A Novel Underactuated Cam Mechanism

Zhoudong Yan[1](✉), Xinbo Chen[1], and Wei Wang[2]

[1] Tongji University, Shanghai 201804, China
yan_zhou_dong@163.com
[2] Tokyo University of Agriculture and Technology, Tokyo 184-8588, Japan

Abstract. In order to obtain an efficient and strong load-bearing clamping mechanism, this paper proposes a novel underactuated cam mechanism through the mechanism evolution method. This mechanism combines the advantages of the negative radius roller cam mechanism and has the transmission characteristics of the flat-bottomed cam, making it a good actuator for clamping. Finally, a set of conceptual prototypes is given in this paper when the mechanism is applied to the vehicle electronic mechanical brake (EMB) actuator.

Keywords: Clamping mechanism · Cam mechanism · Underactuated mechanism

1 Introduction

With the development of electric vehicles, the need for new brake actuators are improved urgently. In the process of vehicle braking, small displacement and large force are required. It is usually realized by actuators such as hydraulic pressure and air pressure. However, the structure of hydraulic and pneumatic power sources is complex with large volumes, and the control execution effect is poor. The EMB uses motors and mechanisms to achieve clamping, and is now a research focus [1]. In theory, EMB has advantages such as faster response speed, continuous and more accurate control of clamping force. Besides, each actuator can be controlled separately and there is no risk of leakage pollution [2]. But how to design a suitable clamping mechanism as an actuator for EMB is one of the main difficulties. The clamping mechanisms need to realize large load force and small deformation often with nonlinearity [3]. At the same time, the space between the wheels and rims is limited. So, the EMB needs to have a large transmission ratio, high efficiency, small volume, and a long lifespan.

2 Existing Clamping Mechanism for EMB

The main clamping mechanism currently used on EMB is the ball screw mechanism. But there are some mechanisms that can achieve the same function are enlightening and worth referring. The new cam mechanism proposed in this paper is invented and inspired by them.

© The Author(s), under exclusive license to Springer Nature Switzerland AG 2024
M. Okada (Ed.): IFToMM WC 2023, MMS 149, pp. 826–833, 2024.
https://doi.org/10.1007/978-3-031-45709-8_80

Screw mechanism [4]**.** The relationship between the displacement and rotation angle output is shown in Eq. (1). The screw mechanism is a strict linear transmission mechanism.

$$s = \frac{p}{2\pi}\varphi \tag{1}$$

where, s is the output displacement, p is the screw lead and φ is the input rotation angle.

The pressure angle is the angle between the direction of force and the direction of motion on the follower motion pair. As shown in **Fig. 1a**, according to the definition, the pressure angle of the lead screw mechanism is

$$\alpha = \arctan\left(\frac{\pi d}{p}\right) \tag{2}$$

where, α is the pressure angle and d is the pitch diameter of lead screw.

According to Eqs. (2) and (1), with the larger lead of the lead screw, the pressure angle becomes smaller. However, the bearing capacity and the volume of the ball screw is positively correlated with the size of the lead, so it is difficult to design a ball screw with a small size, large bearing capacity, and high efficiency.

The crank slider mechanism [4]**.** As shown in **Fig. 1b**, the relationship between the input angle and output displacement of the crank slider mechanism is

$$s = l_1 \cos\varphi + \sqrt{l_2^2 - (l_1 \sin\varphi + e)^2} \tag{3}$$

where, l_1 is the length of the crank, l_2 is the length of the connecting rod and e is the offset.

The pressure angle on the follower of the crank slider mechanism is

$$\alpha = \arcsin\left(\frac{l_1 \sin\varphi + e}{l_2}\right) \tag{4}$$

It can be seen that the pressure angle of the crank slider is related to its every parameter. When the crank length and offset is determined, the longer the connecting rod length is, the smaller the pressure angle is, but the larger the size of the mechanism is.

