characterization, and long-term wear behavior are not quantified in this manuscript and are reserved for the separate tire-focused study.

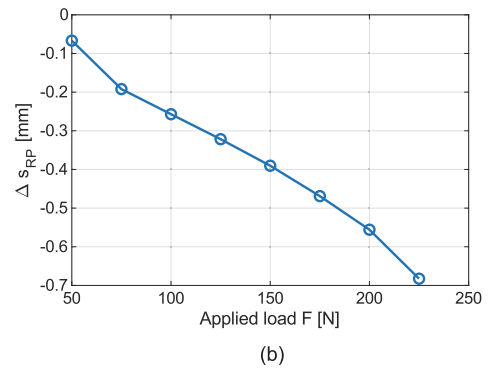
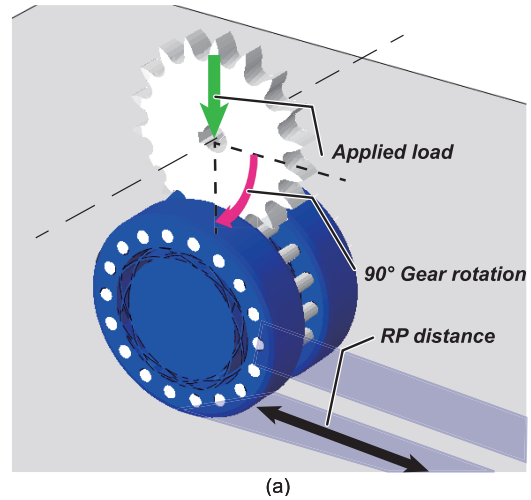


FIGURE 7. Roller-pinion gear feasibility simulation. (a) Simplified setup: a normal load F is applied to the pinion axis while the pinion is rotated by 90° against the compliant tire. (b) Signed deviation of the realized lateral travel $\Delta s_{RP} = s_{RP} - s_{nom}$ as a function of the applied load. Negative values indicate slightly shorter travel than the nominal geometric value $s_{nom} = 30\pi/2$ mm.

IV. PROTOTYPE AND EXPERIMENTS

The prototype evaluation was arranged in two stages. First, the AOW was tested in a suspended condition to compare the motor-side simulation response with measured motor velocities and q-axis currents before adding ground-contact effects. Second, the complete AOW-POW platform was tested on the ground to confirm that the wheel mechanism generates omnidirectional platform motion. This order separates the motor-side response confirmation from the platform-level trajectory demonstration, since the velocity-control-based response model in Section II-C does not include tire-ground slip or time-varying normal contact force.

A. PROTOTYPE CONFIGURATION

A prototype two-wheeled platform was constructed to evaluate the proposed wheel mechanism (Fig. 8). The left wheel is the AOW, and the right wheel is a POW of identical geometry. The POW employed the same TPU tire segments to maintain comparable tire construction. The AOW provides independent rolling and lateral degrees of freedom (DOF), and the POW provides an additional rolling DOF. Together, these three controllable DOFs span the full planar motion

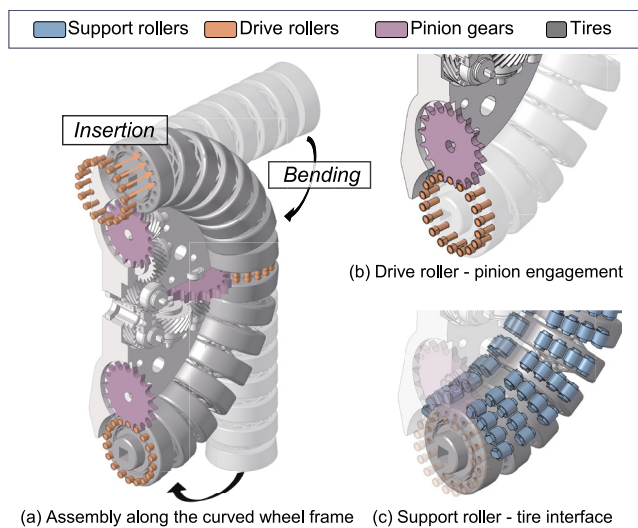


FIGURE 6. Schematic of wheel-tire coupling mechanism. (a) Tire assembly along the curved wheel frame, showing roller insertion and bending during installation. (b) Drive roller-pinion engagement enabling torque transmission from wheel to tire. (c) Support roller-tire interface for load transfer.

