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Output Parameters of a Drum Brake with Floating Shoes

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Abstract—A method is developed for calculating the output characteristics of a drum brake with floating shoes on the basis of the system configuration, the dimensions of the components, the frictional coefficient in contact zones of the linings, and the driving forces on the shoes. By this means, the structural parameters of the brake, their inference on the braking torque, the effectiveness of the shoes, the reaction forces in the shoe supports, and the maximum pressure on the frictional linings may be determined at the design stage. As an illustration, brake mechanisms with a single shoe and with servo amplification are calculated.

Keywords: braking system, drum brake, floating shoe, braking efficiency

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Drum brakes with floating shoes are used in cars and light trucks. The floating shoes may slip over the inclined support or interact with one another through an intermediate component. That ensures self-stabilization of the shoes at the drum's inner surface, eliminating misalignment of the concentric frictional surfaces of the drum and shoes. Consequently, shoe wear is uniform. Such brakes do not require high manufacturing precision. In addition, in comparison with fixed shoes, they are rapidly prepared for operation [1, 2].

Classical methods of determining the output characteristics of a drum brake with floating shoes are based on the following assumptions: the drum and shoe are absolutely rigid; and the pressure on the frictional lining is proportional to its radial deformation [3–8].

Simple methods of calculating the braking torque by graphical analysis were proposed in [3–5]. However, the series of calculations of the braking torque with different structural parameters is time-consuming.

An analytical method of determining the braking torque based on trial and error solution of the equilibrium equations for the shoes was proposed in [6]. This approach is unsuitable for multivariant calculations.

Formulas for the braking efficiency of the shoes were derived in [7]. They were used to study the influence of the brake's structural parameters on the braking efficiency with parallel working planes of the shoe supports.

Analytical expressions for the braking efficiency of floating and fixed shoes when using different clamping systems were presented in [8].

The maximum contact pressures at the shoes' frictional surfaces and the reaction forces in the shoe supports cannot be determined by the methods in [7, 8]. However, this information is required to select the width of the frictional linings and to calculate the strength of the shoe supports.

Analysis of classical methods of brake calculation shows that they all have deficiencies and limitations. Despite the successful use of finite-element analysis for brake mechanisms, classical analytical methods remain effective for design purposes, since they determine the braking torque with great accuracy [9, 10]; and they do not require impractical expenditures of computational resources or time in determining the structural parameters. Therefore, it is expedient to minimize the deficiencies and expand the capabilities of analytical methods for drum brakes.

In the present work, we describe a method of finding analytical expressions for the braking torque of a drum brake with floating shoes; the braking efficiency of the shoes; the performance of the brake mechanism; the reaction forces in the shoe supports; and the maximum pressure on the frictional linings. The method is based on a simplified brake configuration, the brake's dimensions, and the forces created by the clamp.

BASIC CONFIGURATION OF FLOATING SHOES

The floating shoe in the drum brake has two degrees of freedom: it may rotate and slide along the supports. We may distinguish between driving and driven modes of the shoe: in the driving mode, it rotates around the support in the same direction as the drum; it is driven if it rotates in the opposite direction.

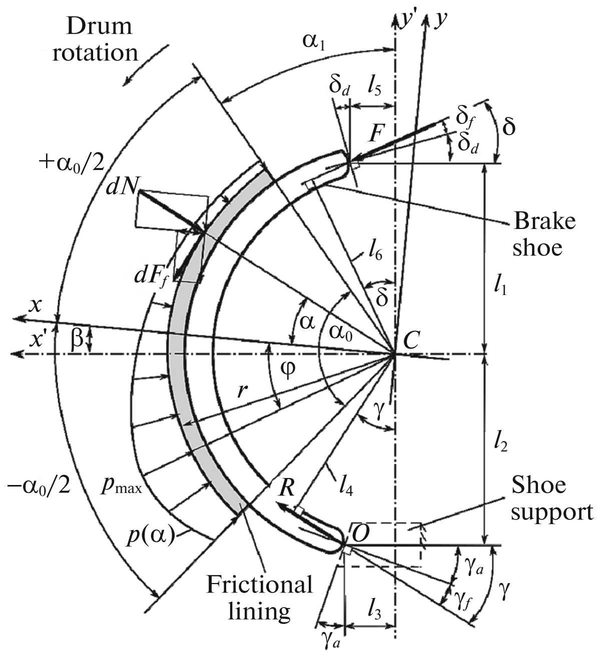

Fig. 1. Basic structure of the driving shoe in a drum brake.

