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CQU-UC Joint Co-op Institute (JCI) Student Project Report

Final project of theory of machines and mechanisms

Front drive tricycle

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ABSTRACT

This report designs a front drive tricycle for the enterprise, which mainly includes the steering system, differential and its driving characteristics and weight distribution analysis. In the part I, the report introduces the steering system of the front drive tricycle from structural composition of mechanical steering system, comparison of the advantages and disadvantages of the three steering systems, plane structure and degree of freedom analysis, kinematics and dynamics analysis. In the part II, the report introduces differential mechanism analysis from scheme comparison, degree of freedom, kinematic and dynamic analysis. In the part III, the report introduces characteristic and weight distribution analysis respectively.

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Part I: Steering System Analysis

1. Structural composition of mechanical steering system

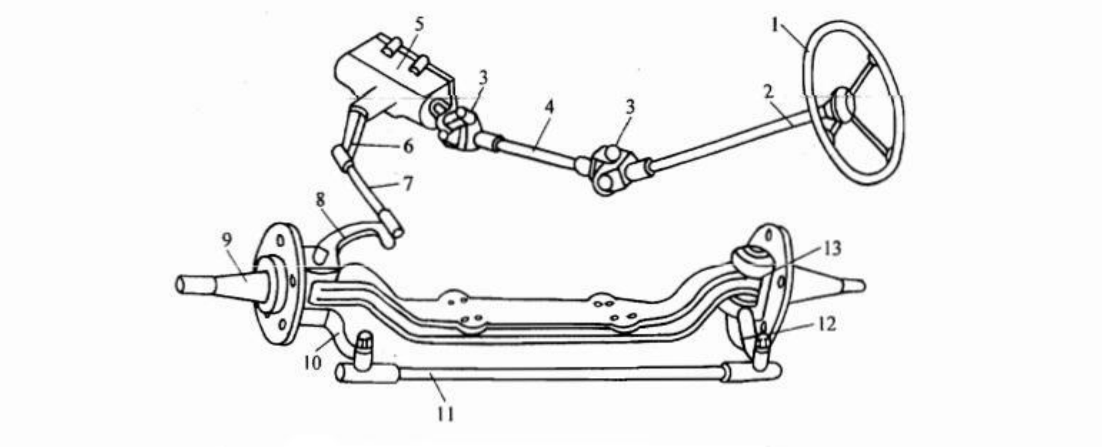

Fig. 1. 3D structure of mechanical steering system

1- Steering wheel 2-Steering shaft 3-Steering universal joint 4-Steering transmission shaft 5-Steering gear 6-Steering rocker arm 7-Steering tie rod 8-Steering knuckle arm 9- Left steering knuckle 10,12- Trapezoidal arm 11- Steering tie rod 13- Right steering knuckle

The properties of steering system:

1. There are various structural forms of automobile steering systems, but they all include three basic components: steering mechanism, steering gear, and steering transmission mechanism.

2. The function of the steering mechanism is to transmit the steering force of the driver turning the steering wheel to the steering gear. It is mainly composed of steering wheel, steering shaft, steering column, steering universal joint, etc.

3. The steering gear is a mechanism that converts the rotation of the

steering wheel into the swing of the steering rocker arm or the linear reciprocating motion of the rack shaft, and amplifies the steering force.

4. The steering transmission mechanism is a mechanism that transmits the force and motion output by the steering gear to the wheels (steering knuckle), and makes the left and right wheels deflect according to a certain relationship. The steering transmission mechanism is composed of a steering straight tie rod, a steering knuckle arm, a steering trapezoidal arm, and a steering tie rod.

2. Comparison of three steering systems

1. Mechanical steering system: high reliability and relatively cheap parts, but the output steering torque is relatively small.

2. The power steering system is formed by adding a set of steering assist device on the basis of the mechanical steering system. Most of the steering energy is provided by power-assisted devices, which mainly include hydraulic power-assisted systems, electronically controlled hydraulic power-assisted systems, and electric power steering systems. It's easier and safer to drive.

3. The electric power steering system (EPS) can make the steering light and flexible at low speeds; when the car is turning in the middle speed area, it can ensure the optimal power magnification and stable steering feel, thereby improving the steering stability of high-speed driving. It's easier and safer to drive

3. Plane structure and degree of freedom analysis

In this project, we will focus on the mechanical transmission system. We will first discuss the structure of the mechanical steering system and then calculate its degrees of freedom.

3.1 Plane structure

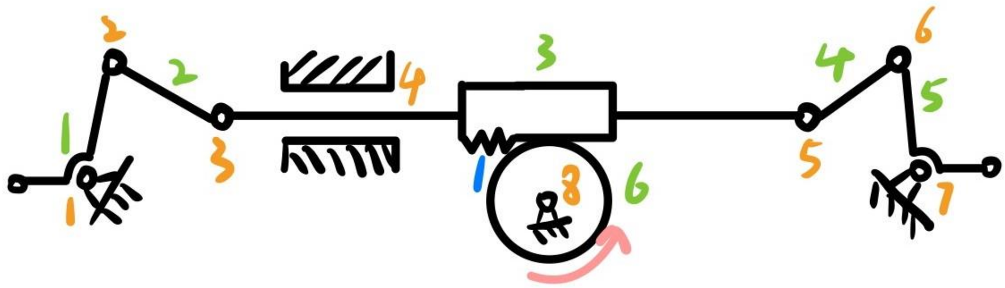

Fig. 2. Plane structure diagram of mechanical steering system

3.2 Degree of freedom

The structure diagram of the mechanical steering system is composed of 6 links, 7 rotating pairs, a moving pair and a high pair.

So the degree of freedom is :

$$Degree\ of\ Freedom = 3 \times n - 2 \times P_L - 1 \times P_H$$
$$= 3 \times 6 - 2 \times 8 - 1 \times 1$$
$$= 1$$

Among them, the component 6 is the driving component, which drives the component 3 to move left and right, then drives the components 2 and 4 to rotate, and finally drives the components 1 and 7 to rotate, thereby realizing the rotation of the wheels.

