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# STRUCTURAL DESIGN AND ANTI-ROLLOVER STABILITY ANALYSIS OF AN AGV

Peng Zhang*, Tao Yang
*School of Mechanical Engineering, Chengdu University, Chengdu 610106, Sichuan, China.*
**Corresponding Author: Peng Zhang*

**Abstract:** To address the high frequency and heavy workload of cargo handling in small and medium-sized logistics warehouses, this paper designs an Automated Guided Vehicle (AGV) forklift to improve transport efficiency and reduce labor costs. Design calculations are performed for the driving and steering system, shock absorption mechanism, and lifting system of the vehicle. The Center of Gravity (CoG) method is applied to analyze and verify the dynamic and static stability of the AGV forklift under various working environments, ensuring that the designed vehicle meets stability requirements.
**Keywords:** Automated Guided Vehicle (AGV); Structural design; Center of Gravity (CoG); Dynamic and static stability

## 1 INTRODUCTION

In recent years, driven by rising labor costs and the continuous expansion of the logistics sector, conventional manual labor has become increasingly insufficient to meet the demands of high-intensity cargo handling operations [1-2]. To improve production efficiency and lower manufacturing expenses, automating the rapidly growing logistics industry has emerged as a crucial breakthrough [3].
Automated Guided Vehicles (AGVs) are smart logistics robots that integrate motion, navigation, and dynamic path planning [3-4]. Characterized by high flexibility and adaptability, AGVs operate without the need for fixed tracks or supporting structures, thereby bypassing the spatial and physical limitations typical of traditional systems [4-5]. Furthermore, as AGV designs evolve toward specialized material handling, optimizing structural stability and wheel configurations (such as steer-drive systems) has become imperative to prevent tipping or rollover during high-load and dynamic acceleration scenarios [6, 7]. Consequently, deploying intelligent AGVs in small and medium-sized logistics warehouses effectively mitigates operational and labor costs, simplifies managerial complexity, and substantially enhances material handling efficiency [2, 8].

## 2 STRUCTURAL DESIGN OF THE AGV VEHICLES

### 2.1 Design of the Drive and Steering Mechanism

Three-wheeled configurations offer simple control and operation but suffer from poor stability. Conversely, four-wheeled configurations provide superior stability, albeit at the expense of greater control complexity. Balancing these trade-offs, the AGV proposed in this paper adopts a five-wheel structure with a single steerable drive wheel (steer-drive wheel), as illustrated in Figure 1.

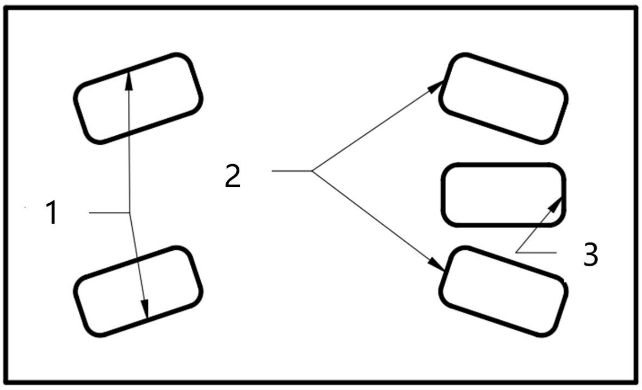


**Figure 1** A Five-Wheel Structure with a Single Steerable Drive Wheel
Note: 1. Front wheels, 2. Rear wheels, 3. Steerable drive wheel

### 2.2 Design of the Steerable Drive Wheel

As depicted in Figure 2, the steerable drive wheel consists of a drive motor, a steering motor, and external gear slewing bearing, among other components. The traction is supplied by a drive motor integrated with a reducer and an electromagnetic power-off brake. To deliver steering torque, a steering motor with an integrated encoder is selected.