In Fig. 1, we show the shoe in driving mode when the brake is turned on. The clamp exerts a force F on the upper ring of the shoe, pressing the frictional lining against the rotating drum. Distributed radial load p acts on the lining in that case; normal force dN and frictional force dF_f appear at an elementary area of the lining. At the lower ring of the shoe, the support exerts reaction force R.

In driven mode, the configuration is analogous except that the elementary frictional force dF_f is in the opposite direction.

In the analysis, we make the following assumptions: the drum and shoe are absolutely rigid; the lining is elastic; the lining's radius of curvature is equal to the drum radius; there is no slipping between the drum and lining; the frictional coefficient of the lining is constant; the pressure on any elementary area is the same over the width of the lining; and the pressure distribution over the arc of the lining is described by a cosine function [2, 3, 6]

$$p(\alpha) = p_{\max}\cos(\varphi - \alpha), \quad (1)$$

where p_{max} is the maximum pressure at the lining; φ is the angle corresponding to the maximum pressure (between the x axis and the axis of maximum pressure at the lining); and α is the angular coordinate of the frictional lining measured from the x axis.

We consider two Cartesian coordinate systems. The system $x'y'$ is used to determine the position of the brake components; its origin is at the axis of drum rotation; the y' axis is the symmetry axis of the brake mechanism. To obtain the equilibrium equations of the shoe, we use the coordinate system xy, which is turned by an angle β relative to the $x'y'$ system so that the x axis is aligned with the bisector of the angle α_0 corresponding to the extent of the lining. Because the x axis is symmetric relative to the angle α_0, the integrals of the trigonometric functions may more easily be calculated, and the resulting analytical formulas are of simpler structure.

The angle β characterizes the asymmetric position of the lining relative to the x' axis. When $\beta = 0$, the lining is symmetric with respect to the x' axis. We assume that β is positive when the bisectrix of the lining arc rotates in the opposite direction to the shoe around the support; and negative when these rotations are in the saem direction. The angle α_1 between the beginning of the frictional lining and the y' axis determines the position of the lining with respect to the brake shoe: $\alpha_1 = \pi/2 - \beta - \alpha_0/2$.

The notation adopted is as follows (Fig. 1): r is the radius of the frictional surfaces of the drum and frictional lining; l_1 and l_5 are, respectively, the coordinates of the points of shoe–clamp contact; l_2 and l_3 are, respectively, the coordinates of the points of shoe–support contact; l_4 and l_6 are the distances at which the reaction R of the support and the driving force F act relative to the axis of drum rotation; γ_a is the inclination of the working plane of the shoe support; γ_f is the frictional angle at the point of shoe–support contact; δ_d is the inclination of the clamp's supporting plate; δ_f is the frictional angle at the contact point between the shoe and the clamp's supporting plate; and δ and γ are the angles between the x' axis and the vectors F and R.

The angles δ and γ are determined from the formulas

$$\delta = \delta_d + \delta_f = \delta_d + \arctan\mu_d;$$

$$\gamma = \gamma_a + \gamma_f = \gamma_a + \arctan\mu_a,$$

where μ_d and μ_a are the frictional coefficients of the shoe with the clamp and support, respectively.

It follows from Fig. 1 that

$$l_4 = (l_2 + l_3\tan\gamma)\cos\gamma;$$

and $$l_6 = (l_1 + l_5\tan\delta)\cos\delta.$$

This approach permits determination of the output parameters of most drum brakes with floating shoes when using different clamps and different configurations of the shoe support's working planes: with inclination $\gamma_a > 0$ or $\gamma_a < 0$, in a vertical position ($\gamma_a = 0$), or for brake mechanisms in which the lower ends of the shoes interact through an intermediate element.