Sinusoidal mechanism [4]**.** When the length of the connecting rod of the crank slider mechanism is infinite $l_2 \to \infty$, the rotating pair is equivalent to the moving pair, and the crank slider mechanism becomes a sinusoidal mechanism. Its input-output relationship is shown in Eq. (5). According to Eq. (4), the pressure angle becomes zero.

$$s = l_1 \cos\varphi \tag{5}$$

Connecting rod mechanism is often difficult to reduce the volume under a large load. However, the idea of evolving from a crank slider mechanism to a sinusoidal mechanism is enlightening for the proposal of a new cam in this paper.

Negative radius roller cam mechanism. The negative radius roller cam mechanism was put forward by Italian scholar R. Garziera in 1997 [5]. The follower of the negative radius roller cam mechanism is the inner concave surface of an arc. The advantage of this mechanism is that the follower is a bearing, so there is no sliding friction on the cam surface, and the cam and roller have small internal contact stress and strong bearing capacity.

The motion characteristics of negative radius roller cam mechanisms $s = s(\varphi)$ can be designed flexibly in theory, and the overall size changes are not significant. Besides, using the instantaneous center method [6], the pressure angle can be calculated as

$$\alpha = \arctan\left(\frac{|ds/d\varphi - e|}{s_0 - s}\right) \tag{6}$$

where, $s_0 = \sqrt{r_r - r_{\min}^2 - e^2}$, r_r represents the inner diameter of the roller and $r_{\min}$ is the radius of the cam base circle.

For any variable transmission characteristic, it is difficult to ensure that $|ds/d\varphi - e| = 0$ is always true. So, the negative radius roller cam pressure angle is not zero under most conditions, and there is lateral force. Increasing the inner diameter of the roller can reduce the pressure angle, but according to the Hertz formula in Eq. (7), the larger the difference between the curvature radius of the cam and the roller radius, the greater the contact stress, and the bearing capacity will become worse.

$$\sigma_H = \sqrt{\frac{F\left(\frac{1}{\rho_1} - \frac{1}{\rho_2}\right)}{\pi L\left(\frac{1-\mu_1^2}{E_1} + \frac{1-\mu_2^2}{E_2}\right)}} \tag{7}$$

In the above equation, σ_H is the contact stress, F is the normal pressure, ρ_1 and ρ_2 are the curvature radii at the contact point, L is the length of the contact line, μ_1 and μ_2 are Poisson's ratio, E_1 and E_2 are Young's modulus.

Flat-bottomed cam mechanism [4]**.** Although the negative radius roller cam mechanism has many advantages, obviously there is still room for improvement in transmission efficiency because of the existence of pressure angle. As shown in **Fig.** 1 d), using the method of high pair and low joint substitution, the negative radius roller cam mechanism can be instantly equivalent to a crank slider mechanism. The evolution method similar to the crank slider can amplify the diameter of the driven roller to infinity, resulting in a flat-bottomed cam mechanism.

However, compared to the negative radius roller cam mechanism, it's difficult to use rolling elements to modify a flat-bottomed cam for continuous rotation. In addition, according to the Eq. (7), the contact stress of flat-bottomed cam mechanism is bigger than negative radius roller cam mechanism under the same load. To avoid introducing the aforementioned weaknesses, this article proposes a new type of cam mechanism.

3 New Cam Mechanisms

By using the method of mechanism evolution, to obtain a cam mechanism that is instantly equivalent to a sinusoidal mechanism, a flat-bottomed cam mechanism is obtained. However, the flat-bottomed cam mechanism is not ideal enough, and some features of the negative radius roller cam mechanism are worth preserving.

