Theory and Practice of Science and Technology
2025, VOL. 6, NO. 11, 139-143
DOI: 10.47297/taposatWSP2633-456927.20250611

Design and Simulation of a Cycloidal Pinwheel Harmonic Reducer

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ABSTRACT
To overcome the design limitations of traditional harmonic reducers, which are predominantly based on involute tooth profiles, this paper investigates the full-process design and simulation of a novel harmonic reducer utilizing a cycloidal pinwheel. Taking rated output torque and transmission ratio as core metrics, the design parameters for the reducer's flexible gear, rigid gear, and wave generator are determined. Three-dimensional modeling of the flexible gear, rigid gear, and wave generator was completed using SolidWorks software. The cycloidal tooth profile was generated via "Equation-Driven Curve," followed by assembly constraints to complete the overall assembly. A simulation model was constructed using RecurDyn, employing the Step-IF function to achieve smooth startup and delayed loading. Simulation results indicate: steady-state output speed values closely match theoretical predictions, transmission ratio deviation is negligible, and maximum contact forces meet strength requirements, validating the model's feasibility. This design fills the gap in harmonic drives utilizing cycloidal tooth profiles, providing a theoretical framework and technical pathway for similar reducer designs.

KEYWORDS
Harmonic reducer; 3D modeling; RecurDyn; Simulation

1 Introduction

Harmonic reducers are precision transmission devices that transmit motion and power through the elastic deformation of flexible components. They feature compact size, high gear ratios, light weight, and high transmission accuracy. Primarily composed of four basic components—flexible gears, rigid gears, flexible bearings, and wave generators[1].They find extensive application in fields such as lightweight industrial robots, robotic arms, aerospace, and medical equipment.

Harmonic drive technology was developed by American scholar C.W. Mus-ser in the 1950s [2]. Research on harmonic reducers in China began in the 1980s when scholars Shen Yunwen and Ye Qingtai introduced the transmission principle domestically [3]. Wang Dahuo et al. proposed methods for designing involute tooth profiles, tooth direction refinement, and tooth profile metrological inspection for harmonic reducers, while also developing computational and experimental methods for load and acceleration modeling of the reducers [4]. Wang Zhen et al. proposed a novel harmonic reducer featuring a split-type flexible wheel structure with an involute tooth profile. This design represents an improvement over the monolithic cup-type flexible wheel structure, enhancing the flexible wheel's strength and extending the reducer's service life [5]. Dong Huimin et al. developed a design method for paired harmonic reducers with double-arc tooth profiles, grounded in the theory of instantaneous tooth centerlines [6]. Huang Bing et al. selected a double-arc tooth profile top hat-shaped flexible wheel as their research subject. Considering its tooth structure characteristics, they employed a two-stage refinement method for helical refinement design, demonstrating that the refined flexible wheel yields harmonic reducers with superior transmission performance [7]. Zhang Meng et al. systematically analyzed the motion error, stiffness, and friction direction dynamics of harmonic reducers from experimental, mechanistic, and modeling perspectives [8]. Shao Weilong developed a micro-miniature harmonic reducer by completing its structural design, optimizing the harmonic drive tooth profile, and refining gear parameters based on harmonic transmission theory [9]. Xu Hanxin et al. employed Visual Basic and SolidWorks for the parametric design and virtual assembly of a harmonic reducer. They conducted structural analysis on the critical flexible wheel component and established a model featuring a double-arc tooth profile with a common tangent line [10]. Through these studies, we observe that the development and optimization analysis of harmonic reducers are both based on involute tooth profiles. Therefore, this research proposes a novel harmonic reducer based on cycloidal pinwheels, using the LCD-32-30 model as a reference. The cycloidal pinwheel harmonic reducer introduced herein provides a new design approach for future reducers.

2 Structural Design of Harmonic Drive Gearboxes

2.1 Parameters of the Cycloidal Flexible Gear

The flexible gear structure, wave generator, rigid gear structure, and output mechanism are designed according to relevant design guidelines for harmonic reducers. This paper employs a cycloidal wheel as the flexible gear and a pinion wheel as the rigid gear. Based on the selected harmonic reducer, the cycloidal tooth profile was designed with a rated output torque of 60,000 N•mm and a gear ratio of 30. The number of teeth for the cycloidal wheel was determined to be 60, and for the pinion wheel, 62. Based on the calculation guidelines in the Mechanical Design Handbook [11], the

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fundamental parameters required for designing the cycloidal wheel and pinion were computed, with detailed information presented in Table 1. Parametric design of the cycloidal wheel (flexible gear) for the cycloidal profile harmonic reducer under study was performed according to these calculated fundamental parameters.