EQUILIBRIUM EQUATIONS OF BRAKE SHOES

The necessary and sufficient condition for equilibrium of the shoe is that the sum of the projections of all the forces onto the x and y axes and the sum of their algebraic torques relative to the drum axis (point C) are zero.

The system of equilibrium equations of the drum shoes in driving and driven modes is as follows

$$\begin{cases} \sum F_x = 0, \\ F\cos(\delta+\beta) + R\cos(\gamma-\beta) - \int\limits_{-\alpha_0/2}^{+\alpha_0/2} dN\cos\alpha \\ \pm \int\limits_{-\alpha_0/2}^{+\alpha_0/2} dF_f \sin\alpha = 0; \\ \sum F_y = 0, \\ -F\sin(\delta+\beta) + R\sin(\gamma-\beta) - \int\limits_{-\alpha_0/2}^{+\alpha_0/2} dN\cos\alpha \\ \pm \int\limits_{-\alpha_0/2}^{+\alpha_0/2} dF_f \sin\alpha = 0; \\ \sum M_C = 0, \quad Fl_6 - Rl_4 \pm r\int\limits_{-\alpha_0/2}^{+\alpha_0/2} dF_f = 0. \end{cases} \tag{2}$$

In Eq. (2) and the following formulas, the upper sign in the operations ± and ∓ correspond to driving mode and the lower sign to driven mode.

To simplify the solution of Eq. (2), we write Eq. (1) in the form

$$p(\alpha) = p_c\cos\alpha + p_s\sin\alpha,$$

where p_c and p_s are parameters determined from the equations

$$p_c = p_{\max}\cos\varphi, \quad p_s = p_{\max}\sin\varphi. \tag{3}$$

The elementary forces associated with the pressure and friction on an infinitesimal area of the lining are determined by the formulas

$$dN = wrp(\alpha)d\alpha = wr(p_c\cos\alpha + p_s\sin\alpha)d\alpha;$$

$$dF_f = \mu dN = \mu wr(p_c\cos\alpha + p_s\sin\alpha)d\alpha,$$

where w is the width of the frictional lining; and μ is the frictional coefficient of the lining.

The integrals in Eq. (2) take the form

$$\int\limits_{-\alpha_2/2}^{+\alpha_2/2} dN\cos\alpha = wr(p_cI_{cc} + p_sI_{sc}); \tag{4}$$

$$\int\limits_{-\alpha_2/2}^{+\alpha_2/2} dN\sin\alpha = wr(p_cI_{sc} + p_sI_{ss}); \tag{5}$$

$$\int\limits_{-\alpha_2/2}^{+\alpha_2/2} dF_f\sin\alpha = \mu wr(p_cI_{sc} + p_sI_{ss}); \tag{6}$$

$$\int\limits_{-\alpha_2/2}^{+\alpha_2/2} dF_f\cos\alpha = \mu wr(p_cI_{cc} + p_sI_{sc}); \tag{7}$$

$$\int\limits_{-\alpha_2/2}^{+\alpha_2/2} dF_f = \mu wr(p_cI_c + p_sI_s), \tag{8}$$

where I_{ss}, I_{sc}, I_{cc}, I_s, and I_c are tabular values of the integrals of trigonometric functions

$$I_{ss} = \int\limits_{-\alpha_0/2}^{+\alpha_0/2} \sin^2\alpha d\alpha = (\alpha_0 - \sin\alpha_0)/2;$$

$$I_{sc} = \int\limits_{-\alpha_0/2}^{+\alpha_0/2} \sin\alpha\cos\alpha d\alpha = 0;$$

$$I_{cc} = \int\limits_{-\alpha_0/2}^{+\alpha_0/2} \cos^2\alpha d\alpha = (\alpha_0 + \sin\alpha_0)/2;$$

$$I_s = \int\limits_{-\alpha_0/2}^{+\alpha_0/2} \sin\alpha d\alpha = 0;$$

$$I_c = \int\limits_{\alpha_1}^{\alpha_2} \cos\alpha d\alpha = 2\sin(\alpha_0/2).$$