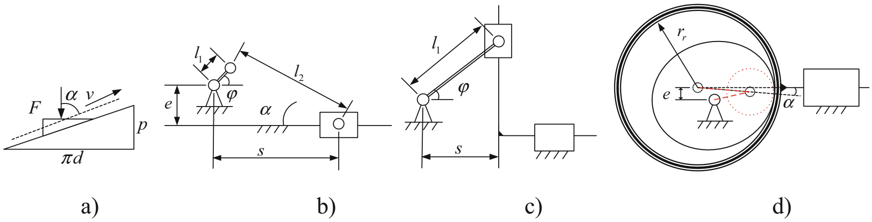


Fig. 1. Some existing mechanisms that can be used for EMB clamping. a) Pressure angle of lead screw mechanism; b) Crank slider mechanism; c) Sinusoidal mechanism; d) Negative radius roller cam mechanism

Fortunately, we find the force self-constraint characteristics between the negative radius roller and cam. As shown in **Fig.** 2, when the cam does not rotate, regardless of component gravity and friction, the position of the unconstrained roller and cam is uniquely determined under the force acting horizontally to the left. The red dotted line in **Fig.** 2 represents the theoretical profile of the negative radius roller cam. Under the action of horizontal force, the potential energy of the force F at the tangent point on the leftmost side of the theoretical profile is the lowest at the center of the roller circle.

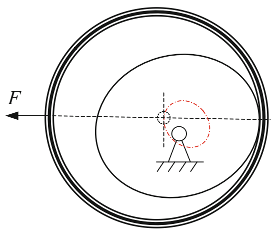
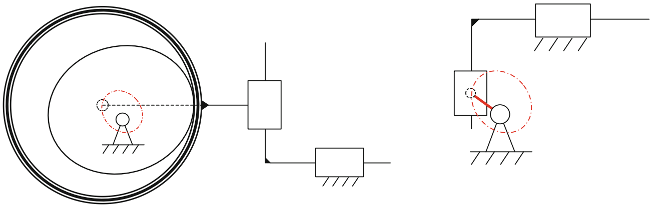


Fig. 2. Negative radius force self-constraint characteristics

The analysis shows that the roller has two degrees of freedom besides rotation, which is under constraint but can uniquely determine its position under the action of external force F. Because there is always a load force when the clamping mechanism works, we obtain another pathway for mechanism evolution, as shown in **Fig.** 3.

As shown in **Fig.** 3**a**, the calculated degree of freedom for the mechanism is also two. Under the action of load force, although the mechanism has two degrees of freedom but only one definite position state, the mechanism can be instantly equivalent to a Sinusoidal mechanism as shown in **Fig.** 3**b**.

It can be seen that the new cam mechanism can first be equivalent to a flat-bottomed cam mechanism with the theoretical profile of a negative radius roller cam as the working profile. And then the flat-bottomed cam mechanism can be equivalent to a sinusoidal mechanism with high pair and low generation. Therefore, the new cam mechanism has the same kinematic characteristics as the flat-bottomed cam mechanism, but it retains the advantages of the negative radius roller cam mechanism, such as large bearing capacity, and easy to achieve rolling contact. At the same time, it can achieve the same pressure angle as the flat-bottomed cam and high efficiency.



a) b)

Fig. 3. New cam mechanism. a) Schematic diagram; b) Schematic diagram of the mechanism after the high joint and low joint substitution.

4 Simulation Verification

To simplify the modeling of cam curves, an eccentric ellipse is used as the cam, and the dimensions of the cam and outer roller are shown in **Fig.** 4.

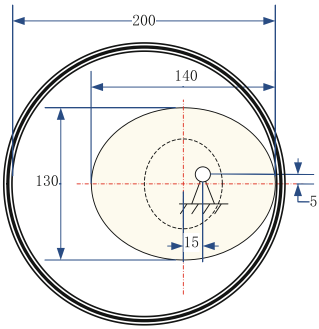
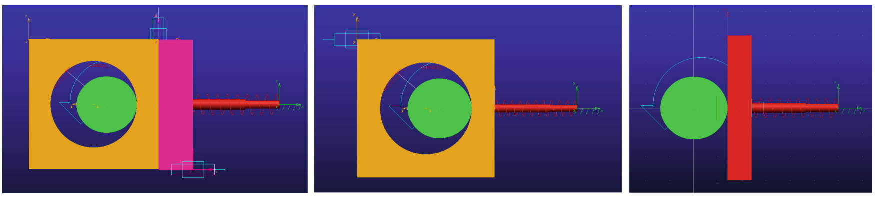