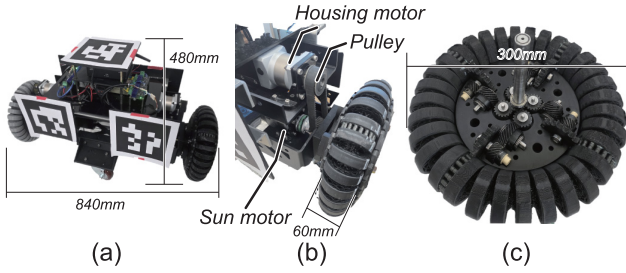


FIGURE 8. Prototype of the omnidirectional robot platform: (a) assembled robot with AOW-POW configuration, (b) enlarged view of the AOW showing the compact single-sided drive, and (c) internal differential assembly.

(x, y, yaw) , making the robot kinematically holonomic. Both wheel inputs were driven by BLDC motors through belt-pulley reductions, controlled via ODrive v3.6 boards and an NVIDIA Jetson Nano. The two passive castor wheels provided additional support for balance during the lateral motion tests.

B. EXPERIMENTAL CONDITIONS

Before the ground trajectory experiments, the AOW was tested in a suspended condition. This test was used to check the motor-side response of the wheel mechanism before adding ground-contact effects. In this condition, the wheel does not produce platform displacement, but the rolling and lateral actuation paths can still be commanded through the velocity controller and measured through the motor encoders and q-axis currents. The validation command set included pure rolling, pure lateral, and diagonal velocity commands. The diagonal command in the positive rolling and negative lateral direction was additionally tested at three command magnitudes. Each command case used the same trapezoidal velocity profile, consisting of ramp-up, constant-velocity hold, and ramp-down phases. The protocol is summarized in Fig. 9.

For the ground-contact test, experiments were performed on a $2\text{ m} \times 2\text{ m}$ vinyl surface to evaluate the trajectory accuracy and heading stability. Motion was captured by four overhead cameras (30 fps, 1920×1080) tracking five AprilTags on the robot. The synchronized camera views were fused to compute the planar position and yaw. Three reference trajectories were tested: (1) *Square*—independent v_x/v_y segments, (2) *Triangle*—mixed linear segments, and (3) *Circle*—continuous curved motion. Each trajectory used a trapezoidal velocity profile for consistent acceleration.

C. OFF-GROUND MOTOR-SIDE RESPONSE

For the simulation–experiment comparison, the measured motor velocity references from each off-ground segment were used as the simulation inputs. The comparison therefore evaluates the motor-side response of the model under the same command history applied to the prototype. The evaluated signals were the housing and sun motor velocities and the q-axis motor currents, which correspond directly to

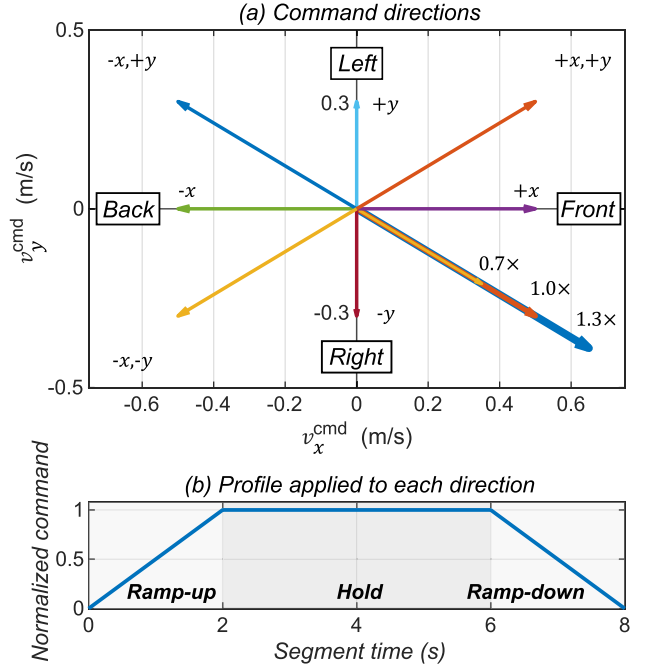


FIGURE 9. Off-ground motor-side validation protocol. The command set includes pure rolling, pure lateral, diagonal, and speed-scaled diagonal cases. Each case was applied using the same trapezoidal velocity profile.