Substituting Eqs. (4)–(8) into Eq. (2) and taking into account that the integrals I_{sc} and I_s are zero, we find that

$$\begin{cases} \sum F_x = 0, \\ F\cos(\delta+\beta) + R\cos(\gamma-\beta) \\ - wrp_cI_{cc} \pm \mu wrp_sI_{ss} = 0; \\ \sum F_y = 0, \\ -F\sin(\delta+\beta) + R\sin(\gamma-\beta) \\ - wrp_sI_{ss} \mp \mu wrp_cI_{cc} = 0; \\ \sum M_C = 0; \quad Fl_6 - Rl_4 \pm \mu wr^2p_cI_c = 0. \end{cases} \tag{9}$$

This system of equations is linear with respect to R, p_c, and p_s. It is solved in symbolic form by means of the MuPAD computer algebra system. We obtain the distribution parameters of the pressures p_c and p_s and the reaction in the shoe supports R

$$p_c = F\frac{a_1 \mp \mu a_2}{2wr(b_0 \mp b_1\mu + b_2\mu^2)\sin(\alpha_0/2)}; \tag{10}$$

$$p_s = F\frac{\pm\mu[\sin(\delta+\gamma) - ha_1] - ha_2}{0.5wr(b_0 \mp b_1\mu + b_2\mu^2)(\alpha_0 - \sin\alpha_0)}; \tag{11}$$

$$R = F\frac{(hl_6/r)(1+\mu^2) \pm \mu\cos(\beta+\delta) - \mu^2\sin(\beta+\delta)}{b_0 \mp b_1\mu + b_2\mu^2}, \tag{12}$$

where

$$h = 0.25(\alpha_0 + \sin\alpha_0)/\sin(\alpha_0/2);$$

$$a_1 = (l_4/r)\cos(\beta + \delta) + (l_6/r)\cos(\beta - \gamma);$$

$$a_2 = (l_4/r)\sin(\beta + \delta) + (l_6/r)\sin(\beta - \gamma);$$

$$b_0 = hl_4/r; \quad b_1 = \cos(\beta - \gamma); \quad b_2 = b_0 + \sin(\beta - \gamma).$$

DETERMINING THE OUTPUT PARAMETERS OF THE DRUM BRAKE

The basic output characteristics of the drum brake are the braking torque, the braking efficiency of the shoes, the brake's margin of performance, the maximum pressure at the frictional linings, and the reaction forces in the shoe supports.

The braking torque on the drum due to shoes in driving mode (M_1) and driven mode (M_2) is equal to the sum of the elementary frictional torques along the arc of the frictional lining

$$M_{1,2} = r\int_{\alpha_1}^{\alpha_2} dF_f = \mu w r^2 p_c I_c. \quad (13)$$

Substituting Eq. (10) into Eq. (13), we may write the frictional torque of a single shoe in the form

$$M_{1,2} = F_{1,2}\frac{r(a_1\mu \mp a_2\mu^2)}{b_0 \mp b_1\mu + b_2\mu^2}. \quad (14)$$

The braking efficiency $C_{1,2}$ of the shoe is defined as the ratio of the equivalent frictional force of the shoe on the drum and the driving force of the shoe

$$C_{1,2} = F_{\text{eq}1,2}/F_{1,2}. \quad (15)$$

The equivalent frictional force $F_{\text{eq}1,2}$ is the force that creates a frictional torque corresponding to Eq. (14) when applied to the point of intersection of the x' axis and the frictional surface of the lining at a distance equal to the drum radius. In other words, it is the force such that $M_{1,2} = rF_{\text{eq}1,2}$. Hence, on the basis of Eqs. (14) and (15), we may write the braking efficiency in the form

$$C_{1,2} = \frac{a_1\mu \mp a_2\mu^2}{b_0 \mp b_1\mu + b_2\mu^2}. \quad (16)$$

Consequently, the braking torque developed by a single shoe is $M_{1,2} = rF_{1,2}C_{1,2}$.