Fig. 4. Cam and roller dimensions

As shown in **Fig.** 5, we built simulation models of flat-bottomed cam mechanism, negative radius roller cam mechanism, and new cam mechanism using the same cam in ADAMS.



a) b) c)

Fig. 5. ADAMS simulation model. a) New cam mechanism; b) Negative radius roller cam mechanism; c) Flat-bottomed cam mechanism

Figure 6**a** shows that the motion characteristics of the new cam mechanism and the flat-bottomed cam mechanism are basically consistent. Because the motion position of the new cam mechanism is the lowest point of the load potential energy of the negative radius roller cam mechanism, its motion amplitude is always smaller compared to the negative radius roller cam mechanism. As shown in **Fig.** 6**b**, the contact force of the new cam mechanism is the same as the flat-bottomed cam mechanism. But according to the Hertz formula, the contact stress of the new cam mechanism is smaller. Compared to the negative radius roller cam mechanism, the contact force of the new cam mechanism is always relatively small. **Figure** 6**c** shows the calculated pressure angle, which can prove that the pressure angle of the new cam mechanism is always zero.

The oscillations of contact force and pressure angle in the simulation results is caused by the contact model based on *impact* in ADAMS simulation, which will be affected by the discrete calculation process. Both the cam and roller profiles are smooth, so the theoretical contact and pressure angle should be smooth.

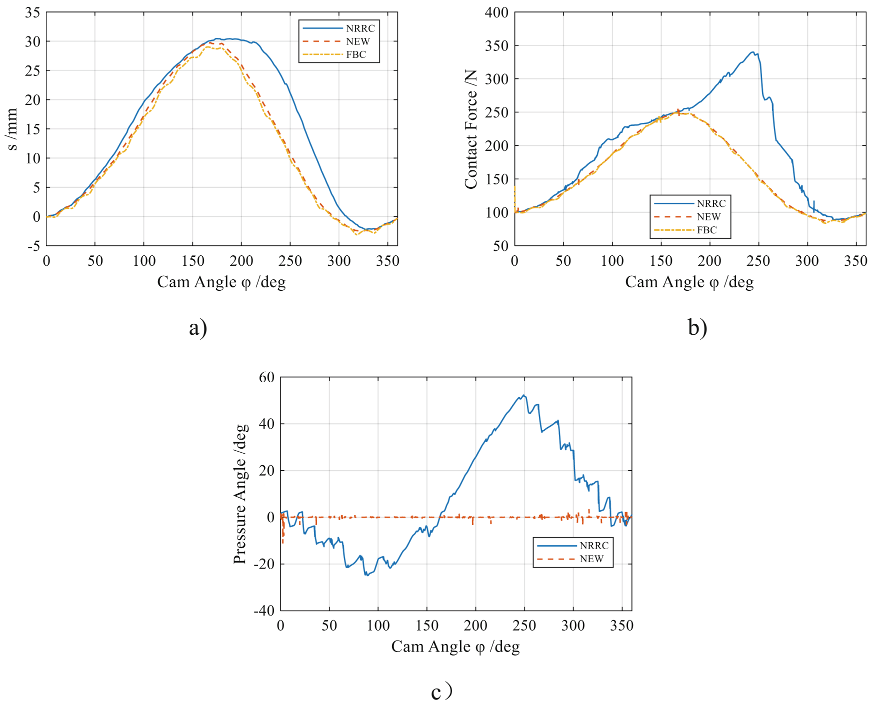


Fig. 6. Comparison of ADAMS simulation results. a) The motion of the follower under the same cam input angle; b) The force at the contact point between the cam and the follower; c) The pressure angle. In the legend, NRRC represents the negative radius roller cam mechanism; NEW represents the new cam mechanism; FBC represents the flat-bottomed cam mechanism

