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# Study on brake performance of new drum brake

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# Study on brake performance of new drum brake

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**Abstract.** This article studied an effective way to improve the braking performance of drum brake and proposed a kind of floating shoe drum brake. We built a model of the drum brake based on rigid-flexible coupling for the simulation and analysis of its' dynamic performance. Through theoretical analysis, the influence of floating direction on braking process is obtained. Then a finite element analysis according to the corresponding operating conditions is simulated, and we obtained the stress and strain distribution cloud, braking torque of the floating drum brake. Comparing braking performance with traditional drum brakes, the results show that floating drum brake will provide higher braking torque with the same corresponding operating conditions. In addition, the floating action of the brake shoe changes the contact condition and makes the contact pressure smaller.

## 1. Introduction

Drum brake is commonly used in medium and large passenger vehicle because of its' compact structure, reliable performance, high braking power and easy installation of parking devices [1-2]. When it is working, pneumatic device applies torque to the cam to drives the brake shoes close to the brake drum, braking torque generated between brake drum and shoes stops the vehicle. However, the traditional drum brake structure has obvious deficiencies, firstly, the friction stress in the friction plate isn't distribute uniformly. Secondly, brake clearance makes the friction plate cannot completely coincide with the brake drum in braking process, there is only partial contact happened in the friction plate. As a result, it worsening the contact conditions between the drum and friction plate and intensifies local wear and shortens the service life of the friction plate.

This paper comprehensively considers the effects of various factors on brake performance, proposing a mathematical analysis and using Matlab simulate how the contact position change with the brake shoe's floating action. Finally, we establish the floating shoe drum brake in Solidworks and use ANSYS Workbench establish the rigid-flexible coupling dynamic model.

## 2. Analysis to contact area of the friction plate

### 2.1. Contact area

*2.1.1. Initial contact position.* The initial contact position is the point where the friction plate contacts the brake drum firstly. The friction plate's radius is slightly smaller than brake drum's radius, so the friction plate surface does not come into contact with the brake drum completely. The structure

diagram of drum brake is shown in Figure 1, we regard the brake components are rigid body so that we can ignore their deformation.
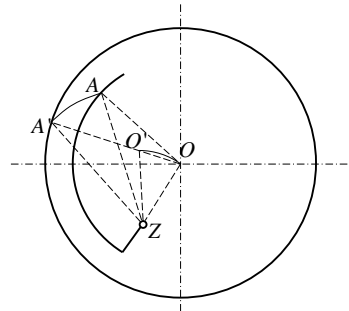


**Figure 1.** Structure diagram of drum brake.

O is the center of the drum, O is the center of the friction plate's outline surface. When it is begin to brake, only A' is contacted between friction plate and drum, and O, O', A' is collinear. Each size meets the following equation:

$$\angle AOZ = \angle A'O'Z \tag{1}$$

$$\angle A'O'Z = \pi - \angle ZO'O \tag{2}$$

$$\angle ZO'O = \angle ZOO' \tag{3}$$

$$\angle ZOO' = \arccos(|OO'|/|2OZ|) \tag{4}$$

We define $\angle AOZ$ is the initial contact position. Eq. 1-4 reflects the relationship between $|OO'|$ and the initial contact position. The relationship between them is shown in Figure 2 drawn by Matlab. As the braking clearance decreases, the initial contact point is located close to the position where makes $\angle AOZ = 90°$, and the point is always located in the upper part of the friction plate.
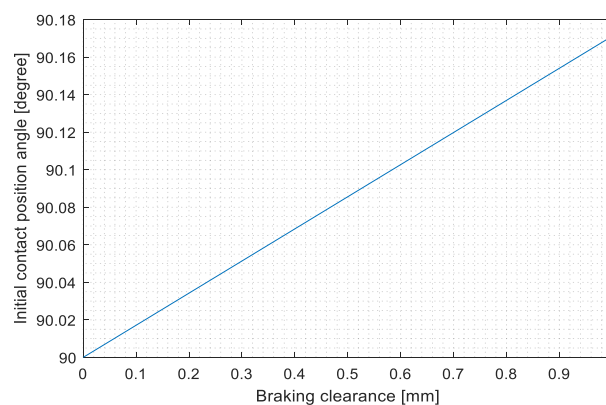


