Distribution of Static Deformations along the Friction Lining of the Brake Mechanism in the Cars of GAZelle Series

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A R T I C L E   I N F O

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A B S T R A C T

The study of the shoe lining in the drum brake mechanism is of paramount importance to evaluate its wear regularity. Thereto, the computational scheme of the drum brake-shoe mechanism has been designed, the calculation formulae of tangential and radial components of the external points displacement in the lining have been analyzed, by means of which their numerical values have been determined, the diagram of the radial displacements has been designed and compared with descriptors of the wear change along the friction lining, according to which they alter with similar regularities.

Introduction

To determine the pressure distribution pattern along the shoe lining and to evaluate its effect on the wear of the brake mechanism elements, it is necessary to study the stress-strain state of the shoe lining. As a rule, the brake shoes have a high bending rigidity and relatively rigid supports, the deformation of which can be ignored. It is evident that the main deformable element of the drum brake-shoe mechanism is the friction lining which has a thickness of 10 mm - 16 mm and an elasticity modulus smaller in several ranges as compared to that of the shoe substance. Let's examine the work of the brake mechanism to determine its stress-strain state (Vardanyan, et al., 2019).

Since the inner radius of the brake drum and the outer radius of the shoe are not equal, their contact throughout the whole friction plane is not simultaneous and a friction

pair contact with small area appears at the start of braking, which grows up along with the increase of the shoe pressing P power encompassing the full friction area of the lining and drum. To compress the drum, the shoe lining completely rotates round the O_1 point and all external points of the lining rotate round the radius ρ_i (Figure 1) (Bazikyan, Djinyan, 2005).

Materials and methods

Let's analyze the triangle O_1AO introduced in figure 1 to determine the external points A_i of the lining, where O_1O is the radius R_l of the lining circle, AO_i is the radius of the drum R_d respectively, R_d - $(R_l + h) = \Delta$ is the clearance between the drum and lining, and h is the thickness of lining.
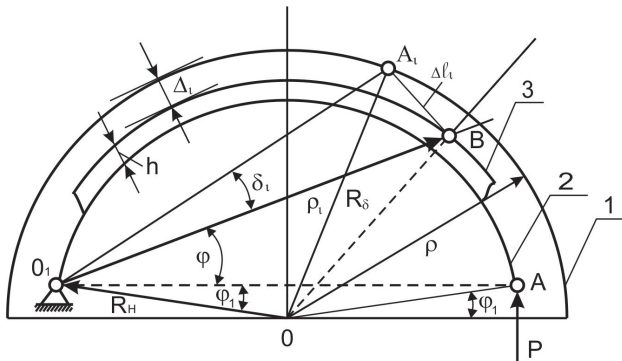
Figure 1. Drum brake-shoe mechanism. *1- drum, 2- shoe, 3- lining* (*composed by the authors*).

Let's determine the current value of the ρ_i radius vector depending on the rotation angle φ_i of the A_i point through the O_1A_i0 isosceles triangle considering that $\sphericalangle\varphi_i = \sphericalangle 0_1AB$ is a variable value for different A_i:

$$\rho_i = 2R_i \cdot \cos(\varphi + \varphi_i), \qquad (1)$$

it is accepted that $O_1O \approx OB = R_l$,

we'll get from triangle O_1A_iO:

$$R_d = R_d^2 + \rho_i^2 - 2R_i \cdot \rho_i^2 \cdot \cos(\varphi + \varphi_i + \delta_i). \qquad (2)$$

Placing the equation (1) in the (2) one and making some modification we'll have the following:

$$\frac{R_d^2 - R_l}{4R_l^2} = \cos^2\beta(1 - \cos\delta_i) + \frac{1}{2}\cos 2\delta_i \cdot \sin\delta_i, \quad (3)$$

where $\beta = \varphi + \varphi_i$.

From the derived (3) equation the rotation angle δ_i of the radius vector for A_i point in the brake mechanism can be determined.