The frictional torque developed by the drum brake is

$$M = r(F_1C_1 + F_2C_2). \quad (17)$$

The braking efficiency $C_{1,2}$ determines the efficiency with which the shoe's driving force is converted into a frictional force. In other words, it determines the frictional force on the drum that may be created by a shoe with a specified driving force. This characteristic may be used to compare brakes of different configurations.

It follows from Eq. (14) that the braking torque of a shoe in driving mode increases with decrease in the denominator and tends to infinity when $b_0 - b_1\mu + b_2\mu^2 = 0$. Then, with shoe–drum contact, the braking torque increases uncontrollably (even with constant driving force) until drum motion is blocked. After removing the force from the brake pedal, the shoe remains engaged with the drum; in other words, the brake is self-locking [1, 7]. That prevents smooth regulation of the braking torque by adjusting the pressure in the drive.

The frictional coefficient of the lining when the brake is self-locked is found from the formula

$$\mu_\infty = \frac{b_1 - \sqrt{b_1^2 - 4b_0b_2}}{2b_2}. \quad (18)$$

The value of μ_∞ depends only on the brake's design parameters.

The degree to which self-locking is prevented may be assessed by means of the brake's margin of performance

$$k_m = \mu_\infty/\mu. \quad (19)$$

If the brake is at the limit of self-locking, then $k_m = 1$.

To eliminate self-locking, we require that $k_m > 1.5$.

We now consider the load distribution over the arc of the lining.

From Eq. (3), the angle corresponding to maximum pressure is

$$\varphi = \arctan(p_s/p_c). \quad (20)$$

Then the maximum pressure may be determined by means of the formula

$$p_{\max} = p_c/\cos\varphi \quad \text{or} \quad p_{\max} = p_s/\sin\varphi. \quad (21)$$

The maximum pressure must not exceed the limiting permissible value for the chosen lining material.

Thus, the output parameters of the drum brake are as follows: the efficiency of the brake shoes; the frictional torque of the brake mechanism; the brake's margin of performance; the maximum pressure at the frictional lining; and the reaction of the shoe supports. The corresponding formulas are Eqs. (16), (17), (19), (21), and (12).

EXAMPLES

Example 1

We want to determine the output parameters of a drum brake with a single shoe in driving mode (Fig. 2a). The initial data are as follows: $r = 127$ mm; $l_1 = 100$ mm; $l_2 = 80$ mm; $l_3 = 20$ mm; $l_5 = 30$ mm; $\alpha_0 = 100°$; $w = 50$ mm; $\gamma_a = 0$; $\delta_d = 0$; $\mu = 0.4$; $\mu_d = 0.12$; $\mu_a = 0.12$; $F_{1,2} = 2500$ N; $\beta_1 = -5°$; and $\beta_2 = +5°$.

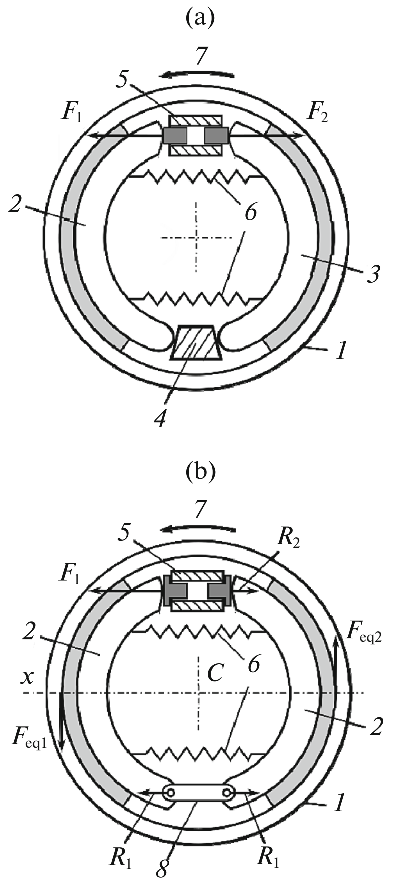

Fig. 2. Drum brake with floating shoes: (a) with one driving shoe; (b) with two driving shoes; (*1*) drum; (*2*) driving shoe; (*3*) driven shoe; (*4*) shoe support; (*5*) hydraulic cylinder; (*6*) tension spring; (*7*) direction of drum rotation; (*8*) floating element.