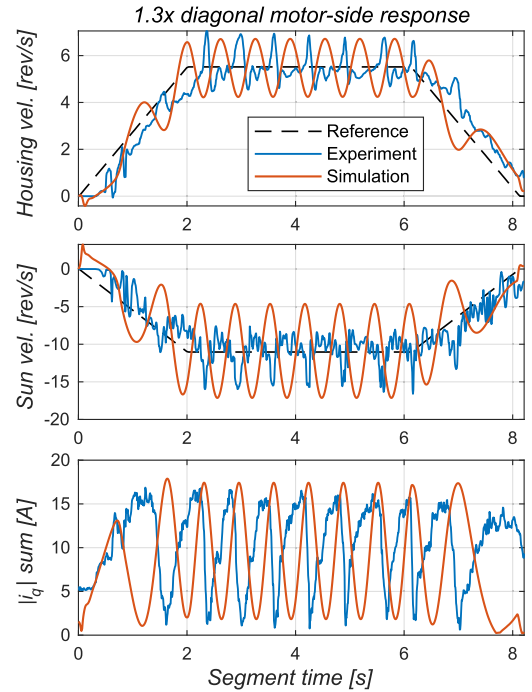


FIGURE 10. Representative off-ground motor-side comparison for the $1.3\times$ diagonal command. The housing and sun motor velocities are compared with the recorded motor velocity references, and the current demand is shown as $i_{q,\text{sum}} = |i_{q,H}| + |i_{q,S}|$. One representative experimental trial is compared with the calibrated velocity-response simulation without averaging repeated time histories.

the velocity-control-based model in Section II-C. The total current demand was represented as