5 Summary

The new cam mechanism proposed in this paper retains the advantages of the negative radius roller cam mechanism and has the same transmission characteristics as the flat-bottomed cam mechanism without pressure angle. The design of the profile of a new type of cam can be carried out using the profile method of a flat-bottomed cam, but the distortion of the convex profile of a negative radius roller needs to be met. This mechanism provides a good choice for the design of efficient and heavy-duty brake clamping mechanisms. In addition, because there is no relative sliding in the mechanism, the wear is smaller, which can replace some applications of flat-bottomed cams [7]. Finally, this paper applies the new cam mechanism to the clamping mechanism of EMB and designs a concept prototype as shown in **Fig.** 7.

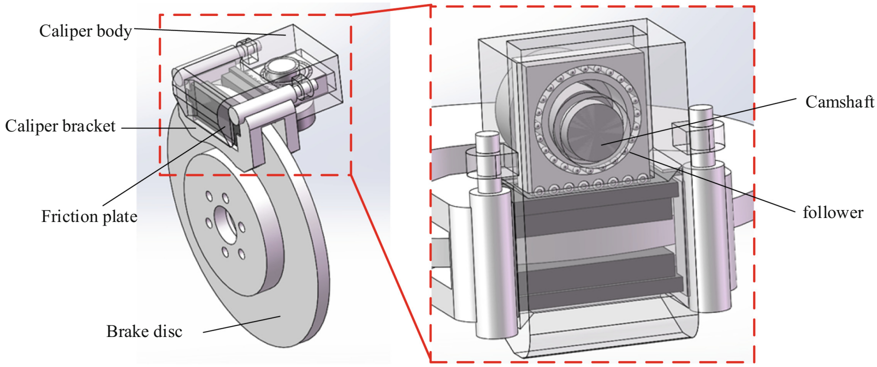


Fig. 7. Three-dimensional concept prototype

In the process of theoretical analysis, it was found that the mechanism has two degrees of freedom, and the motion of the mechanism is still determined in the case of only one actuator. This is due to the existence of a stable position with the lowest potential energy of load force when the concave roller is tangent to the cam, so the mechanism will automatically balance to the position with no pressure angle without the second motion input.

References

1. Schwarz, R., Isermann, R., Böhm, J., Nell, J., Rieth, P.: Clamping force estimation for a brake-by-wire actuator. Brake Technology and ABS/TCS Systems (1999)
2. Christof Maron, T.D.S.H.: In: Electromechanical Brake System: Actuator Control Development System
3. Kwon, S., Lee, S., Lee, J., Kum, D.: Accurate state estimation for electro-mechanical brake systems. J Electr Eng Technol **14**(2), 889–896 (2019). https://doi.org/10.1007/s42835-019-00124-x
4. Zongquan, D., Hongying, Y., Zhixing, W.: In: Mechanical Principles. 3rd edn. Higher Education Press (2015)

5. Carra, S., Garziera, R., Pellegrini, M.: Synthesis of cams with negative radius follower and evaluation of the pressure angle. Mech. Mach. Theory **39**(10), 1017–1032 (2004). https://doi.org/10.1016/j.mechmachtheory.2004.05.001
6. Xia, B., Liu, X., Shang, X., Ren, S.: Improving cam profile design optimization based on classical splines and dynamic model. J Cent South Univ **24**(8), 1817–1825 (2017). https://doi.org/10.1007/s11771-017-3590-x
7. Spiegelberg, C., Andersson, S.: Simulation of friction and wear in the contact between the valve bridge and rocker arm pad in a cam mechanism. Wear **261**(1), 58–67 (2006). https://doi.org/10.1016/j.wear.2005.09.022