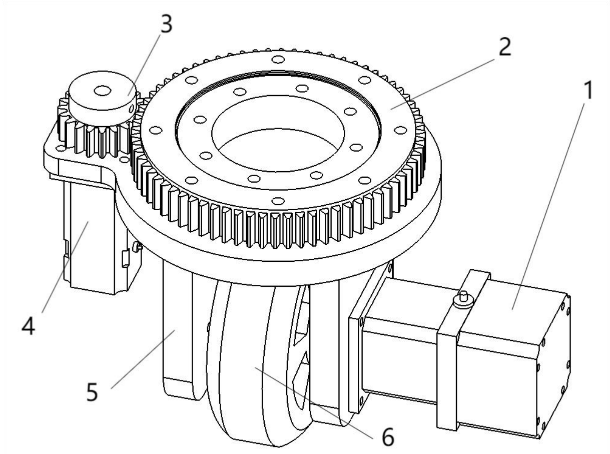


**Figure 2** Schematic Diagram of the Steerable Drive Wheel Structure
Note: 1. Drive motor, 2. External gear slewing bearing, 3. Steering gear, 4. Steering motor, 5. Wheel frame, 6. Rubber wheel

***2.2.1 Selection of drive motor***

The maximum output power of the drive motor occurs when the AGV travels uphill at full load. Under this condition, the vehicle is primarily subject to four resistance forces: rolling resistance $F_f$, grade resistance $F_g$, acceleration resistance $F_a$, and aerodynamic resistance $F_w$.

$$F = F_f + F_g + F_a + F_w \tag{1}$$

$$P = \frac{T \cdot n}{9550 \cdot \eta} \tag{2}$$

The calculated required power for the drive motor is P=0.756 kW. Based on the selection manual, a 48V, 800W DC servo motor is selected as the drive motor.

***2.2.2 Selection of steering motor***

The AGV designed in this study utilizes a five-wheel system with a single steerable drive wheel. The load borne by the steerable drive wheel corresponds to the ground reaction force obtained from the overall vehicle force analysis. The output power of the steering motor must exceed the power required to overcome friction during steering. The calculated steering power required for the steerable drive wheel is $P$=0.059 kW. According to the selection manual, a 24V, 100W DC servo motor is selected as the steering motor.

**2.3 Design of Shock Absorption Mechanism**

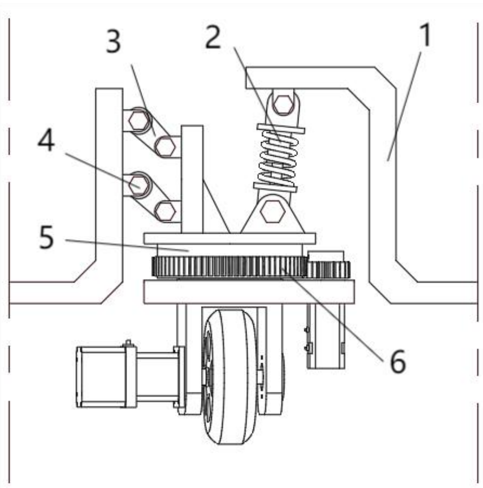


**Figure 3** Schematic Diagram of the Shock Absorption Mechanism
Note: 1. Vehicle chassis, 2. Spring-damping shock absorber, 3. Guidance connecting rod, 4. Bolt rotating pin, 5. Connecting seat, 6. External gear slewing bearing

The AGV shock absorption mechanism consists of a spring-damper absorber, a mounting base, guide connecting rods, and a pivot pin, as shown in Figure 3. Two parallel guide connecting rods ensure that the wheel assembly performs purely translational motion, keeping the wheel strictly perpendicular to the ground to maximize the contact area. Meanwhile, the spring-damper absorber provides vibration damping during the translational motion of the wheel. The mounting base is connected to the slewing bearing. This mechanism guarantees high stability and strong adaptability when the AGV operates on uneven or rough road conditions.