$$i_{q,\text{sum}} = |i_{q,H}| + |i_{q,S}|, \quad (28)$$

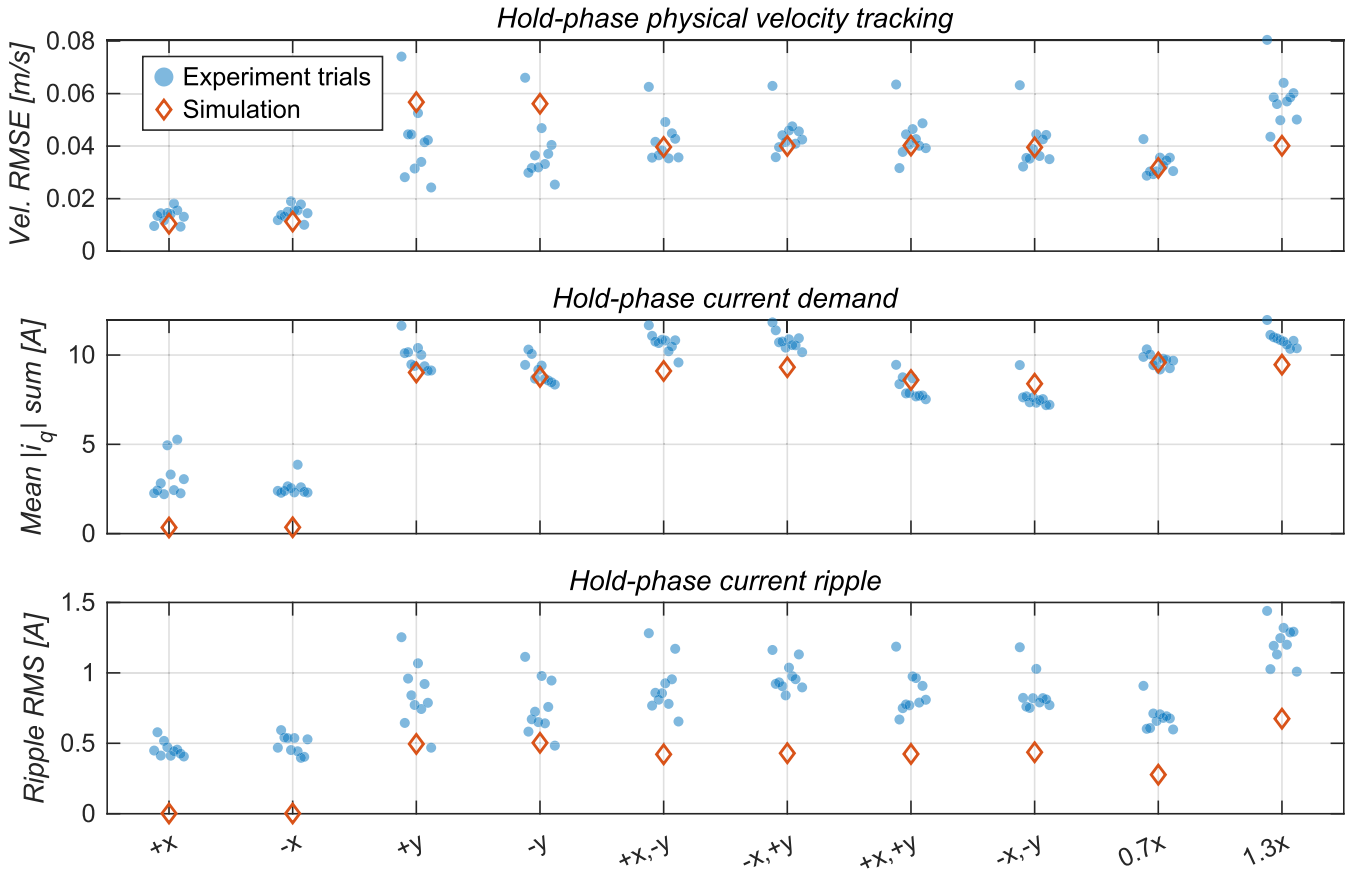


FIGURE 11. Phase-independent hold-phase summary of the off-ground repeated runs. Experimental markers indicate individual trials, and the simulation marker indicates the calibrated model response under the recorded command profiles. The quantities are computed only during the constant-velocity hold phase and include physical velocity tracking error, mean total q-axis current $i_{q,sum}$, and current ripple magnitude. The figure is used to show reproduced motor-load trends and model limitations rather than pointwise waveform agreement.

where the absolute value was used because the sign of the measured q-axis current depends on the motor-side coordinate convention, while the current magnitude indicates the required motor effort.

The periodic disturbance in (24)–(25) is associated with the relative lateral position of the segmented TPU tire and roller-pinion engagement. Since the initial tire posture was not fixed before each run, repeated experiments can have different periodic phases even under the same velocity command. For this reason, repeated time histories were not averaged. Instead, Fig. 10 shows one representative trial together with the calibrated simulation, while Fig. 11 summarizes all repeated trials using hold-phase scalar quantities that do not remove the phase-dependent periodic response.

Fig. 10 shows the direct waveform-level comparison for the $1.3\times$ diagonal command. The housing and sun velocity responses follow the same command direction and acceleration/hold/deceleration sequence as the recorded references. During the hold phase, both the experiment and simulation show repeated current fluctuation rather than a constant current level, which is consistent with the relative-position-periodic lateral disturbance in (24)–(25). The simulated waveform does not exactly overlap the experimental waveform because the initial tire posture

determines the disturbance phase. Therefore, the figure is used to confirm that the model reproduces the observed coupled response pattern, not to claim unique identification of the tire-force parameters.

The repeated-run comparison in Fig. 11 separates the response into physical velocity tracking, current demand, and current ripple. The velocity RMSE was computed after converting the housing and sun motor velocities into the corresponding physical velocity components using the kinematic relation in (7). In pure rolling, the experimental median velocity RMSE was approximately 0.014 m/s, while the simulation produced approximately 0.011 m/s. For commands containing lateral motion, the experimental median velocity RMSE was approximately 0.041 m/s, and the simulation produced approximately 0.040 m/s. This indicates that the calibrated model reproduces the hold-phase physical velocity tracking level of the prototype, although the waveform-level phase and ripple are not expected to coincide exactly between repeated runs.

In contrast, the current-demand trend was reproduced more consistently for lateral and diagonal commands. The experimental median $i_{q,sum}$ increased from approximately 2.43 A in pure rolling to 9.77 A in commands containing lateral motion, while the simulation produced a