**Figure 2.** Relationship between braking clearance and initial contact position angle.

*2.1.2. Relationship between the initial contact position and contact area.* After the contact occured, the area near the initial contact point of the friction plate will deform and comes into contact with the brake drum. So the initial contact position could reflect the contact area's location, we can research contact area through studying on the initial contact position.

*2.2. Effect of floating action on initial contact position*
The floating shoe drum brake's is shown in Figure 3, there is a floating device in the support pin. The device allows the shoe to float in a certain direction, thus changing the initial contact position. When the floating direction is on the left side of the supporting pin, the structure diagram of the leading shoe

part of the floating brake is shown in Figure 4, $O'$ is the center of the friction plate. We know that when the contact started, A, O', O is collinear, $\angle DZC$ is the floating angle, |CZ| is the floating distance, C is the position of the support pin after floating, $\angle AO'C$ is the initial contact position.
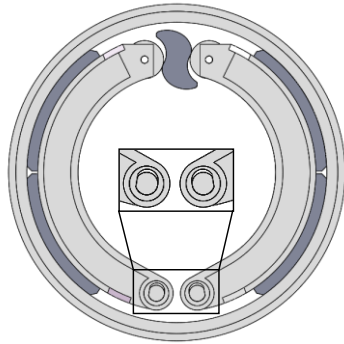
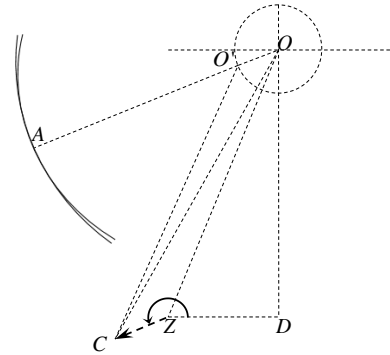

**Figure 3.** The float brake.         **Figure 4.** Structure diagram of the leading shoe.

Obviously $\angle AO'C$ is determined by the floating distance and the floating direction. The equations of motion of the model are as follows:

$$\angle CZO = \angle DZC - (\pi/2 - \angle ZOD) \tag{5}$$

$$CO^2 = CZ^2 + ZO^2 - 2 \times CZ \times ZO \times \cos\angle CZO \tag{6}$$

$$OO' = AO - AO' \tag{7}$$

$$\cos\angle O'OC = (OO'^2 + CO^2 - OC^2)/(2 \times OO' \times CO) \tag{8}$$

$$\cos\angle O'CO = (CO'^2 + CO^2 - OO'^2)/(2 \times CO' \times CO) \tag{9}$$

$$\angle AO'C = \angle O'OC + \angle O'CO \tag{10}$$

Eq. 5-10 reflects the variation of $\angle AO'C$ with the floating distance and the floating direction. The Figure 5 drawn by Matlab shows that when the float direction is within the range of [170.4°, 350.4°], $\angle AO'C$ can be changed smaller obviously, it can improve the contact condition. In addition, simulations from Matlab reflect that when $\angle DZC = 260.4°$, the initial contact position is most sensitive to the change of the floating distance.
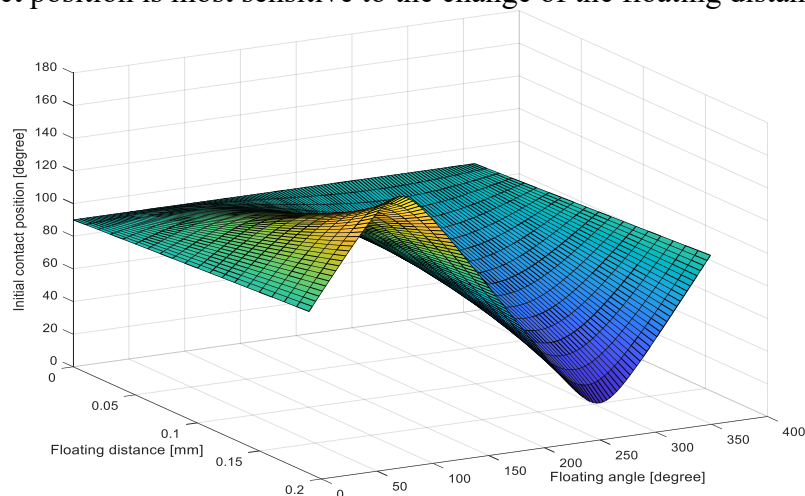


