THEORETICAL AND EXPERIMENTAL INVESTIGATION OF THE FORCE OF THE PAD FRICTION ON THE BRAKE DISC

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ABSTRACT: The braking is an extremely important stage in the running of a railway vehicles as far as the safety of traffic is concerned due to the multitude and diversity of factors that may influence it. The disk brake was first mounted on the passenger cars designed for the suburban traffic, while at present it acts as a main brake in the high speed trains that run along the high speed railroads in Europe and all over the world. The paper presents theoretical and experimental results on the forces produced between coupling elements of the disc brake friction (the pads – brake disc). So, after a theoretical calculation of distance (a), from the brake disk centre at which force F_s has to apply so as to generate an approximately even wear of the friction lining in this paper it is presented a force cell designed and produced by the author in order to experimentally determine not only the pressure distribution of the friction lining on the brake disk, by measuring force in three points located at the marginal points of the area, but also the pressure of the friction lining on the disk by subsuming the three component forces. The problem tackled is the achievement of three flexible, identical and adequately shaped elements supplied with a tensiometer acting as transducers and forming a force cell mounted on a plate. The experimental determinations carried out by means of this force cell allow for the detection of certain flaws in the construction of the 'tongs' type brake linkage.

Key-words: force transducer, pad friction, brake disc, force cell.

1. INTRODUCTION

The disk brake was first mounted on fast trains, then on trains designed for the suburban and urban traffic both on passenger and freight cars running at speeds higher than 120 km/h for the following reasons:

- The braking power margin of the shoe-brake has been surpassed mainly at high speeds;
- The maintenance of the disk brake is cheaper;
- The comfort of the journey is increased;
- The variation of the friction index against the specific speed and pressure is lower;
- The pressure forces of the friction lining on the disk brake are smaller as compared to those occurring at the shoe-brake and therefore smaller brake cylinders and simpler brake linkages can be used.

2. THE DISK BRAKE CALCULUS

An important value for the calculus of the disk brake is the *mean friction radius* (Fig.1), [1], [2].
Further on, the braking moment has to be determined. At a symmetrical load and an even distribution of pressure all over the surface of the friction lining, the braking moment given by the disk is:

$$M = F_S \cdot r \cdot \mu_S \quad [\text{daNm}] \tag{1}$$

where: F_s – the pressure force of the lining on the disk, [daN];
r - the mean radius of the brake disk;
μ_s - the friction index of the friction lining on the brake disk.

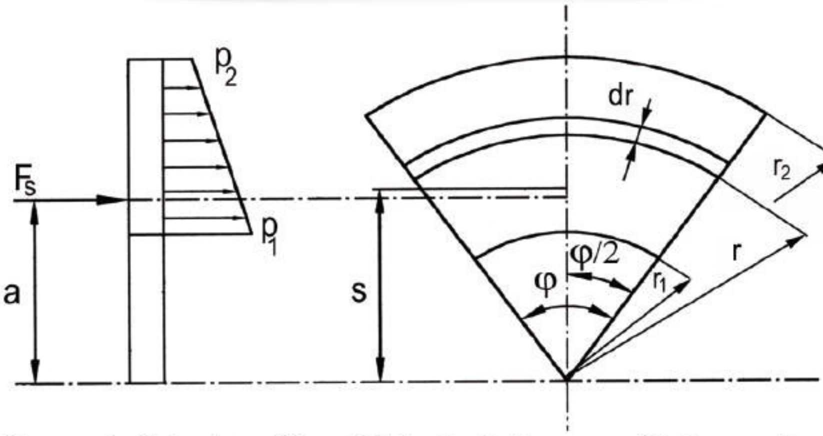

Figure 1. Calculus of the disk brake is the mean friction radius

The pressure between the disk and lining is considered to be evenly distributed only on a narrow area of its surface. Due to the great differences of speed between the points on the gray iron disk, located closer or farther away from the wheel axis, the friction lining wears off unevenly if the pressure of the lining on the disk applies to its load centre. In this case, pressures occur between the lining and the disk, which are smaller or bigger depending whether the areas are closer or farther away from the wheel axis. An even wear off of the friction linings is ensured if the pressure force applies far from the wheel axis (Fig.1.)