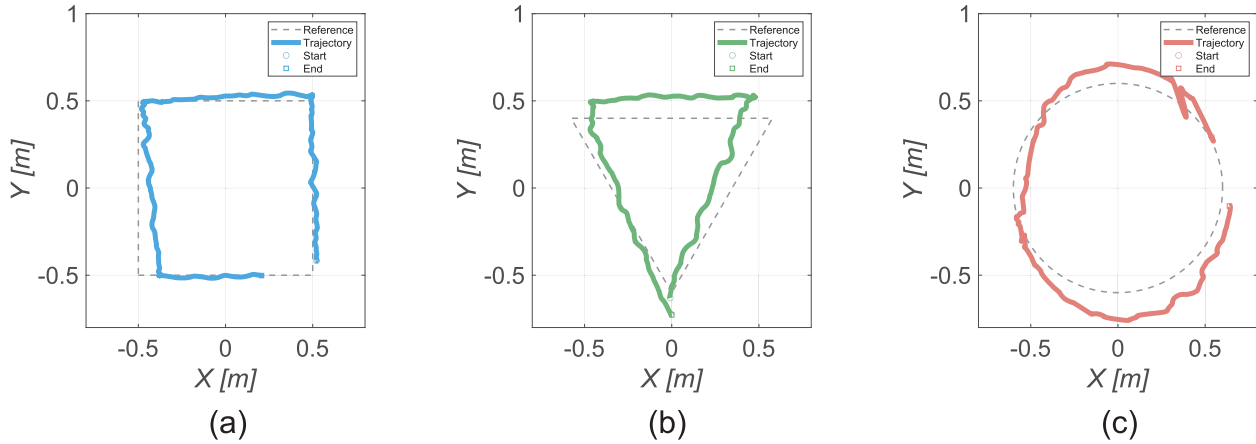


FIGURE 12. Measured trajectories of the AOW-POW robot: (a) square, (b) triangle, and (c) circle. Dashed lines represent commanded paths, and solid lines represent measured positions. Each trajectory was executed using identical trapezoidal velocity profiles.

TABLE 3. Yaw drift performance under repeated trajectory tests.

Traj.	RMS ($^{\circ}$)	Max $ \Delta $ ($^{\circ}$)	Pk-Pk ($^{\circ}$)	$\pm 1^{\circ}$ range (%)	Drift ($^{\circ}/s$)
Circle	0.726	1.192	1.793	80.0	+0.0144
Triangle	0.702	1.138	1.344	85.8	-0.0401
Square	0.656	1.177	2.017	90.9	-0.0607

corresponding increase from approximately 0.35 A to 9.06 A. The pure rolling current is underestimated because baseline motor and drivetrain losses are not fully represented, but the additional current demand caused by relative/lateral actuation is reproduced with comparable magnitude. The current-ripple result also remains nearly zero in the simulated pure rolling cases and appears in the lateral/diagonal cases, although the simulated median ripple amplitude (0.43 A) is smaller than the experimental median (0.84 A). Therefore, Fig. 11 supports the proposed motor-side load structure in (18)–(27), while showing that the fitted disturbance coefficients should be interpreted as effective prototype-level parameters rather than independently measured tire constants.

D. GROUND-CONTACT TRAJECTORY RESULTS

The robot executed all commanded motion types with smooth velocity transitions. The representative trajectories and yaw profiles are shown in Figs. 12 and 13, respectively. In all trials, the heading remained within $\pm 2.5^{\circ}$. No obvious macroscopic slip was observed in the recorded trajectories, suggesting that the roller-pinion interface was sufficient for the tested ground-contact motions.

To quantify the XY trajectory accuracy, the geometric path deviation was computed as the shortest Euclidean distance from each measured position to the corresponding reference path. The resulting RMS and maximum path deviations are summarized in Table 4. These results indicate that the prototype maintained the commanded path geometry during representative straight-segment and curved omnidirectional motions.

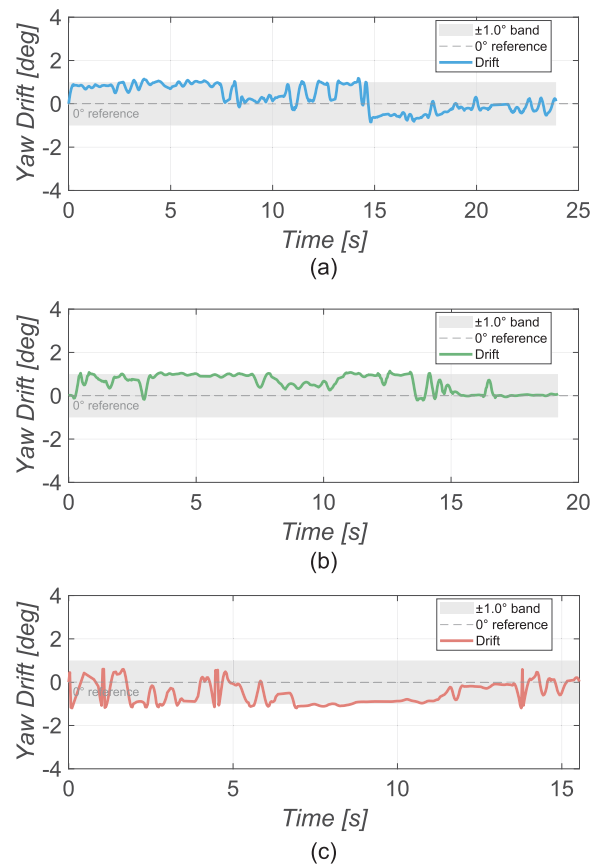


FIGURE 13. Yaw angle responses corresponding to the trajectories in Fig. 12. The robot maintained heading within $\pm 2.5^{\circ}$ in all cases, with minimal accumulated drift over time.