4. Kinematics and dynamics analysis

4.1 Kinematics analysis

4.1.1 Positive efficiency of steering gear η+

The factors that affect the positive efficiency of the steering gear include: the type, structural characteristics, structural parameters, and manufacturing quality of the steering gear.

Steering gear type, structural features and efficiency. Among the aforementioned four types of steering gears, rack and pinion type and recirculating ball type steering gears have higher positive efficiency, while worm pin type steering gears, especially fixed pin and worm roller type steering gears. The positive efficiency is significantly lower. The same type of steering gear has different efficiency due to different structures. For example, the bearing between the roller and the supporting shaft of the worm roller steering gear can be selected from one of three structures: needle roller bearings, tapered roller bearings and ball bearings. In the first structure, in addition to the friction loss between the roller and the needle, there is also a sliding friction loss between the side wing of the roller and the gasket, so the efficiency of this steering gear is only 54%. The diverter efficiency of the other two structures is 70% and 75% respectively according to the test results.

The form of the steering rocker shaft bearing also affects the efficiency.

The use of needle roller bearings can increase the forward or reverse efficiency by about 10% compared with the use of sliding bearings.

The structural parameters and efficiency of the steering gear. If the friction loss of the bearing and other places is ignored, only the friction loss of the meshing pair is considered. For the worm and screw type steering gear, the efficiency can be calculated by the following formula:

$$\eta = \frac{tan\alpha}{\tan{(\alpha - \rho)}} \quad (1)$$

In the formula, αo is the spiral lead angle of the worm (or screw); ρ is the friction angle, ρ=arctanf; f is the friction factor.

4.1.2 Steering ratio

The transmission ratio of the steering system includes the angular transmission ratio iw of the steering system and the force transmission ratio of the steering system ip.

The ratio of the resultant force 2Fw acting on the two steering wheels from the center of the tire contact surface to the hand force Fh acting on the steering wheel is called the force transmission ratio, namely ip=2Fw/Fh.

The ratio of the steering wheel rotation angular velocity ωw to the steering knuckle deflection angular velocity ωk on the same side is called the steering system angular transmission ratio, namely:

$$\mathrm{I}\omega o = \frac{\omega w}{\omega k} = \frac{d\varphi/dt}{d\beta k/dt} = \frac{d\varphi}{d\beta k} \quad (2)$$

In the formula, dφ is the steering wheel angle increment; dβk is the steering knuckle angle increment; dt is the time increment. It is composed of steering gear angular transmission ratio iw and steering gear angular transmission ratio iw′, namely iwo=iw/iw′.

The ratio of the steering wheel angular velocity ωw to the rocker shaft rotation angular velocity ωK is called the steering gear angular transmission ratio iw :

$$\mathrm{I}\omega= \frac{\omega w}{\omega p} = \frac{d\varphi/dt}{d\beta p/dt} = \frac{d\varphi}{d\beta p} \tag{3}$$

In the formula, dβp is the rotation angle increment of the rocker shaft.

The ratio of the rotational angular velocity ωp of the rocker arm shaft to the deflection angular velocity ωk of the steering knuckle on the same side is called the angular transmission ratio iw′ of the steering transmission mechanism, so the iw′ is :

$$\mathrm{I}\omega'= \frac{\omega p}{\omega k} = \frac{d\beta p/dt}{d\beta k/dt} = \frac{d\beta p}{d\beta k} \tag{4}$$

The angular transmission ratio of the steering transmission mechanism, in addition to being expressed by iw'=dβp/dβk, can also be expressed approximately by the ratio of the steering knuckle arm length L2 to the rocker arm length Ll, that is, iw'=dβp/dβki≈L2/ Ll. In modern automobile structure, the ratio of L2 to L1 is between 0.85 and 1.1, which can be approximated as iwo ≈ iw = dφ/dβ.

4.2 Dynamics analysis

There is the following relationship between the steering resistance Fw between the tire and the ground and the steering resistance torque Mr acting on the steering knuckle:

$$Fw = \frac{Mr}{a} \tag{5}$$

In the formula, α is the main pin offset, which refers to the distance from the intersection of the extension line of the kingpin axis of the steering knuckle and the support plane to the intersection of the wheel center plane and the support plane.

The hand force Fh acting on the steering wheel can be expressed by the following formula:

$$Fh= \frac{2Mh}{Dsw} \tag{6}$$

In the formula, Mh is the torque acting on the steering wheel; Dsw is the diameter of the steering wheel.

Substituting formula (5) and formula (6) into ip=2Fw/Fh, we get:

$$Ip = \frac{Mr \times Dsw}{Mh \times a} \tag{7}$$

Analytical formula (7) shows that when the kingpin offset a is small, the force transmission ratio ip should be larger to ensure light steering. Usually, the value of a for cars is selected in the range of 0.4 to 0.6 times the tire tread width, and the value of d for trucks is selected in the range of 40 to 60 mm. The steering wheel diameter Dsw is selected from the series specified in the JB4505—86 steering wheel size standard according to different models .

If the friction loss is neglected, according to the principle of conservation of energy, 2Mr/Mh can be expressed by the following formula:

$$\frac{2Mr}{Mh} = \frac{d\varphi}{d\beta k} = \mathrm{I}\omega o \tag{8}$$

Substituting equation (8) for human equation (7), we get :

$$\mathrm{Ip} = \frac{iwo \times Dsw}{2a} \tag{9}$$

When α and Dsw do not change, the larger the force transmission ratio ip, the lighter the steering, but the larger iwo, indicating that the steering is not sensitive.