**Figure 5.** Change of initial contact position.

### 2.3. Optimum of the floating angle

*2.3.1. Reaction force to the friction plate.* The brake shoe's rotation is very small in the braking process, so we regard it local in the initial position approximately. When the friction plate contact with drum completely, the location relationship of the friction plate and the brake drum is as follows Figure 6:
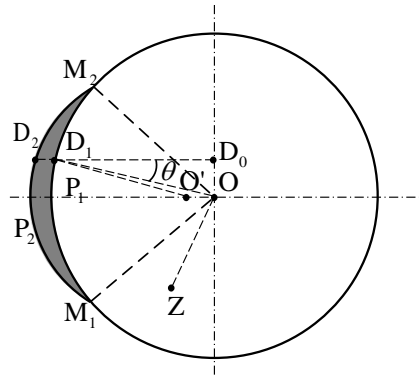


**Figure 6.** Location and deformation of the friction plate.

$\angle M_1OM_2 = 2\angle M_1OP_1 = \beta$, The grey area is the deformed area of the friction plate and it is on the symmetry of the horizon, we suppose that all the deformation follows the Hooke Law, and every point on the friction deforms along the O'O direction. As the young's modulus is much smaller than other component, so we only regard the friction plate as a flexible body. $M_1$, $M_2$ are two boundary point of the friction plate. Friction coefficient $\mu = 0.4$. So direction of the reaction force $\omega$:

$$l = \left| D_1 D_2 \right| = \sqrt{R^2 - r^2 \times \sin^2 \frac{\beta}{2}} \tag{11}$$

$$F_x = kbR \int_{-\frac{\beta}{2}}^{\frac{\beta}{2}} l \cos^3 \theta d\theta \tag{12}$$

$$F_y = \mu kbR \int_{-\frac{\beta}{2}}^{\frac{\beta}{2}} l \cos^3 \theta d\theta \tag{13}$$

$$\tan \omega = \frac{F_y}{F_x} = \frac{\mu kbR \int_{-\frac{\beta}{2}}^{\frac{\beta}{2}} l \cos^3 \theta d\theta}{kbR \int_{-\frac{\beta}{2}}^{\frac{\beta}{2}} l \cos^3 \theta d\theta} = \mu \tag{14}$$

$$M_0 = \mu kbR^2 \int_{-\frac{\beta}{2}}^{\frac{\beta}{2}} l \cos^2 \theta d\theta \tag{15}$$

$$L_0 = \frac{M_0}{F_y} = \frac{\mu kbR^2 \int_{-\frac{\beta}{2}}^{\frac{\beta}{2}} l \cos^2 \theta d\theta}{\mu kbR \int_{-\frac{\beta}{2}}^{\frac{\beta}{2}} l \cos^3 \theta d\theta} = 212.5437 \text{mm} \tag{16}$$

Where, $F$——reaction force from the drum, $F_x$——the horizontal component of $F$, $F_y$——the vertical component of $F$, $M_0$——the moment reaction from drum to the brake, $L_0$——arm of force of $F_y$, $F_1$——reaction force from the cam.