Through force analysis and calculations of the shock absorption mechanism under flat, depressed, and raised road conditions, the required spring stiffness of the spring-damper absorber is determined to be $42.8\ \mathrm{N/mm} < K < 86.9\ \mathrm{N/mm}$.

### 2.4 Design of the Lifting Mechanism

The lifting system is the primary operating mechanism of the AGV forklift, playing a critical role in carrying out cargo picking, elevating, and placing operations. As shown in Figure 4, it comprises a mast, forks, a fork carriage, and a lifting power system. The design of the lifting system encompasses both structural and powertrain aspects. It incorporates a single-stage mast, hook-type forks mounted on a plate-type fork carriage, and a motor-driven timing belt-pulley drive system (equipped with a 500W power motor and an 8M timing belt/pulley with a belt width of 30 mm).

The Three-dimensional Model of the AGV Vehicle is shown in Figure 5.

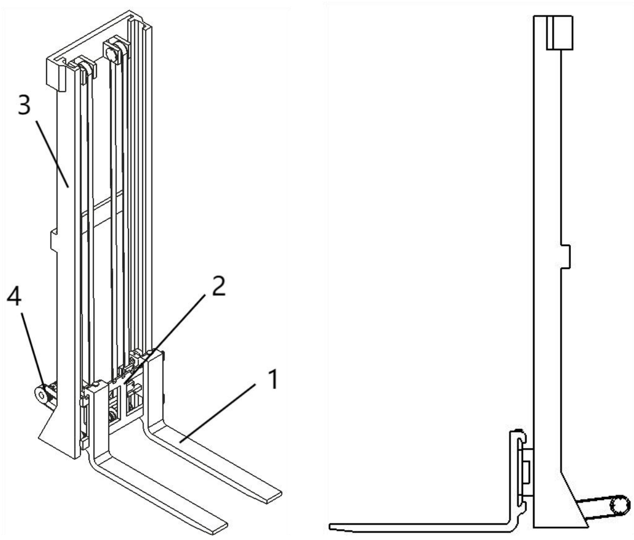


**Figure 4** The Structural Design Diagram of the Lifting System
Note: 1. Fork, 2. Fork carriage, 3. Mast, 4. Lifting power system

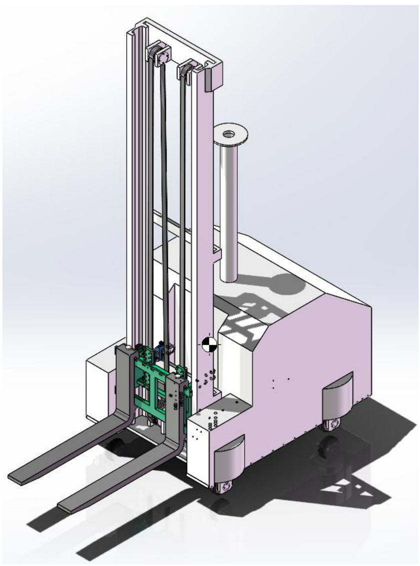

**Figure 5** The Three-Dimensional Model of the AGV Vehicle

## 3 STABILITY ANALYSIS OF THE AGV

During motion, the AGV experiences an increased tendency to pitch forward due to the combined effect of cargo weight and overall inertial forces. This behavior is particularly pronounced when steering on slopes or negotiating turns at full speed. Furthermore, centrifugal and inertial forces acting on the vehicle make it more susceptible to rollover. Therefore, calculating and verifying both the lateral and longitudinal stability of the AGV is imperative to guarantee safe operation across all working conditions.

Stability analysis of the AGV is conducted using the Center of Gravity (CoG) method across four specific scenarios: longitudinal static stability, longitudinal dynamic stability, lateral static stability, and lateral dynamic stability [9].