Part II: Differential Mechanism Analysis

1. Scheme comparison

1.1 Gear differential

Common bevel gear differential is a differential that evenly distributes torque to the left and right wheels. According to the characteristics of output torque, it can be divided into two types: symmetrical conical planetary gear differential and asymmetrical conical planetary gear differential. In addition to bevel gears, there are cylindrical gears.

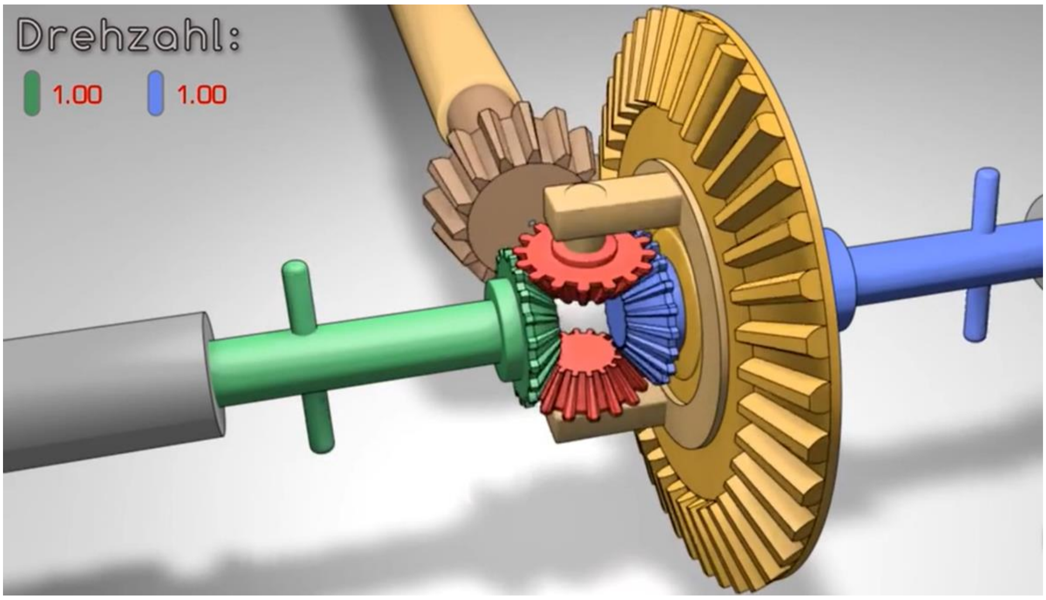

Fig. 1. The gear differential

The open differential is the most commonly used, which can distribute the same amount of torque to the left and right drive axles. When the vehicle is traveling in a straight line, the left and right wheels bear equal forces, and there is no difference in speed between the two half shaft gears, so the planetary gear does not rotate, and the main reducer driven ring gear is equivalent to directly driving the two half shaft gears. The half shaft gear is connected to the wheel through the drive half shaft, so after a series of power transmission processes, the wheel obtains the same speed as the driven ring gear of the final drive. When the vehicle is turning, the outer wheels hope to obtain a higher speed than the inner wheels. At this time, the planetary gear intervenes to allow a slight speed difference between the two half shaft gears while maintaining the torque transmission.

1.1.1 Advantage

The structure of the gear differential is simple, and the design cost and manufacturing cost are low. At the same time, it has a high degree of reliability and is not easily damaged. It is a much commonly used differential structure.

1.1.2 Shortcoming

Due to structural reasons, this differential distributes the same torque to the left and right wheels. The torque averaging characteristic of this differential can satisfy the normal driving of the car on a good road. But when the car is driving on a bad road, it seriously affects the passing ability. For example, when one driving wheel of a car is stuck on a muddy road, although the other driving wheel is on a good road, the car often cannot move forward (commonly known as skidding). At this time, the driving wheel on the muddy road slipped in place, but the wheel on a good road was still. This is because the adhesion between the wheels on the muddy road and the road is small, and the road can only exert a small reaction moment on the axle through this wheel, so the torque distributed by the differential to this wheel is also small. Although the adhesion between the other driving wheel and the good road surface is relatively large, due to the characteristics of evenly distributed torque, this driving wheel can only be divided into the same amount of torque as the slip driving wheel, so that the driving force is not enough. Overcoming the driving resistance, the car cannot move forward, and the power is consumed on the slip driving wheels. At this time, increasing the throttle will not only cause the car to move forward, but it will also waste fuel and accelerate the wear of parts, especially tire wear. The effective solution is to dig out the thin mud under the slip driving wheel or put dry soil, gravel, branches, hay, etc. under this wheel.

1.2 Torsen limited slip differential

Torsen differential, also known as Torsen self-locking differential, uses the principle of irreversibility of worm gear transmission and high friction conditions on the tooth surface to make the differential according to its internal differential torque (That is, the internal friction torque of the differential) is automatically locked or released, that is, when the differential torque in the differential is small, it will act as a differential, and when the differential torque in the differential is too large, the differential The device will automatically lock, which can effectively improve the car's passing capacity.