The Figure 7 is the forced diagram. Where, $\delta_1 = 17.16°$. The reaction forces from the cam, support pin and brake drum make the brake shoe in equilibrium condition of forces, so we can gain equation:

$$F_1 \times \cos\delta_1 \times |CD| + F \times \sin\omega \times L_0 = F_1 \times \sin\delta_1 \times (|CG| - |ZD|) \tag{17}$$

$$F \times \cos\omega - F_1 \times \cos\delta_1 = F_2 \times \cos\delta_2 \tag{18}$$

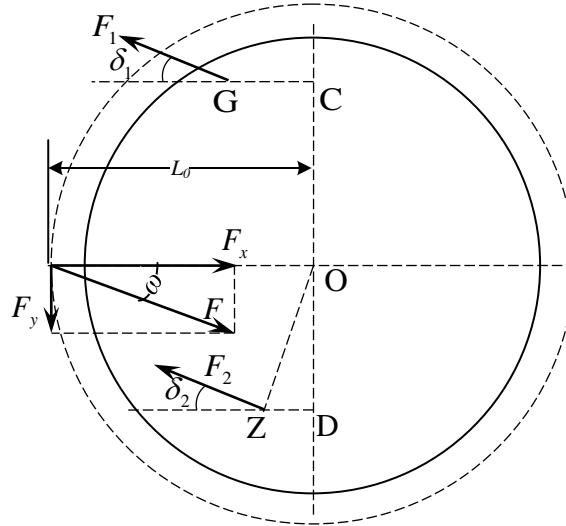$$F \times \sin\omega - F_1 \times \sin\delta_1 = F_2 \times \sin\delta_2 \tag{19}$$



**Figure 7.** The forced diagram of the friction plate.

Solving Eq.11-19 through Matlab, we can gain the calculation that $\delta_2 = 23.37°$, the result is independent with the value of $F$ and $F_1$. The reasonable floating angle should makes $F_2$ along the direction, so the normal direction of the pin's support surface must along this direction. The floating angle is $270° - 23.37° = 246.63°$.

## 3. Finite element simulation

### 3.1. Finite element model establishment
Finite element technology can indicate conditions at friction interface of drum brakes [3]. In order to facilitate simulation analysis, we simplified the brake model. The analysis method based the rigid-flexible coupling model could solve dynamic problem effectively [4]. Traditional analyze method regard brake shoe and brake drum as rigid body [5], to gain precise result, we set brake drum as flexible body. The key dimensions of the brake are as follows: Braking drum radius $R = 205$mm, friction plate radius $r = 204.8$mm, the area of the air brake chamber is $S = 30$in$^2$, the pressure of brake tube $p = 0.5$MPa, and length of the adjusting arm $L = 165$ mm. So the input torque on the cam is $M$:

$$M = p \times S \times L = 1596.8 \text{N·m} \tag{20}$$

The radius of the installed tire on the brake $R_0 = 660$ mm, brake drum rotation velocity corresponding to 30km/h travel speed is $\omega$:

$$\omega = \frac{v}{R_0} = 12.63 \text{rad/s} = 722.72°/\text{s} \tag{21}$$

The process of the finite element simulation could be divided into two load steps [6]. In the first load step, a torque of 1596.8 N is applied to the camshaft. In the second load step, a 21.66° rotation is applied to the brake drum. To prove whether the angle is the optimum, we set eight floating angle in the simulation.