Table 1 Parameter

Name	Unit	Result
Short-range coefficient k_1	mm	0.689
Needle diameter coefficient k_2	mm	1.52
Tooth width b_c	mm	7
Eccentricity e		0.5
Radian of the needle tooth sleeve r_z	mm	1.2
Isometric Exercise Volume	mm	0.3815
Shift coefficient	mm	0.2765

Calculate the radius of the center circle of the needle teeth using empirical formula (1):

$$R_z = (0.85 \sim 1.3)\sqrt[3]{T} \tag{1}$$

The calculated radius of the pitch circle is $R_z = 45mm$;

Calculate the root circle diameter dfc of the cycloidal gear using formula (2).:

$$d_{fc} = d_p - 2e - d_{rp} - 2 \times (\Delta_{rp} + \Delta_p) \tag{2}$$

The calculated dfc is 85.7 mm;

The diameter dac of the tooth tip circle is calculated using formula (3):

$$d_{ac} = d_p + 2e - d_{rp} - 2 \times (\Delta_{rp} + \Delta_p) \tag{3}$$

Calculate dac is 87.7 mm;

2.2 Waveform Generator Parameters

The wave generator, serving as the input end, continuously induces elastic deformation waves in the flexible wheel through its own rotation. By utilizing the tooth misalignment caused by the difference in tooth count between the two wheels, it converts high-speed input into low-speed, high-torque output. The harmonic reducer studied in this research employs a cam wave generator. By eliminating flexible bearings, this cam wave generator achieves zero backlash, high rigidity, and extended service life. It offers advantages such as high transmission efficiency and high load capacity, while its simple structure facilitates machining and simulation analysis. Its structural parameters are:

Elliptical cam long half shaft:

$$a = \frac{d_n}{2} + \omega_0 \tag{4}$$

Elliptical cam short half shaft:

$$b = \frac{d_n}{2} - \omega_0 \tag{5}$$

$$\omega_0 = (0.8\sim1.2)\ \mathrm{m}\frac{d_z}{N} \tag{6}$$

$$h_0 = (0.01\sim0.03)\ d_{fc} \tag{7}$$

$$d_n = d_{fc} - h_0 \tag{8}$$

In the formula, is the maximum deformation of the wave generator, calculated as 0.57 from equation (6); is the wall thickness of the flexible wheel beneath the tooth root, calculated as 1.7 mm from equation (7); is the inner diameter of the flexible wheel, calculated as 84 mm from equation (8). According to equation (4), the major axis of the elliptical cam is determined to be 42.57. and the short semi-axis of the elliptical cam is 41.43, as determined by equation (5).

3 3D Modeling with SolidWorks Software

3.1 Three-dimensional Modeling of Flexible Gear

When creating a 3D model of a flexible gear (epicycloidal gear), the epicycloidal transmission structure is employed, making this flexible gear an epicycloidal gear. The specific modeling process is as follows: In the SolidWorks environment, create a new part, select the front view reference plane, and use an equation-driven curve as per formula (9) to sketch the epicycloidal tooth profile and apply constraints; extrude a boss to form the tooth section, then extrude and trim to remove the tooth root; Determine the boss dimensions based on design parameters and extrude the boss to model it. Based on shaft parameters, extrude the boss to create the shaft section and extrude and cut to form the shaft bore. Select the key according to the shaft system requirements, sketch the keyway for positioning, and extrude and cut. The final completed 3D model of the flexible gear is shown in Figure 1.

Cycloid equation:

$$\begin{cases} x(t)=R_z*cos(t)-k1*e*cos(z_b*t) \\ y(t)=R_z*sin(t)-k1*e*sin(z_b*t) \end{cases} \tag{9}$$

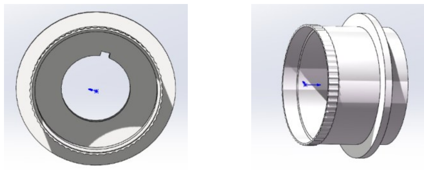

Figure 1 Flexible Gear Model

3.2 Three-dimensional Modeling of Rigid Gear

When creating a 3D model of the rigid gear (pinion), its structure adopts a pinion form to align with transmission characteristics. The specific modeling process is as follows: Create a new part file in SolidWorks, select the front view reference plane as the sketch reference plane, sketch the pinion chassis outline, and complete geometric constraints. Execute the “Extrude Boss/Base” feature to generate the chassis base. Create a new sketch on the base end face. Sketch the cross-sectional profile of a single pin tooth and apply constraints for positioning. Use the “Circular Array” feature to arrange all pin teeth according to the specified circular angle. Then apply the “Extrude Boss/Base” feature to the pin tooth sketch, generating a uniformly distributed pin tooth structure. Based on the mating dimensions of the flexible sleeve, sketch the outer contour of the steel wheel's outer wall on the chassis periphery. Execute the “Extrude Boss/Base” feature to form the outer wall. The final 3D model of the steel wheel (pin wheel) is shown in Figure 2.