The quantitative yaw drift statistics are listed in Table 3. The square trajectory exhibited the smallest RMS drift (0.656°) and the highest time ratio within $\pm 1^{\circ}$ (90.9%). The circular trajectory accumulated slightly greater drift ($0.0144^{\circ}/s$) owing to its longer duration, while the triangular trajectory exhibited intermediate performance. Overall, the

TABLE 4. XY path deviation under ground-contact trajectory tests.

Trajectory	RMS deviation (m)	Max deviation (m)
Square	0.037	0.117
Triangle	0.090	0.138
Circle	0.089	0.161

trajectories maintained less than 2° heading deviation, demonstrating reliable heading retention even under coupled drive operation.

V. DISCUSSION AND CONCLUSION

Experiments have shown that the holonomic wheel-based system provides smooth omnidirectional motion using a single-sided dual-input differential transmission. With a 45° helical gear set, the wheel routed torque within a compact hub, and the roller-pinion interaction provided positive mechanical engagement with the compliant toroidal tire. This demonstrates that two orthogonal motion channels can be coupled within a single-wheel architecture while preserving heading stability over the tested trajectories.

The yaw-drift analysis in Table 3 indicates that the mechanical coupling between the sun and housing drives did not prevent stable heading retention under the tested trajectory commands. The drift rates remained below $0.02^\circ/\text{s}$ under continuous curved motion, suggesting that the compliant interface did not introduce an unbounded orientation error. The small drift rates observed in the polygonal trajectories may be influenced by the compliant tire response, but isolating this effect requires dedicated tire-contact experiments.

In parallel with the drivetrain, this study also verified the feasibility of a holonomic nonpneumatic tire tailored to the proposed wheel. The segmented toroidal TPU lattice was obtained by computational design and multi-objective optimization of Bezier curve-based cellular profiles, balancing bendability for assembly, payload-bearing stiffness, and traction while preserving a coreless, deformable structure. The resulting Cell A/Cell B/Cell C architecture coupled to the wheel through drive rollers, support rollers, and a roller-pinion interface enables longitudinal and lateral actuation without introducing a rigid core. The simplified roller-pinion feasibility result in Fig. 7 showed that the realized lateral travel remained close to the nominal geometric value under representative loads, indicating that the compliant lattice can transmit the commanded motion without excessive geometric deformation error at the drive interface. A separate tire-focused study will detail the FEA and material characterization of the toroidal lattice to refine its viscoelastic response and contact mechanics for future prototypes.

The present study is positioned as a feasibility demonstration rather than a performance optimization. No experimental tests were conducted to quantify the torque efficiency, traction coefficient, or energy loss. These metrics will be evaluated on a future dual-AOW platform equipped with improved tire prototypes. Nonetheless, the results reported

here confirm that the mechanism behaves in accordance with the modeled kinematics and observed coupling characteristics, while the off-ground comparison supports the proposed motor-side response model before ground-contact effects are added. The compliant tire and roller-pinion interface are compatible with the single-sided differential wheel concept.

In summary, the proposed holonomic wheel offers a compact and mechanically constrained approach to achieving omnidirectional motion without relying on friction-only lateral drive. By demonstrating a compact single-sided AOW with a compliant tire and measurable coupled motor-side response, this study provides a basis for further development of holonomic wheel modules for mobile robots that require lateral maneuverability in confined indoor environments, such as service, inspection, and logistics applications. The demonstrated functionality and observed coupling behavior establish a foundation for a forthcoming dual-AOW platform, and motivate the development of dynamic models and control strategies based on virtual shaft or mean difference formulations to resolve torque redundancy and manage coupling effects in compliant, gear-driven omnidirectional actuation.

ACKNOWLEDGMENT

(Jeeho Won and Chanju Lee contributed equally to this work.)