The clearance Δ between the brake lining and drum is expressed through a rather small value due to which the rotation angle δ_i of the radius vector will be small as well, thus it can be accepted that $sin\ \delta_i = tg\ \delta_i = \delta_i$ and after some modifications the (3) equation will look like the following:

$$\delta_i^2\cos^2\beta + \frac{1}{2}\delta_i \cdot \sin^2\beta - a_i = 0,$$

where. $a_i = \dfrac{R_d^2 - R_l}{4R_l^2}.$

The solution of the derived equation will be as follows:

$$\delta_{i1.2} = \frac{\sin\beta \pm \sqrt{\sin^2\beta + a_i}}{\cos\beta}. \qquad (4)$$

From the derived solution it follows that the rotation angle δ_i of the ρ_i radius vector will be at minimum if the first derivative of the (4) expression is equal to 0:

$$\sin\beta = \frac{1}{\sqrt{2 + a_i}} \quad \text{or} \quad \sin(\varphi + \varphi_i) = \frac{1}{\sqrt{2 + a_i}}. \qquad (5)$$

Therefore, the minimum rotation angle of radius vector will be:

$$\delta_i = \frac{a_i}{\sqrt{a_i + 1}}. \qquad (6)$$

Let's determine the value of a_i depending on the clearance between the drum and lining, taking into account that $\Delta_i = R_d - R_l$, since while rotating round the point the O_1 distance of the A_i points in the lining from the R_l grows up with the size of Δ_i. For that purpose after some modification of the expression we'll have:

$$a_i = \frac{R_d^2 - R_l^2}{4R_l^2},$$

$$a_i \cong \frac{\Delta_i}{2R_l}.$$

To contact the drum, the displacement value of A_i any point in the lining will be:

$$l_i = \rho_i \cdot \delta_i. \qquad (7)$$

The first point to press the drum will be the one, the radius vector ρ_i of which matches the rotation angle φ:

$$\varphi = arc\sin\frac{1}{\sqrt{2 + a_i}} - \varphi_1.$$

Therefore, all external points of the lining will be displaced with the size of Δl_i before pressing the drum:

$$\Delta l_i = \rho_i \cdot \delta_{i(\varphi=0)} - \rho_i \cdot \delta_i, \qquad (8)$$

where Δl_i is the displacement of A_i point in the lining in case of rotation angle $\varphi = \dfrac{\pi}{2} - \dfrac{\delta_i}{2}$ of ρ_i radius vector.

Let's divide the Δl_i displacement into tangential V_i and radial W_i components and we'll have:

$$\begin{cases} V_i = \Delta l_i \cos\left(\varphi_i + \varphi - \dfrac{\delta_i}{2}\right) \\ W_i = \Delta l_i \sin\left(\varphi_i + \varphi - \dfrac{\delta_i}{2}\right) \end{cases}. \qquad (9)$$

Results and discussions

Taking into account that the stress-strain state of the lining, the regularity of the pressure distribution along the lining, as well as the wear of the brake mechanism parts are characterized through the components derived in the system of lining displacement (9), let's determine their numerical values and design the descriptor $W_i = f(\varphi_i)$ (Figure 2).

The numbered values identify a_i for the following changing ranges: $a_i = 0.1\text{-}0.28$, according to which $\delta_{imin} = 0.02$ rad. The rotation angle φ_i of the radius vector is determined through the (5) formula:

$$\varphi_i = arc \quad \sin \frac{1}{\sqrt{2 + a_i}} - \varphi,$$

where $\varphi = 10^0$.

The computation results are introduced in Table.

The derived descriptor (Figure 2) testifies that the radial displacements of the external points in the lining grow up along with the increase of rotation angle of the points and it amounts to the maximum in the frontal part of the lining. Such characteristic will cause to the same regularity in the pressure distribution along the lining and to the similar descriptors of the lining wear (Figure 3), (Vardanyan, 2020).