### 3.2. Results of finite element model analysis

In order to reduce the computation, only half of the models are analyzed by ANSYS. The stress distribution are shown in Figure 8 and Figure 9. Table 1 shows the braking torque of the two types of brakes. The pressure distribution on the friction plate surface of the floating brake's leading shoe is more uniform and distributed in the brake shoe's middle area. In addition, the floating brake's maximum stress of it is 2.5MPa lower than ordinary brake. Stress distributions in the trailing shoe's friction plates of two types brake are not significant. But the maximum stress is also 2.1MPa lower than ordinary brake' trailing shoe.
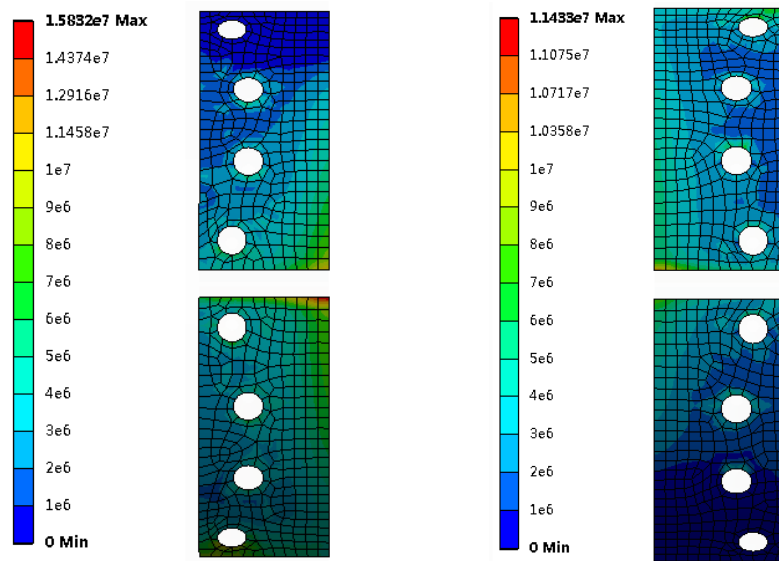


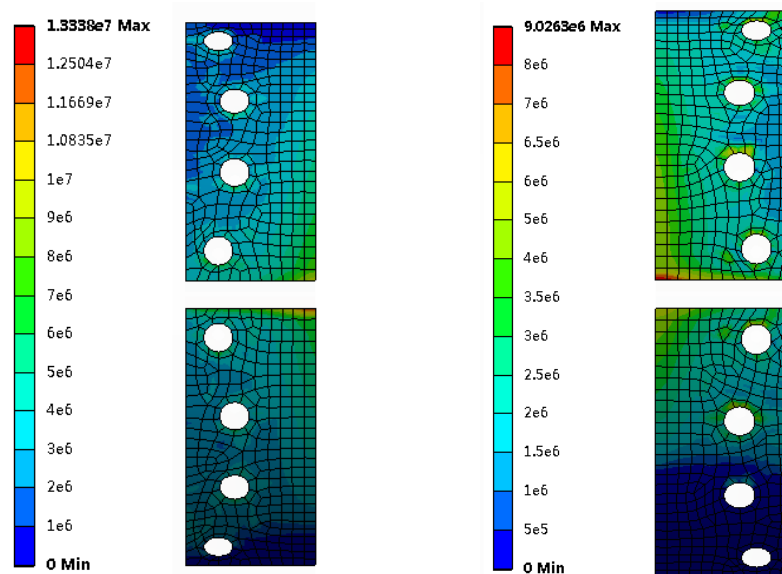**Figure 8.** Friction plates' stress distribution cloud of ordinary brakes.



**Figure 9.** Friction plates' stress distribution cloud of floating brakes.

**Table 1.** Brake torque of the two brakes.

|  | Friction Moment of Leading Shoe (N·m) | Friction Moment of Trailing Shoe (N·m) | Total Friction Moment (N·m) |
|---|---|---|---|
| Ordinary brake | 7472.51 | 4785.90 | 12258.41 |
| Floating brake | 8292.98 | 4874.80 | 13167.78 |

## 4. Summary

The finite element analysis shows that the floating shoe drum brake own better braking performance compared with traditional drum brake. Firstly, the mathematical analysis proved that in the former's leading shoe, contact area is tends to move to the middle position of the friction plate, thus the stress distribute more even and its' maximum is smaller. Secondly, in the floating shoe drum brake, the total braking moment is larger so that it could provide efficient braking performance. Thirdly, it is prove that there is an optimal floating angle, and the friction torque is maximum with the angle.

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