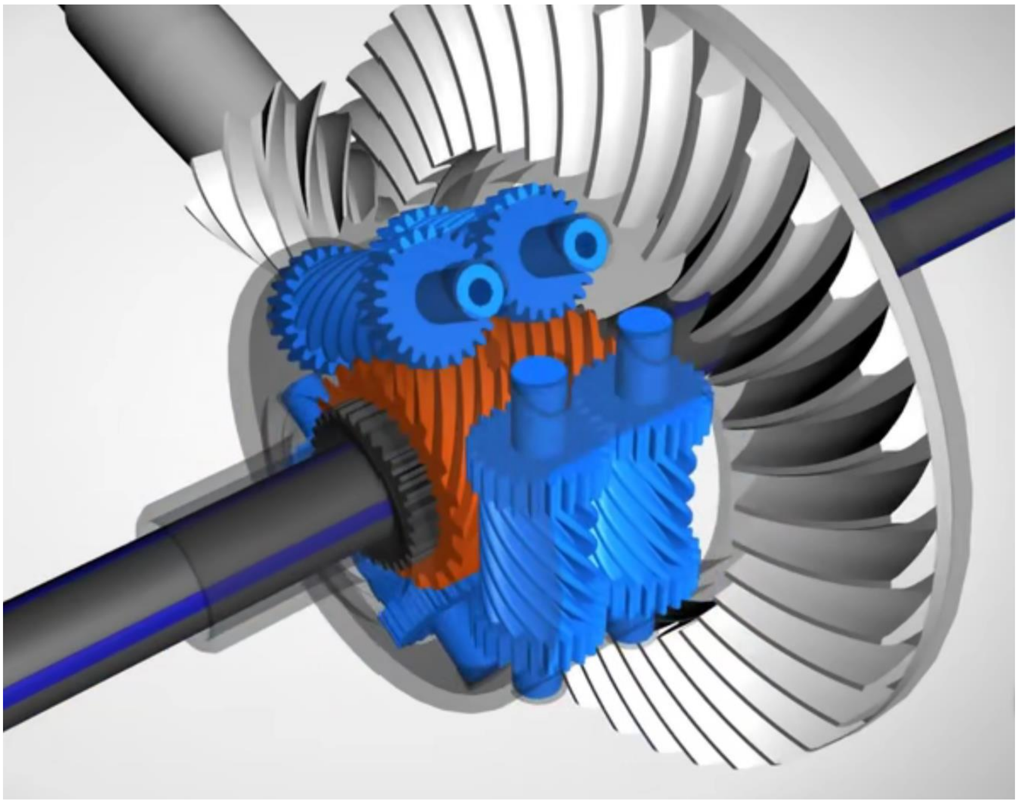

Fig. 2. The Torsen limited slip differential

1.2.1 Advantage

The Torson differential realizes constant-time and continuous torque control management. It works continuously without time delay but does not intervene in the adjustment of the total torque output, so there is no torque loss. It is in line with traction control and body stability control.

Compared with the system, it has greater advantages. Because there is no multi-disc clutch equipped with a traditional self-locking differential, there is no wear and no maintenance. Pure mechanical LSD has good reliability.

The Torson differential can be matched with any transmission and transfer case, and is compatible with other vehicle safety control systems ABS, TCS (Traction Control Systems, traction control), and SCS (Stability Control Systems, body stability control). The Torsen differential is a purely mechanical structure, it will take effect as soon as the wheel slips, and it has linear locking characteristics.

Torson type limited slip differential is a fully automatic and purely mechanical limited slip differential, which is very reliable and durable, and responds quickly. From some perspectives, it is a very balanced design. It can respond to the torque difference between the driving wheels in a very short time, adjust the torque output to solve the problem of wheel difference, and the locking characteristics are also very linear, and can be adjusted within a relatively wide torque range. , Without being affected by the structural space of the differential case and restricting the play of its role.

1.2.2 Shortcoming

However, compared with other torque-sensing limited-slip differentials, Torson type limited-slip differentials are relatively complex in structure, heavy in weight, and relatively expensive. At the same time, the high internal friction torque of worm gear and worm transmission also promote component aging, which is detrimental to the service life.

2. Degree of freedom

In this project, we will focus on the gear differential. We will first discuss the structure of the gear differential and then calculate its degree of freedom.

2.1 Structure of gear differential:

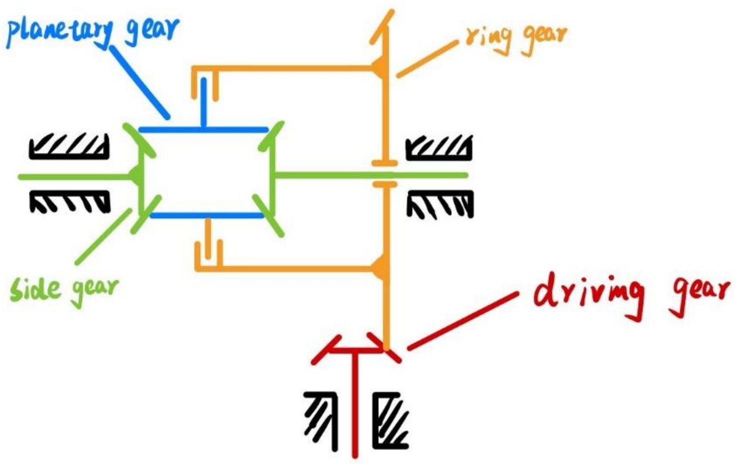

Fig. 3. The basic structure of gear differential

The gear differential can be divided into four parts. They are drive gears, ring gears, planetary gears and side gears. The power enters from the drive gear, passes through the ring gear and planetary gear, and then is transmitted to the side gear. The side gears ultimately output power to the wheels.

2.2 Degree of freedom calculation:

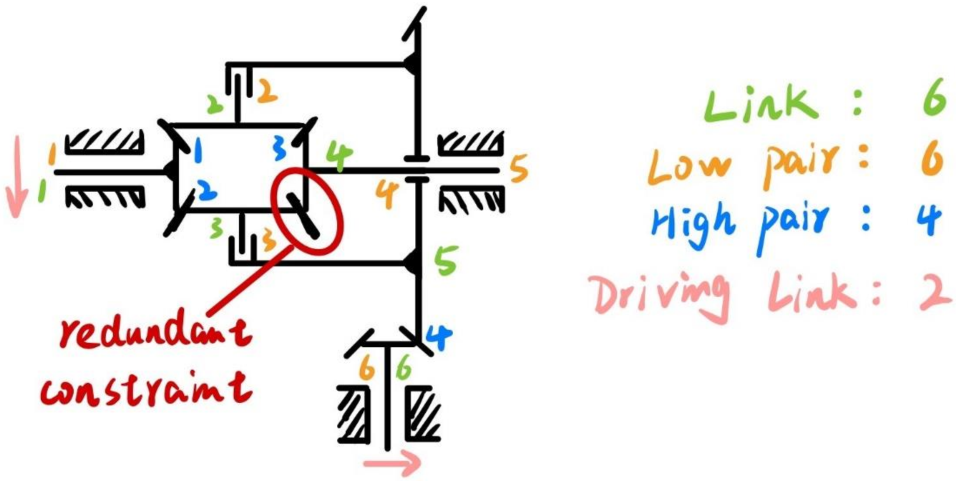

Fig. 4. The DOF calculating of gear differential

From the figure we know that: there are 6 links, 4 low pairs and 2 high pairs. Based on the degree of freedom calculation formula, we have:

$$
\begin{aligned}
Degree\ of\ Freedom &= 3 \times n - 2 \times P_L - 1 \times P_H \\
&= 3 \times 6 - 2 \times 6 - 1 \times 4 \\
&= 2
\end{aligned}
$$

In order to verify the correctness of the degree of freedom calculation, we also marked the number of drive components. The driving part is 2. They are the driving gear and the side gear driven by ground friction respectively.

3. Kinematic and dynamic analyses

3.1 When vehicle is traveling straight

When driving in a straight line, the resistance of the left and right driving wheels is roughly the same. The power output from the engine is first transmitted to the differential housing to start the rotation of the differential housing, and then the power is transmitted from the housing to the left and right half shafts. We can understand that the side gears on both sides "compete" with each other. Since the resistance of the wheels on both sides is the same, neither of the two can break each other. Therefore, the planetary gears in the differential housing revolve with the housing without rotating at the same time. Rotate at speed so that the car can drive in a straight line!

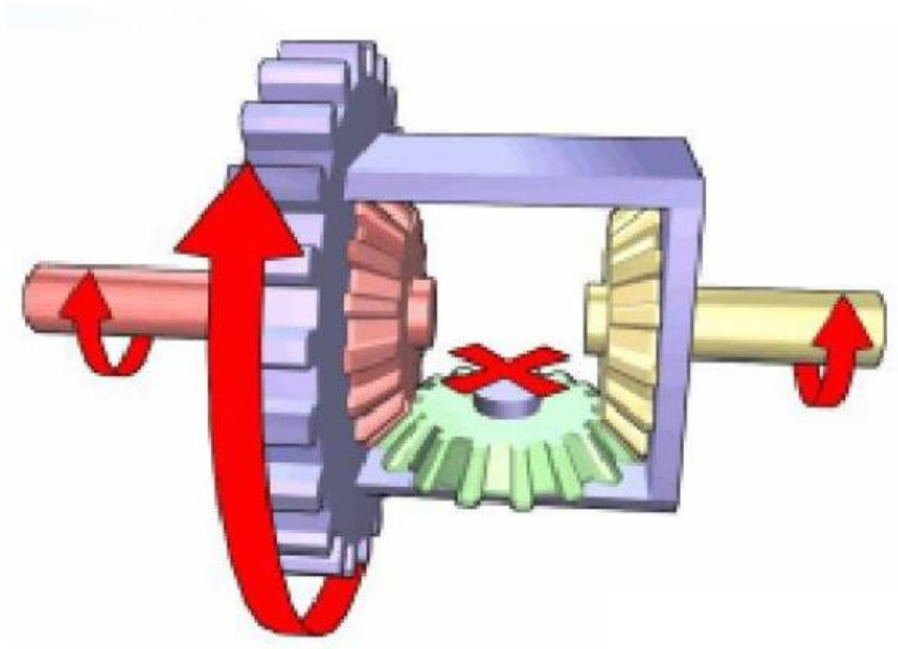

Fig. 5. The state of differential when vehicle traveling straight

3.2 When vehicle is traveling at differential speed

Assuming that the vehicle is now turning to the left, the distance traveled by the left-hand drive wheel is relatively short, and relatively speaking, more resistance will be generated. The differential housing is connected to the output shaft by a gear. The speed of the differential housing does not change when the speed of the drive shaft is unchanged. Therefore, the left side gear will rotate more slowly than the differential housing. It is equivalent to that the planetary gear drives the left side shaft to be more laborious. At this time, the planetary gear will generate autobiography, which will transmit more torque to the right-side half shaft gear. Due to the revolution of the planetary gear plus its own autobiography, the right-side half shaft will be caused. The axle gear will increase the speed on the basis of the speed of the differential housing, so that the right wheel will turn faster than the left wheel, so that the vehicle can achieve a smooth turn.

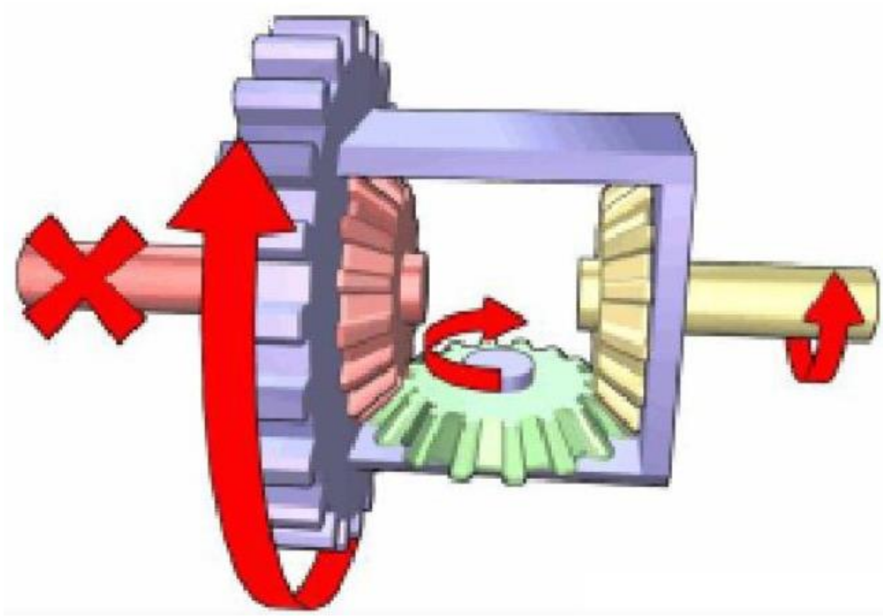

Fig. 6. The state of differential when vehicle traveling at different speed

3.3 Gear kinematic analysis

As we discussed in section 2.1, the structure of gear differential is shown as following. ω_1, z1, z2 are given and z6 = z5.