Table. The computation results of the displacement components*

| a_i	φ_i, radius	φ_i, degree	ρ_i mm	δ_i, radius	l_i mm	Δl_i mm	W_i mm
1.7716	0.27911	16.0	240.02717	1.06414	255.423	15.1723	0.134122
1.56278	0.29656	17.0	237.47569	0.97621	231.825	14.4	1.010796
1.37283	0.314	18.0	234.85195	0.89122	209.304	13.6296	1.769448
1.19958	0.33144	19.0	232.15674	0.80883	187.776	12.8533	2.412668
1.04114	0.34889	20.0	229.39089	0.72874	167.166	12.0624	2.941236
0.8959	0.36633	21.0	226.55523	0.65066	147.41	11.2474	3.353951
0.76247	0.38378	22.0	223.65064	0.57433	128.449	10.3977	3.647379
0.63962	0.40122	23.0	220.67798	0.49952	110.233	9.50074	3.815496
0.52629	0.41867	24.0	217.63818	0.426	92.7135	8.54179	3.84916

*Composed by the authors.
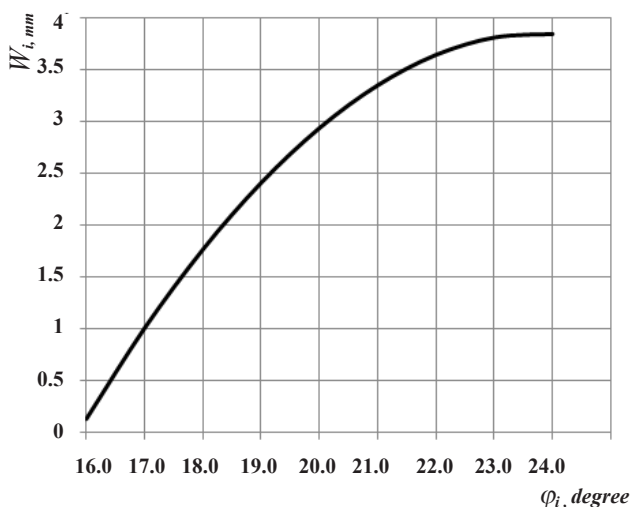
Figure 2. Radial displacement diagram of the external points in the friction lining (*composed by the authors*).
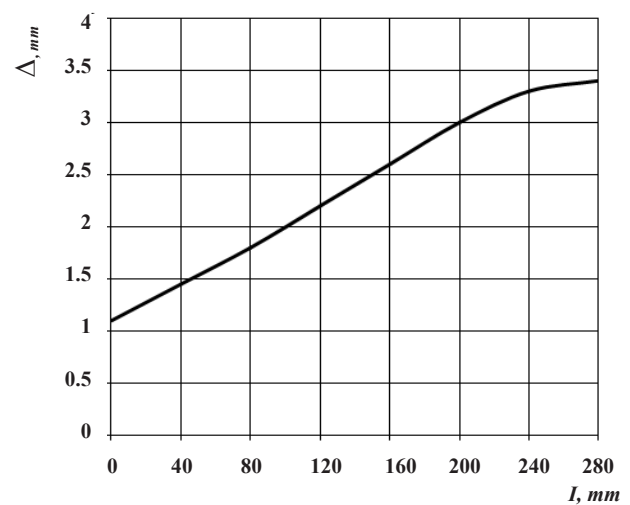

Figure 3. Description of wear change along the friction lining (*composed by the authors*).

Conclusion

Per the descriptor $W_i = f(\varphi_i)$ of the radial displacement changes in the external A_i points of the lining in the brake-shoe mechanism a conclusion can be drawn that along with the increase of the shoe rotation angle φ_i the radial displacements grow up with the sine theorem accepting the minimum value at the shoe base equal to 0.13 mm and the maximum one at the frontal part of the shoe amounting to 3.84 mm. Such characteristics will simultaneously bring forward similar descriptors in the wear changes and pressure distribution along the lining.

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