As shown in Figure 6, the steerable drive wheel A and the two front auxiliary support wheels B and C form the base isosceles triangle ABC of a triangular pyramid; this triangle is referred to as the stability triangle. The two equal sides, AB and AC, represent the primary lateral rollover axes, while BD and CE denote the secondary lateral rollover axes. The base BC of the isosceles triangle serves as the longitudinal rollover axis. Point G, representing the AGV's center of gravity, constitutes the apex of the triangular pyramid. Previous studies have demonstrated that when the AGV's center of gravity shifts laterally along either of the primary lateral rollover axes, the vehicle enters an unstable state. Consequently, to maintain AGV stability, the projection of its center of gravity must remain strictly within the stability triangle.

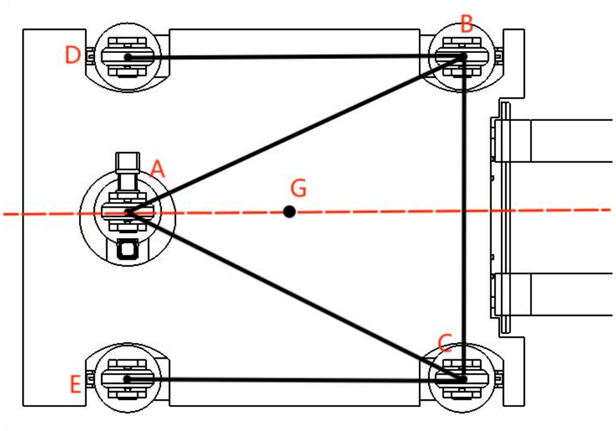


**Figure 6** The Stable Triangle of AGV

### 3.1 Longitudinal Static Stability Analysis

The longitudinal stability of the AGV is most critical when carrying the rated load at its maximum lifting height. Based on the CoG method, ensuring satisfactory longitudinal static stability requires that the projection of the combined center of gravity remains within the stability triangle ABC.

At this point, the combined center of gravity must satisfy:

$$\frac{a}{h} > 0.04 \tag{3}$$

The parameters $a$ and $h$ can be determined by the following formulas:

$$\begin{cases} h = \frac{G_0 y_0 + QH}{G_0 + Q} \\ a = \frac{G_0 x_0 - Qb}{G_0 + Q} \end{cases} \tag{4}$$

Where:

$h$ — Vertical distance from the combined CoG of the loaded AGV to the ground;

$a$ — Horizontal distance from the combined CoG of the AGV to the longitudinal rollover axis;

$G_0$ — Self-weight of the vehicle;

$Q$ — Weight of the cargo;

$y_0$ — Height of the unladen vehicle's CoG;

$H$ — Height of the cargo's CoG relative to the ground;

$x_0$ — Horizontal distance from the projection of the unladen vehicle's CoG to the longitudinal rollover axis BC;

$b$ — Horizontal distance from the projection of the cargo's CoG to the longitudinal rollover axis BC.

According to the principles of the CoG method, substituting the parameters $G_0 = 800$ kg, $Q = 100$ kg, $x_0 = 380$ mm, $y_0 = 500$ mm, $H = 2300$ mm, and $b = 550$ mm into Eq. (4) yields: $a = 276.6$ mm and $h = 700$ mm. The requirement for longitudinal static stability is satisfied.

### 3.2 Longitudinal Dynamic Stability Analysis

When the fully loaded vehicle is traveling with the cargo elevated to a medium height, sudden braking under inertial forces increases the tendency to pitch forward. To ensure vehicle stability, the torque created by the inertial force $P$ about the longitudinal rollover axis must balance the moment generated by the fully loaded vehicle's gravity about the same axis.

$$P \cdot h = (Q + G_0)a \tag{5}$$

When the longitudinal dynamic stability requirement of the vehicle is met:

$$\frac{a}{h} > 0.18 \tag{6}$$

Substituting the parameters into the equation yields: $a = 276.6\ \mathrm{mm}$ and $h = 522.2\ \mathrm{mm}$. The requirement for longitudinal dynamic stability is satisfied.