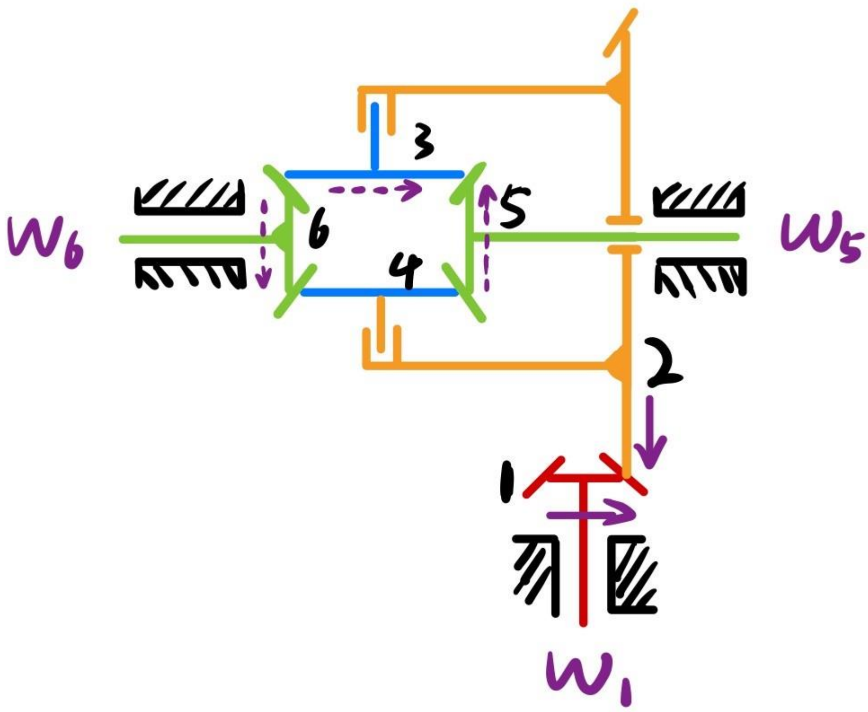

Fig. 7. The kinematic analysis of gear differential

For gear 1 and 2, they constitute a couple of fixed axis gears. Hence, we have:

$$i_{12} = \frac{n_1}{n_2} = \frac{z_1}{z_2} \tag{1}$$

For gear 6, 3, 2 and 5, they constitute epicyclical gears. Hence, we have:

$$i_{65}{}^{2} = \frac{n_6 - n_2}{n_5 - n_2} = -\frac{z_5}{z_6} = -1 \tag{2}$$

Thus, we have:

$$n_5 + n_6 = 2n_2 \tag{3}$$

After simplifying and plug i_{12}, we have:

$$\omega_5 + \omega_6 = \frac{2z_2}{z_1}\omega_1 \tag{4}$$

Overall, we can obtain some conclusions. As long as we have two of ω_1, ω_5 and ω_6, we can get the third one. Meanwhile, when one side of the half shaft does not rotate, the other side of the half shaft will rotate at twice the angular velocity of the differential housing; when the differential housing does not rotate, the left and right half shafts will rotate in opposite directions at the same speed.

3.3 Gear dynamic analysis

The figure of dynamic analysis is shown as following. T_2, T_5, T_6 are the torque exerted by link 2, 5, 6 respectively.

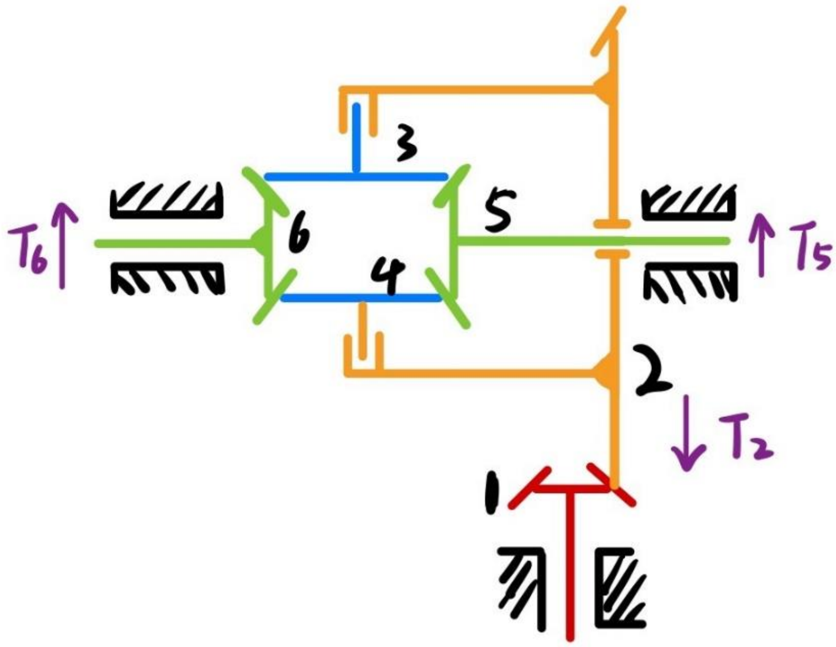

Fig. 8. The dynamic analysis of gear differential

According to the torque balance conditions, we have:

$$T_5 + T_6 = T_2 \tag{5}$$

Meanwhile, the performance of the differential is often characterized by the locking coefficient k, which is defined as the ratio of the internal friction torque of the differential to the torque received by the differential housing, namely:

$$k = \frac{T_5}{T_6} \tag{6}$$

Then we have:

$$\begin{cases} T_5 = \frac{T_2(1-k)}{2} \\ T_6 = \frac{T_2(1+k)}{2} \end{cases} \tag{7}$$

Generally speaking, the locking coefficient of ordinary bevel gear differentials is 0.05~0.15, which shows that the torque difference between the left and right half shafts is not large, so it can be considered that the torques distributed to the two half shafts are roughly equal.