REFERENCES

- [1] J. Blumrich, "Omnidirectional wheel," U.S. Patent 3 789 947, Feb. 5, 1974.
- [2] D. Brescianini, "Increasing the maneuverability of multirotor vehicles: Attitude control, an omni-directional multirotor vehicle, and trajectory generation," Ph.D. dissertation, Inst. Dyn. Syst. Control, ETH Zurich, Zurich, Switzerland, 2018.
- [3] C. Sprunk, B. Lau, P. Pfaff, and W. Burgard, "An accurate and efficient navigation system for omnidirectional robots in industrial environments," *Auto. Robots*, vol. 41, no. 2, pp. 473–493, Feb. 2017.
- [4] H. Taheri and C. X. Zhao, "Omnidirectional mobile robots, mechanisms and navigation approaches," *Mechanism Mach. Theory*, vol. 153, Nov. 2020, Art. no. 103958.
- [5] I. Doroftei, V. Grosu, and V. Spinu, *Omnidirectional Mobile Robot-Design and Implementation*. London, U.K.: INTECH Open Access Publisher, 2007.
- [6] V. Kalman, "Controlled braking for omnidirectional wheels," *Int. J. Control Sci. Eng.*, vol. 3, no. 2, pp. 48–57, 2013.
- [7] W. Lee, J. Won, G. Park, and T. Seo, "Mechanical survey on wheeled mobile robot platform for industrial and personal service robots," *Int. J. Precis. Eng. Manuf.*, vol. 25, no. 8, pp. 1–15, Aug. 2024.
- [8] H. Akkad and H. Almusawi, *A Comparative Review of Omnidirectional Wheel Types for Mobile Robotics*. Rochester, NY, USA: Social Sci. Res. Netw. (SSRN), 2023.
- [9] M. J. A. Safar, "Holonomic and omnidirectional locomotion systems for wheeled mobile robots: A review," *Jurnal Teknologi*, vol. 77, no. 28, pp. 91–97, Dec. 2015.
- [10] L. Tagliavini, G. Colucci, A. Botta, P. Cavallone, L. Baglieri, and G. Quaglia, "Wheeled mobile robots: State of the art overview and kinematic comparison among three omnidirectional locomotion strategies," *J. Intell. Robot. Syst.*, vol. 106, no. 3, p. 57, Nov. 2022.
- [11] L. Komori, K. Matsuda, T. Terakawa, F. Takeoka, H. Nishihara, and H. Ohashi, "Active omni wheel capable of active motion in arbitrary direction and omnidirectional vehicle," *J. Adv. Mech. Design, Syst., Manuf.*, vol. 10, no. 6, 2016, Art. no. JAMDSM0086.
- [12] T. Terakawa, M. Komori, Y. Yamaguchi, and Y. Nishida, "Active omni wheel possessing seamless periphery and omnidirectional vehicle using it," *Precis. Eng.*, vol. 56, pp. 466–475, Mar. 2019.

- [13] J. Shen and D. Hong, "OmBUro: A novel unicycle robot with active omnidirectional wheel," in *Proc. IEEE Int. Conf. Robot. Autom. (ICRA)*, May 2020, pp. 8237–8243.
- [14] H. Gomi, S. Kobashi, T. Takenaka, K. Hamaya, W. Yada, K. Akimoto, Y. Uebayashi, and H. Waita, "Frictional drive device and inverted pendulum type vehicle using the same," U.S. Patent 8 353 378, Jan. 15, 2013.
- [15] S. Long, T. Terakawa, M. Yogou, R. Koyano, and M. Komori, "Kinetic analysis of active omni wheel with barrel-shaped rollers for avoiding slippage and vibration," *J. Mech. Robot.*, vol. 16, no. 5, May 2024, Art. no. 051002.
- [16] T. Terakawa, Y. Kinoshita, T. Watanabe, M. Sakamoto, and M. Komori, "Omnidirectional mobile vehicle using belt-driven seamless active omni wheel," *J. Robot. Mechatronics*, vol. 37, no. 6, pp. 1420–1432, Dec. 2025.
- [17] M. Burkacki, I. Lysy, S. Suchoń, M. Chrzan, and R. Kowolik, "Systematic review of mecanum and omni wheel technologies for motor impairments," *Appl. Sci.*, vol. 15, no. 9, p. 4773, Apr. 2025.
- [18] Honda Motor Co., Ltd. (2019). *Honda Omni Traction Drive System*. Honda Global Website. [Online]. Available: https://global.honda/en/innovation/CES/2019/omni_traction_drive_system.html
- [19] W. Liddiard, "Omnidirectional wheel," U.S. Patent 9 770 943, Sep. 26, 2017.
- [20] S. Hwang, "Vehicle having OMNI-wheel," Korean Patent 1 020 240 145 620, Dec. 23, 2024.
- [21] S. Hwang, "Omnidirectional driving tire and omnidirectional driving device including the same," Korean Patent 1 020 250 147 996, Dec. 13, 2025.
- [22] J. Wang, H. Zeng, Q. Gao, W.-H. Liao, G. Yin, J. Li, C. Wang, and R. Qiu, "Performance analysis and multi-objective optimization of a non-pneumatic tire with bionic petal spokes," *Struct. Multidisciplinary Optim.*, vol. 66, no. 9, p. 194, Sep. 2023.