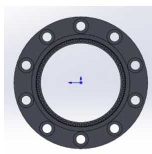

Figure 2 Rigid Gear Model

3.3 Three-Dimensional Modeling of a Wave Generator

The specific workflow for creating a 3D model of the wave generator is as follows: In the SolidWorks environment, create a new part file. Select the front view reference plane as the sketch reference plane. Sketch the cross-sectional profile of the wave generator, complete geometric constraints and dimensional annotations, and execute the “Extrude Boss/Base” feature to generate the main structure of the wave generator. The final completed 3D model of the wave generator is shown in Figure 3.

Figure 3 Wave Generator Model

After completing the 3D modeling of all major components of the cycloidal harmonic reducer, the overall assembly process can commence. In the SolidWorks environment, create a new assembly file. First, import the steel gear component into the assembly environment and designate it as the fixed reference part. Subsequently, import the flexible gear component, establishing an initial meshing relationship with the steel gear. Finally, import the wave generator component to complete its assembly positioning relative to the flexible gear. To ensure assembly accuracy, constraints such as co-location, concentricity, and distance were applied to restrict relative degrees of freedom between components. This guarantees correct meshing of the gear pair and positional accuracy of the overall structure. The final assembled model of the cycloidal harmonic reducer is shown in Figure 4.

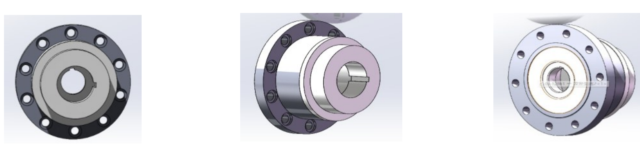

Figure 4 General Assembly Drawing

4 Kinematic Simulation Analysis Based on RecurDyn

4.1 Model Development in RecurDyn

Due to assembly interference between the cam-type wave generator and the flexible pulley, direct assembly is not feasible. Therefore, a split-type structure is adopted for the cam wave generator design. By simulating the radial expansion motion of the split cam through simulation, the actual tensioning process of the wave generator on the flexible pulley under operational conditions is reproduced. Upon completion of the simulation, a model extraction operation was performed within the RecurDyn software environment. This generated a feature model of the wave generator in its clamped state relative to the flexible wheel, providing initial geometric constraints and boundary conditions that accurately reflect the actual assembly state for subsequent dynamic simulation analysis.

Name the extracted flexspline-cam wave generator clamped state model as "Flexspline-CamWaveGen_ClampedState_V1.0". Using the MMKS SI units (millimeters, kilograms, newtons, seconds), set the gravitational acceleration to 9806.65 mm/s^2 along the -Y axis in the RecurDyn global coordinate system. After performing geometric topology verification on the model, export it in STEP AP214/IGES V5.3 format. Import it via RecurDyn's "File-Import-Geometry" interface, checking "Automatic Geometry Repair" and " Preserve Assembly Hierarchy"; post-import, verify geometric continuity using the " Model Verify" tool. In the software's "Material Library", assign S45C steel parameters to the cam wave generator (density 7.85e-6 kg/mm^3, elastic modulus 206000 N/mm^2, Poisson's ratio 0.3). Assign 60Si2Mn steel parameters to the flexible pulley (density 7.85e-6 kg/mm^3, elastic modulus 200000 N/mm^2, Poisson's ratio 0.3). Then adjust the mass via the "Mass Properties" dialog box: - For the cam wave generator, directly input the target mass and distribute it uniformly. - For the flexible wheel, automatically calculate the mass based on material density, then fine-tune using the "Mass Adjust" tool to ensure the mass-inertia parameters deviate ⩽3% from the physical prototype.

Constraint conditions were set for the imported model based on the operational characteristics of the reducer. The added constraints are listed in Table 2.

Table 2 Schedule for Setting Restrictions

Constraint Type	Component 1(Base)	Component 2(Action)
Fixed	ground	rigid gear
	output shaft	flexible gear
	input shaft	cam1
	input shaft	cam2
Revolute	ground	input shaft
	ground	output shaft
Contact	cam1	flexible gear
	cam2	flexible gear
	rigid gear	flexible gear

4.2 Driver Settings

The input shaft drive speed employs RecurDyn's built-in Step function to construct a smooth acceleration curve, defined by the expression step(time, 0, 0, 0.04, 220*2*pi/60), where 220*2*pi/60 converts the input speed of 220 r/min to angular velocity (unit: rad/s), achieving a smooth transition from rest to the target angular velocity within 0–0.04 s. The load torque employs a combination of the Step function and RecurDyn's conditional judgment function (IF function) to construct a delayed-start progressive loading curve. The expression is set as: (step(time, 0.02, 0, 0.04, 72000)) * IF(time-0.02:0,0,1) (load torque unit: N·mm). The IF function implements delayed loading logic: no load before 0.02 s, and gradual loading from 0 to 72000 N·mm between 0.02 and 0.04 s. Multibody dynamics simulation parameters are configured as follows: total simulation duration 0.2 s, number of simulation steps 2000. Execute the simulation after completing the above settings.