Part III: Characteristic and weight distribution analysis

1. Characteristic analysis

1.1 Driving characteristics

This product adopts high-power AC electric precursor device, so the driving characteristics are also related to the parameters of the device. Since the driving device is located at the front, the body will have better restoring force (compared with rear driving) in the face of external disturbance during movement, so as to have better stability. Therefore, in the process of linear motion, the front drive vehicle can play a better role in the external disturbance of the body (such as crosswind) and the road (such as rugged road and snow road). At the same time, due to the complex structure of the front drive device and many transmission processes, it is more prone to idle vibration than the rear drive vehicle.

1.2 Weight characteristics

The weight distribution of the front-wheel drive vehicle is mainly affected by the position of the driving equipment (battery, motor, gearbox, differential). Since this concept product adopts the integrated drive and rotation (high-power AC asynchronous motor front drive) solution, the weight of the drive equipment is basically located near the front wheels . In addition, because the tricycle has one less load wheel than the four wheels, the load on a single wheel will increase. Therefore, the weight of the front wheel of this product will be much higher than that of a four-wheel drive under the same load. A heavy front wheel will result in the following characteristics:

1. The straight and stable track is enhanced, and the restoring force of outward disturbance is large

2. The wheel side resistance is weak, the steering to the United States (steering) is enhanced, the sports state is worse, and the steering under high G is easy to drift.

3. The front wheel brake device is large, and the front wheel heat transfer device is easy to produce braking noise, so it is necessary to analyze and control the noise of the front drive

4. In the process of steering and braking, the front wheels are more easily affected

1.3 Turning characteristic

Due to the weight characteristics, the front wheel side rigidity is reduced and the US (under steering) is increased. The following problems will occur during the steering process:

1. Flowing phenomenon is prone to appear in the high G state, so the horizontal G limit value is reduced

2. When the throttle is released during a turn, the head swings instantly, so there should be no sudden changes in the acceleration during the turning process, so as to prevent the body from losing control.

3. When the drive shaft has a bent angle, torque steering problems will occur in the high G state, thereby increasing the difficulty of the driver's operation, and driving fatigue during long distances

4. Due to the front of the engine, the front wheel tire angle of front-wheel drive vehicles is limited, which will cause the minimum turning radius to become larger. Due to the particularity of the electric front-wheel drive device of this product, it is less affected by this reason.

2. Weight distribution recommendations

When the vehicle is stationary, according to the general layout parameters, the radius of the triangle circumscribed by the three wheels is 1422, so when the center of gravity is 900 from the front wheels, the three-wheel suspension load is the same. At this time, the position of the center of gravity is 33:67 (the center of gravity and the wheelbase of the front wheels) : Center of gravity and rear wheel base). At this time, the total load of the front wheels is twice that of the rear wheels. In order to balance the load of the front and rear wheels, the center of gravity needs to be shifted back when the tricycle is designed. When the load of the rear wheels is twice that of the front wheels, the sum of the load of the front wheels is equal to the load of the rear wheels, and the center of gravity is 1350 from the front wheels.

In the state of motion, the center of gravity will shift according to the movement mode. The offset depends on the suspension stiffness and the height of the center of gravity. Therefore, the center of gravity of the body should be reduced as much as possible during the design process to improve vehicle performance. During exercise, the distribution of the center of gravity has a greater impact on operability. We analyze the three typical motion processes of acceleration, braking, and steering, and calculate the impact of the center of gravity.

2.1 Boot process

During the acceleration process, the overall center of gravity shifts back. Due to the two-wheel front drive, the shift of the center of gravity will reduce the grip of the front wheels, but it will not change the body posture. Because the characteristics of the front-wheel drive vehicle lead to a slow starting speed, it is necessary to step on the accelerator slowly to improve the grip of the driving wheels. The theoretical analysis is as follows.

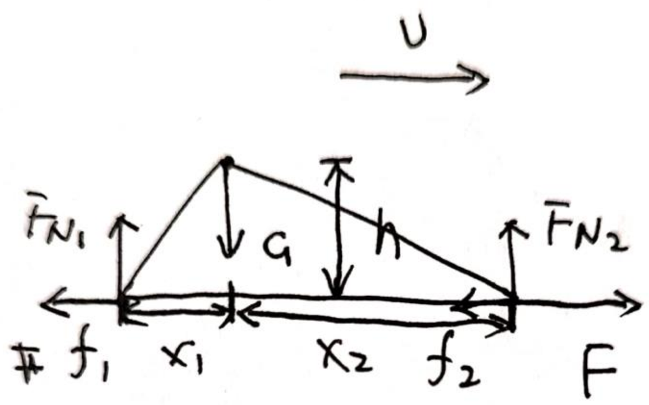

Fig. 1. The sketch map of boot process

Set the height of the center of gravity h, the body length L, the distance between the center of gravity and the front wheel x_2, and the distance between the rear wheel x_1, μ is the ratio of the driving force F to the supporting force F_N, μ_0 is the friction coefficient of resistance, F is the driving force, and the vehicle gravity is G. Through kinetic analysis, we can get

$$F_{N2} = \frac{G}{L+\mu h} x_1 + \frac{\mu_0 h G}{L+\mu h} \tag{1}$$

Due to $\mu_0 h << x_1$

$$F_{N2} = \frac{G}{L+\mu h} x_1 \tag{2}$$

$$F = \frac{G}{h} x_1 - \frac{L}{h} F_{N2} \tag{3}$$

Thus, the driving force F is proportional to x_1. And when x_1 is a fixed value, the greater the driving force, the smaller the front wheel load, which is in line with the above conclusion that the accelerator pedal is stepped slowly to improve the grip of the driving wheels.