### 3.3 Lateral Static Stability Analysis

Vehicle rollover is caused by various factors, including centrifugal forces generated during turns, lateral wind loads, and gravitational components resulting from cross-slopes. Among these, the centrifugal force generated during steering exerts the greatest impact on stability. The lateral static stability analysis of the loaded vehicle is illustrated in Figure 7.

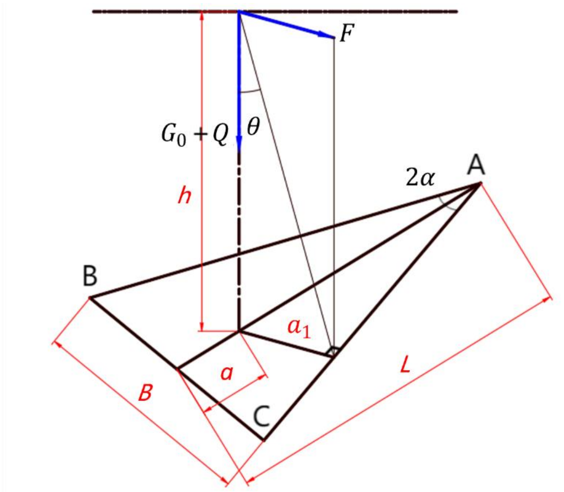


**Figure 7** Diagram of the Lateral Static Stability of the Loaded Vehicle

When the vehicle reaches the critical state of tipping, the component of centrifugal force $F$ and the total gravity of the loaded vehicle $(G_0 + Q)$ satisfy the following relationship:

$$F \cdot h = (G_0 + Q)a_1 \tag{7}$$

In the formula, $a_1$ represents the vertical distance from the synthetic center of gravity to the first-order lateral overturning axis AC.

When the lateral static stability requirement of the vehicle is met:

$$\frac{a_1}{h} > 0.06 \tag{8}$$

Substituting the parameters into the equation yields: $a_1 = 177.87\ \mathrm{mm}$ and $h = 700\ \mathrm{mm}$. The requirement for lateral static stability is satisfied.

### 3.4 Lateral Dynamic Stability Analysis

The operating condition for the lateral dynamic stability analysis of the AGV corresponds to sharp turning under unladen conditions. Considering the influence of other factors on the vehicle, it is specified that:

$$\frac{a_2}{y_0} > (15 + 1.4V)\% \tag{9}$$

Where:

$a_2$ — Perpendicular distance from the projection of the unladen vehicle's CoG to the primary lateral rollover axis AC;

$V$ — Maximum travel speed of the AGV, set to $0.8\ \mathrm{m/s}$.

From Eq. (7):

$$a_2 = (L - x_0)\frac{B}{\sqrt{B^2+4L^2}} \tag{10}$$

Substituting the numerical values yields $a_2 = 177.87$ mm. The requirement for lateral dynamic stability is satisfied.

## 4 CONCLUSION

This paper conducts theoretical analyses on the proposed AGV in terms of structural design, and anti-rollover stability. The primary achievements and conclusions are summarized as follows:

Based on the design objectives, a five-wheel steerable drive system was adopted for the mobile platform. Furthermore, a shock absorption mechanism was designed, and the appropriate spring parameters were selected. The lifting system incorporates a single-stage mast, hook-type forks mounted on a plate-type fork carriage, and a motor-driven timing belt-pulley powertrain equipped with a 500 W drive motor and an 8M timing belt/pulley configuration (30 mm belt width).

Using the Center of Gravity (CoG) method, the stability of the AGV was rigorously analyzed and verified across four scenarios: static longitudinal stability during high-lift conditions, dynamic longitudinal stability during travel braking, static lateral stability during low-speed turning, and dynamic lateral stability during full-speed turning under unladen conditions. The calculation results confirm that the designed vehicle satisfies all stability requirements.

## COMPETING INTERESTS

The authors have no relevant financial or non-financial interests to disclose.

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