4.3 Simulation Results

Using RecurDyn software, a multi-body dynamics (MBD) simulation analysis was conducted on the parametric assembly model of a cycloidal harmonic reducer. Through simulation solutions, the time-domain curves for both input and output speeds were obtained. During simulation, it was observed that the system undergoes a transient phase of velocity smoothing transition within the initial 0.04 seconds. This phase exhibits dynamic response instability, which may affect data accuracy. Therefore, the data stream after 0.04 seconds, when the system enters a steady state, was selected for output speed calculation. Data processing yielded the following: The mean value of the steady-state output speed simulation analysis was 0.7685 rad/s, while the theoretical output speed calculated using the gear reducer's transmission ratio formula was 0.7679 rad/s. The deviation between the two was only 0.08%, indicating a high degree of agreement between the simulation results and the theoretical value. Further utilizing RecurDyn's PostProcessor module, data extraction and fitting calculations were performed on the time-domain curves of input and output speeds during the steady-state segment after 0.04 seconds. This yielded an actual transmission ratio of 30.2252 for the reducer under simulation conditions. The deviation from the designed theoretical transmission ratio of 30 was less than 0.75%, once again validating the accuracy and reliability of the simulation model. Relevant simulation curves and calculation results are shown in Figures 5 to 7.

Figure 8 shows the contact force curve between the flexible gear and rigid gear. Figure 9 displays the local contour plot of the maximum contact force between the flexible gear and rigid gear. From Figures 8 and 9, it can be determined that the maximum contact force is 362.1822 N.

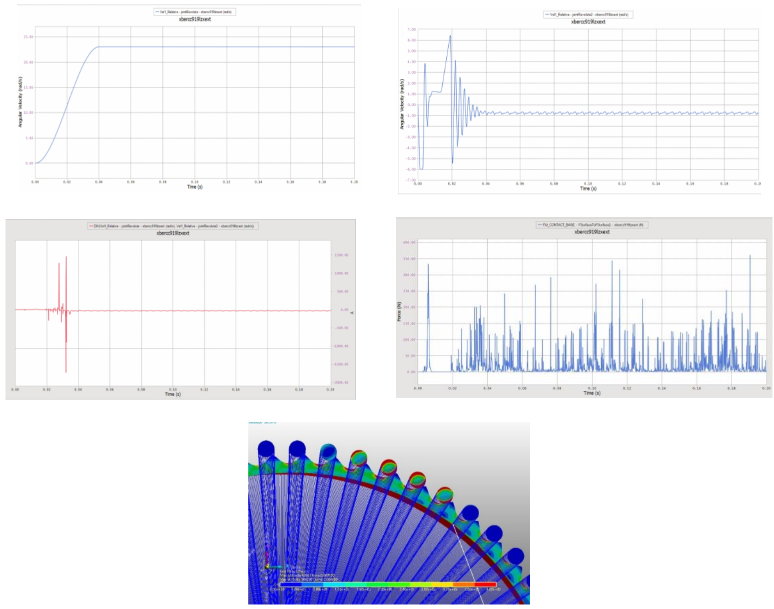

Figure 9 Cloud Chart of Maximum Contact Force

The above simulation results validate the correctness of the multibody dynamics simulation model for this cycloidal gear harmonic reducer.

5 Conclusion

This paper investigates the LCD-32-30 harmonic reducer model, breaking away from the traditional design paradigm based on involute tooth profiles. It completes the full-process design, modeling, and multi-body dynamics (MBD) simulation verification of a cycloidal pinwheel harmonic reducer. A model of the cycloidal-tooth harmonic reducer was created using SolidWorks. Dynamic simulation experiments were conducted on the model via RecurDyn. The drive and load settings employed a Step-IF composite function, achieving smooth startup at an input speed of 220 r/min and delayed loading of a 72,000 N·mm load. Simulation results indicate that the steady-state output speed simulation value of 0.7685 rad/s deviates only 0.08% from the theoretical value of 0.7679 rad/s. The actual transmission ratio of 30.2252 deviated less than 0.75% from the design value of 30. The maximum contact force between the flexible and rigid gears reached 362.1822 N, meeting material strength verification requirements. This solution fills the application gap of cycloidal tooth profiles in harmonic drives. Compared to traditional involute gear reducers, its cycloidal meshing pair exhibits larger contact area and more uniform tooth surface stress distribution. The cycloidal parameter calculation method, 3D modeling workflow, and multibody dynamics simulation framework established during this research provide reusable theoretical models and technical pathways for optimizing similar reducers and developing prototypes.

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