1. We calculate the braking efficiency of the shoes in driving and driven mode from Eq. (16): $C_1 = 2.476$, $C_2 = 0.559$. The shoe in driving mode creates a braking torque almost 4.5 times greater than the shoe in driven mode. That may be explained in that the torques of the elementary frictional forces at the driving shoe are in the same direction as the torque of the driven shoe. That presses the shoe more firmly against the drum. At the driven shoe, the torques of the elementary frictional forces are in the opposite direction to the torque of the driving force and weaken the force pressing the shoe to the drum.

2. Using Eq. (17), we find the braking torque of the mechanism: $M = 963.661$ N m.

3. From Eqs. (18) and (19), we determine the frictional coefficient of the lining $\mu_\infty = 0.845$ corresponding to self-locking of the brake; and the brake's margin of performance $k_m = 2.187$. This is higher than the recommended value.

4. From Eq. (20), we find the angle corresponding to maximum pressure at the linings of the driving and driven shoes: $\varphi_1 = -0.660$ rad; $\varphi_2 = 0.722$ rad.

5. We determine the maximum pressure at the frictional lining for the driving and driven shoes by means of Eq. (21): $p_{1\max} = 2.013 \times 10^6$ N/m^2; $p_{2\max} = 4.786 \times 10^5$ N/m^2.

6. From Eq. (12), we determine the reaction at the supports of the driving and driven shoes: $R_1 = 1.275 \times 10^4$ N; $R_2 = 973.128$ N.

These are the initial data in designing the attachment of the shoe supports to the frame of the mechanism and calculating the crumpling of the lower ends of the brake shoes.

Example 2

We want to determine the output parameters of a drum brake with servo amplification and two shoes in driving mode (Fig. 2b). The initial data for the first shoe are as follows (Fig. 2b, on the left): $r = 127$ mm; $l_1 = 100$ mm; $l_2 = 80$ mm; $l_3 = 20$ mm; $l_5 = 30$ mm; $\alpha_0 = 100°$; $w = 50$ mm; $\gamma_a = 0$; $\delta_d = 0$; $\mu = 0.4$; $\mu_d = 0.12$; $\mu_a = 0.12$; and $\beta = -5°$. The driving force of the first shoe is $F_1 = 2500$ N. The initial data for the second shoe are as follows (Fig. 2b, on the right): $l_1 = 80$ mm; $l_2 = 100$ mm; $l_3 = 26$ mm; and $l_5 = 20$ mm; the other parameters are the same as for the first shoe.

For this brake mechanism, the lower ends of the shoes are connected by a floating element. When the brake is switched on, the piston of a hydraulic cylinder is extended and presses the shoe against the rotating drum. The frictional force results in capture of the shoes by the drum. They rotate until the shoulder of the piston for the second shoe contacts the housing of the hydraulic cylinder, and the shoes reach equilibrium.

For both shoes, the driving force is the force F_1 created by the hydraulic cylinder. The support of the first shoe is a floating element; the support for the second shoe is the housing of the hydraulic cylinder. The following notation is employed here: R_1 and R_2 are the reactions of the shoe supports; F_{eq1} and F_{eq2} are the equivalent frictional forces of the shoes. The angle of rotation of the floating element when the brake is on is assumed to be negligibly small. Therefore, we regard the vector of reaction R_1 as parallel to the horizontal axis x. In this brake mechanism, the second shoe operates with servo amplification. Therefore, it is activated by reaction R_1 and frictional force F_{eq1} of the primary shoe. Since the reaction is much larger than the driving force created by the hydraulic cylinder, this brake is characterized by high braking efficiency.

We now determine the output parameters.