2.2 Braking process

During braking, the overall center of gravity moves forward and the front wheel load increases, which is the main brake wheel. Due to excessive braking acceleration during emergency braking, the vehicle body may lean forward. Therefore, the center of gravity must meet the following conditions to ensure driving safety. The theoretical analysis is as follows.

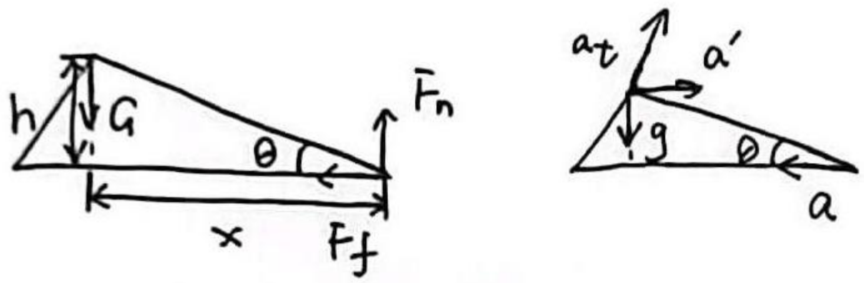

Fig. 2. The sketch map of braking process

Set the height of the center of gravity h, the distance between the center of gravity and the front wheel x, the gravity of the whole vehicle is G, and μ is the sliding friction coefficient. Through kinetic analysis, we can get.

$$\frac{h}{x} = \frac{1}{\mu} \tag{4}$$

Normally worn tires have a sliding friction coefficient μ=0.6 when driving on the road. Therefore, in order to avoid the dangerous state of the car body tilting forward, h/x<1.67 needs to be satisfied, which can be improved by lowering the center of gravity or increasing the distance between the center of gravity and the front wheel.

2.3 Steering system

During the steering process, usually the steering system of the car is placed on the front wheels, which is prone to accidents where the rear wheel grip is insufficient and oversteer. For four-wheel front-wheel drive vehicles, the rear wheel grip will be increased by shifting the center of gravity of the vehicle to improve steering. Excessive situation. But for tricycles, as long as the offset center of gravity is between 33:67 and 50:50, the grip of the rear wheels will always be larger than the front wheels, so that the steering will not lose control due to the small friction wheels of the rear wheels. The front-wheel drive vehicle does not need to shift the center of gravity backward to improve the steering deviation problem. This is also a major advantage of the three-wheel front-wheel drive racing car. The theoretical analysis is as follows.

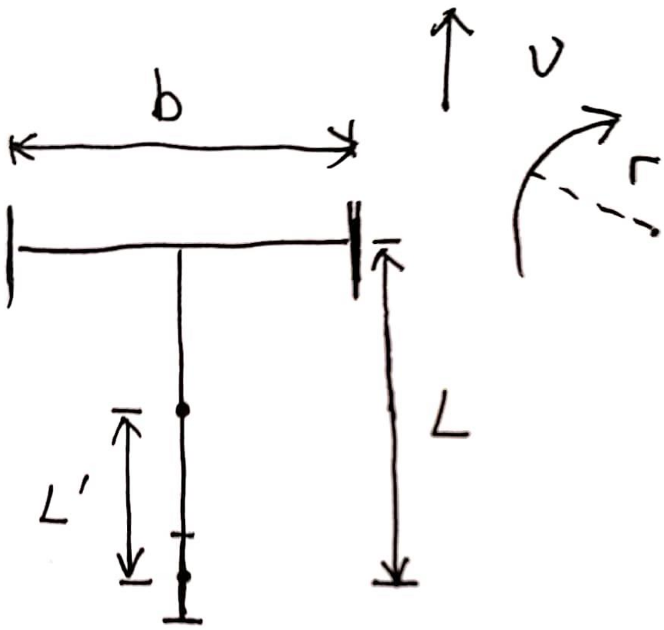

Fig. 3. The sketch map of steering system

Suppose the body length is L, the mass is m, the center of mass is from the rear wheel L', the height is h, and the vehicle width is b. The three-wheel loads F1, F2, and F3 are respectively.

$$F_1 = \left(mg\frac{b}{2}\frac{L}{L'} + mh\frac{v^2}{r}\sqrt{1-(\frac{b}{2L})^2}\right)\frac{\sqrt{L^2+\frac{b^2}{4}}}{bL} \tag{5}$$

$$F_2 = \left(mg\frac{b}{2}\frac{L}{L'} - mh\frac{v^2}{r}\sqrt{1-(\frac{b}{2L})^2}\right)\frac{\sqrt{L^2+\frac{b^2}{4}}}{bL} \tag{6}$$

$$F_3 = mg\left(1-\frac{L'\sqrt{L^2+\frac{b^2}{4}}}{L}\right) \tag{7}$$

In summary, the best center of gravity of the vehicle during the movement should be between 33:67 and 50:50. Assuming that the vehicle's emergency braking acceleration is 1.5 times the starting acceleration, if the front and rear wheel suspensions have the same horizontal stiffness, the braking center of gravity offset is 1.5 times the starting offset, so the best center of gravity position at rest should be 61: 39. But if you want to improve the acceleration performance of the vehicle, you can increase the horizontal stiffness of the front suspension, and then advance the center of gravity.

So the recommended center of gravity range is 55:45 to 65:35

*The limit value of the height of the center of gravity: $1.67 \times 1350 = 2255$ (much greater than the height of the vehicle, so it has no reference value)