JEEHO WON received the B.S. degree in mechanical engineering from Hanyang University, Republic of Korea, in 2020, where he is currently pursuing the Ph.D. degree. He participated as a Visiting Scholar with the Modular Robotics Laboratory, University of Pennsylvania. His research interests include design and control of mobile robotics and robotic gripper design.

CHANJU LEE received the B.S. degree in mechanical engineering from Kyunghee University, Seoul, South Korea, in 2022. He is currently pursuing the integrated M.S. and Ph.D. degrees in mechanical engineering with Korea University, Seoul. His research interests include computational mechanics and structural design optimization.

INHA PARK received the B.S. degree in mechanical system engineering from Kyonggi University, in 2022. She is currently pursuing the Ph.D. degree in mechanical engineering with Hanyang University. Her research interests include robot design and control using artificial intelligence.

MYEONGJIN CHOI received the B.S. degree in electronic engineering and the Ph.D. degree in mechanical engineering from Hanyang University, in 2020 and 2025, respectively. He is currently a Research Fellow with Korea Institute of Machinery & Materials.

SANGHYUN CHO received the B.S. degree in mechanical engineering from Kyunghee University, Seoul, South Korea, in 2023. He is currently pursuing the integrated M.S. and Ph.D. degrees in mechanical engineering with Korea University, Seoul. His research interests include computational mechanics and structural design optimization.

CHEOLGI LYOO received the B.S. degree in mechanical engineering from Sejong University, Seoul, South Korea, in 2024. He is currently pursuing the integrated M.S. and Ph.D. degrees in mechanical engineering with Korea University, Seoul. His research interests include computational mechanics and structural design optimization.

DONGWOO SUH received the B.S. degree in mechanical engineering from Kyunghee University, Seoul, South Korea, in 2024. He is currently pursuing the integrated M.S. and Ph.D. degrees in mechanical engineering with Korea University, Seoul. His research interests include surrogate model and structural design optimization.

SANGWOO HWANG received the B.S. and M.S. degrees in mechanical engineering from Korea University, Seoul, South Korea, in 2013 and 2015, respectively. He has been with Hyundai Motor Company, since 2015, where he has worked at the Suspension & Steering Research Laboratory, Institute of Advanced Technology Development (IATD). He is currently with the Advanced Chassis Development Team. His research interests include mechatronic system design, system functional safety, systems engineering, mobile robotics, and modular robotics.

GUNWOO NOH (Member, IEEE) received the B.S. degree in mechanical engineering from Korea Advanced Institute of Science and Technology, Daejeon, South Korea, in 2009, and the M.S. and Ph.D. degrees in mechanical engineering from Massachusetts Institute of Technology, Cambridge, MA, USA, in 2011 and 2013, respectively. He was a Research Scientist with Korea Institute of Science and Technology, Seoul, South Korea, until 2017, and was an Assistant Professor with the School of Mechanical Engineering, Kyungpook National University, Daegu, South Korea, until 2021. He is currently an Associate Professor with the School of Mechanical Engineering, Korea University, Seoul. His research interests include computational mechanics and structural design optimization.

TAEWON SEO (Senior Member, IEEE) received the B.S. and Ph.D. degrees from the School of Mechanical and Aerospace Engineering, Seoul National University, South Korea, in 2003 and 2008, respectively. He is currently a Professor with the School of Mechanical Engineering and the Director of the Robot Design Engineering Laboratory, Hanyang University, South Korea. Prior to his current position, he worked as a Postdoctoral Researcher with the Nanorobotics Laboratory, Carnegie Mellon University; a Visiting Professor at the Biomimetic Millisystems Laboratory, UC Berkeley; a Scholar at the University of Michigan; and an Associate Professor at the School of Mechanical Engineering, Yeungnam University, South Korea. His research interests include robot design, analysis, control, optimization, and planning. He received the Best Paper Award from IEEE/ASME TRANSACTIONS ON MECHATRONICS, in 2014. He served as a Technical/Associate Editor for IEEE/ASME TRANSACTIONS ON MECHATRONICS and IEEE ROBOTICS AND AUTOMATION LETTERS. Currently, he holds the position of an Associate Editor of *Intelligent Service Robots* and the *International Journal of Precision Engineering and Manufacturing*.

...