1. We calculate the braking efficiency of the first shoes Eq. (16): $C_1 = 2.476$.

2. We find the braking efficiency of the second shoe

$$C_2 = \frac{F_{eq\,2}}{F_1} = \frac{F_{eq\,2}}{R_1}\frac{R_1}{F_1} = C_1^* \frac{R_1}{F_1}. \quad (22)$$

We determine $C_2^* = F_{eq\,2}/R_1$ from Eq. (16), taking into account that the second shoe is in driving mode: $C_2^* = 1.468$.

To find a formula for the ratio R_1/F_1, we write an equation for the torques of all the forces on the first shoe relative to the drum's axis of rotation (point C)

$$\sum M_C = 0, \quad F_1 l_1 + rF_1 C_1 - R_1 l_2 = 0. \quad (23)$$

From Eq. (23)

$$\frac{R_1}{F_1} = \frac{l_1 + rC_1}{l_2}. \quad (24)$$

From Eqs. (22) and (24), we find that $C_2 = 5.706$.

3. We determine the braking torque from Eq. (17): $M = 1.025 \times 10^4$ N m.

4. From Eq. (10), we find the parameters of the pressure distribution over the lining arc for the first (p_{c1}) and second (p_{c2}) shoes, taking into account that the driving force is $F = F_1$ for the first shoe and $F = F_1(l_1 + rC_1)/l_2$ for the second shoe: $p_{c1} = 1.591 \times 10^6$ N/m^2; $p_{c2} = 3.666 \times 10^6$ N/m^2.

5. From Eq. (11), we find the parameters of the pressure distribution over the lining arc for the first (p_{s1}) and second (p_{s2}) shoes: $p_{s1} = -1.233 \times 10^6$ N/m^2; $p_{s2} = -3.221 \times 10^6$ N/m^2.

6. We calculate the angle corresponding to the maximum pressure at the linings of the driving and driven shoes from Eq. (20): $\varphi_1 = -0.660$ rad; $\varphi_2 = -0.721$ rad.

7. Using Eq. (21), we determine the maximum pressure at the frictional linings of the first and second shoes: $p_{1\max} = 2.013 \times 10^6$ N/m^2; $p_{2\max} = 4.880 \times 10^6$ N/m^2.

8. From Eq. (23), we obtain the support reaction for the first shoe: $R_1 = F_1(l_1 + rC_1)/l_2$. Then $R_1 = 1.295 \times 10^4$ N.

9. We write the equation for the torques of the forces on the second shoe relative to the drum axis

$$\sum M_C = 0, \quad R_1 l_1 - R_2 l_2 + rF_{eq\,2} = 0. \quad (25)$$

Writing an expression for F_{eq2} on the basis of Eq. (22), we substitute it into Eq. (25) to find the support reaction for the second shoe: $R_2 = \frac{R_1}{l_2}(l_1 + rC_2^*)$. Then $R_2 = 3.415 \times 10^4$ N.

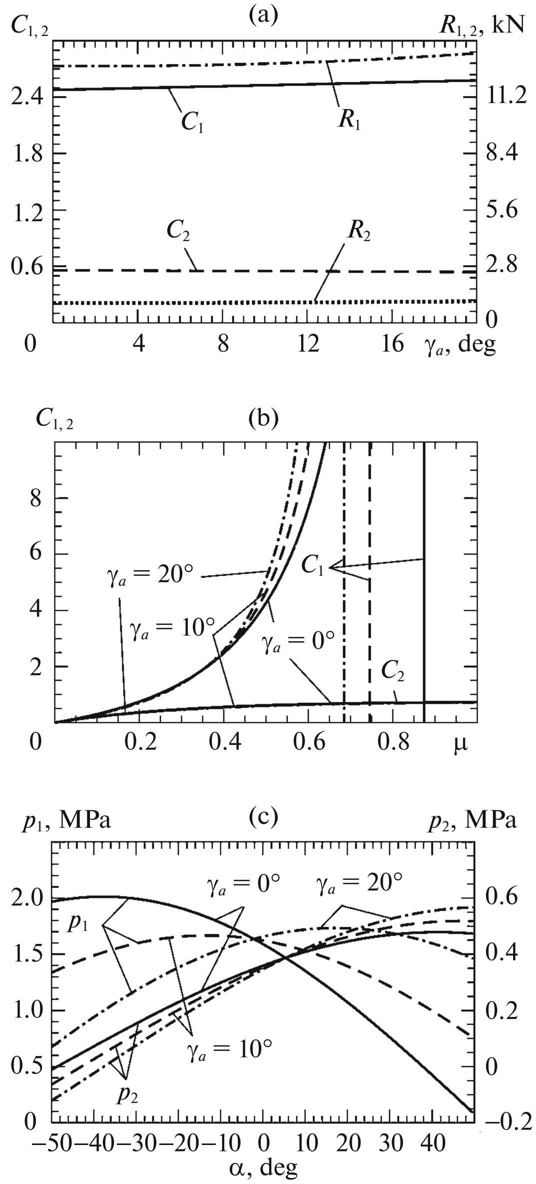

Fig. 3. Braking efficiency $C_{1,2}$ of the shoes and reaction $R_{1,2}$ of the shoe support as a function of the inclination γ_a of the shoe support (a); $C_{1,2}$ as a function of γ_a and the lining's frictional coefficient μ (b); and the pressures p_1 and p_2 along the arc of the lining for the driving and driven shoes as a function of α and γ_a (c).

Example 3

We now investigate the influence of the inclination γ_a of the shoe support on the output parameters of the brake mechanism in Fig. 2a, with variation in γ_a from 0° to 20° in 10° increments. The brake parameters are the same as in Example 1.

To that end, we write a Matlab program and plot Fig. 3. Analysis shows that, within the given range, variation in the inclination γ_a of the shoe support has no influence on the reactions $R_{1,2}$ of the shoe supports and the braking efficiency $C_{1,2}$ of the shoes and hence on the system's braking torque. The driving shoe is fundamental to the creation of the braking torque.

In Fig. 3b, we plot the braking efficiency as a function of the lining's frictional coefficient and the inclination of the shoe support. We see that, with increase in the frictional coefficient of the lining, the braking efficiency C_2 of the driven shoe gradually approaches a constant value, whereas the braking efficiency of the driving shoe increases; the function $C_1 = f(\mu)$ undergoes a discontinuity at the point $\mu = \mu_\infty$, as indicated by the vertical lines. With increase in γ_a, the brake's margin of performance declines. For example, when $\gamma_a = 0°$ and $\gamma_a = 20°$, the parameter μ_∞ is 0.87 and 0.68, respectively, while the brake's margin of performance is 2.17 and 1.70, respectively. The sensitivity of the braking efficiency of the driving shoe to change in the lining's frictional coefficient increases with increase in γ_a. That decreases the operational stability of the brake mechanism.

In Fig. 3c, we show the dependence of the inclination of the shoe support. With zero inclination, the maximum pressure at the driving shoe is in the lower section at the shoe support; for the driven shoe, by contrast, the maximum pressure is in the upper part of the lining at the clamp. With increase in γ_a, the maximum pressure at the driving shoe is shifted to the upper part of the lining toward the clamp, while the location of the maximum pressure at the lining of the driven shoe remains unchanged. Since the pressure distribution over the length of the lining determines its wear, it is obvious that the inclination of the shoe support greatly affects the lining wear for the driving shoe. Negative pressure values in the lower part of the lining for the driven shoe indicate that this lining is not loaded over the whole length but only where the pressure is positive [11]. By varying β and γ_a, we may ensure positive pressure over the whole length of the lining.

CONCLUSIONS

We have investigated how the inclination of the shoe support affects the output parameters of the brake mechanism. We find that it does not affect the frictional torque of the drum brake but significantly affects the pressure distribution over the arc of the lining, especially for the driving shoe, and also the brake's margin of performance. Therefore, the inclination of the shoe support must be selected on the basis of the need to ensure the required shoe wear and the brake's performance.

On the basis of the proposed formulas, the influence of various combinations of structural parameters on the output parameters of the brake mechanism may be rapidly analyzed at the design stage, and parameters ensuring the required braking torque and guaranteeing the required brake performance may be selected.

CONFLICT OF INTEREST

The author declares that he has no conflicts of interest.

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