Fig.1 also shows that in the case of an asymmetrical clamping force F_s, on the upper side of the disk, to which relatively higher speeds correspond, minimal pressures p_2 occur, while in the lower part of the disk, lower speeds and maximum pressures p_1 occur.

Taking into account the fact that the wear of the friction lining can be determined via the mechanical friction work done, in order to ensure an even wear off of the lining the following condition is admitted:

$$p \cdot v = k_1 = \text{const.} \tag{2}$$

where: p- the pressure between the friction lining and the brake disk in the surface element considered [daN/cm^2];

v – the relative speed between the friction lining and the brake disk in the considered spot [m/s];

k_1 – proportionality constant.

As the relative speed between the lining and the disk is proportional to radius r, it may be written that:

$$v = r \cdot k_2 \tag{3}$$

k_2 – proportionality constant.

Given this relation, condition 2 may be written:

$$p \cdot r = k = \text{const.} \tag{4}$$

or

$$p = \frac{k}{r} \tag{5}$$

k – proportionality constant

It can be noticed from Figure 1 that the pressure of the friction lining on the brake disk can be determined by subsuming the elementary forces:

$$\begin{aligned} F_s &= \int_{r_1}^{r_2} p \cdot \varphi \cdot r \cdot dr = \int_{r_1}^{r_2} \frac{k}{r} \cdot \varphi \cdot r \cdot dr = \\ &= \int_{r_1}^{r_2} k \cdot \varphi \cdot dr = k \cdot \varphi \cdot (r_2 - r_1) \end{aligned} \tag{6}$$

Starting out from the equivalence of the moment of asymmetrical forces F_s to the moment of clamping forces symmetrically applied F on arm s, the following formula can be written:

$$F_s \cdot a = F \cdot s = \int_{r_1}^{r_2} p \cdot r \cdot s \cdot \varphi \cdot dr \tag{7}$$

a - the distance, from the brake disk centre at which force F_s has to apply so as to generate an approximately even wear of the friction lining (to be determined).

Considering the expression of the load centre of the disk arc as s: $s = \dfrac{r \cdot \sin\frac{\varphi}{2}}{\frac{\varphi}{2}}$ (8), and relation (5), the equivalence of the moments will be as follows:

$$F_s \cdot a = \int_{r_1}^{r_2} p \cdot r \cdot r \cdot \frac{\sin\frac{\varphi}{2}}{\frac{\varphi}{2}} \cdot \varphi \cdot dr =$$

$$= \int_{r_1}^{r_2} \frac{k}{r} \cdot r \cdot r \cdot \frac{\sin\frac{\varphi}{2}}{\frac{\varphi}{2}} \cdot \varphi \cdot dr = \qquad (9)$$

$$= \int_{r_1}^{r_2} k \cdot 2 \cdot r \cdot \sin\frac{\varphi}{2} \cdot dr = k \cdot \sin\frac{\varphi}{2} \cdot \left(r_2^2 - r_1^2\right)$$

If the expression of force F_s from relation (6) is replaced with relation (9) the definition relation of distance a obtains:

$$k \cdot \varphi \cdot (r_2 - r_1) \cdot a = k \cdot \sin\frac{\varphi}{2} \cdot \left(r_2^2 - r_1^2\right)$$
$$\Rightarrow a = \frac{1}{\varphi} \sin\frac{\varphi}{2} \cdot (r_2 + r_1) \qquad (10)$$

This relation (10), [1], [3], [4] represents the distance, from the brake disk centre at which force F_s has to apply so as to generate an approximately even wear of the friction lining.

3. DESCRIPTION OF THE FORCE CELL

The task of the force cell is to determine experimentally not only the pressure distribution of the friction lining on the brake disk by measuring the force in 3 points located at the marginal points of the area, but also the determination of the force of the friction lining on the disk by subsuming the three component forces.

The problem is solved via the production of three flexible, identical and adequately shaped elements provided with a tensiometer and fitted on a plate, thus forming a force cell capable of fulfilling the aforementioned task.

Further on, the paper illustrates an example of a force cell (Figures 2-4), [2]:

- A view of the force cell (Figure 2);
- The location of the electrical variable-resistance transducers on the flexible element (Figure 3);
- The connection of the electrical variable-resistance transducers onto the electrical balance (Figure 4).

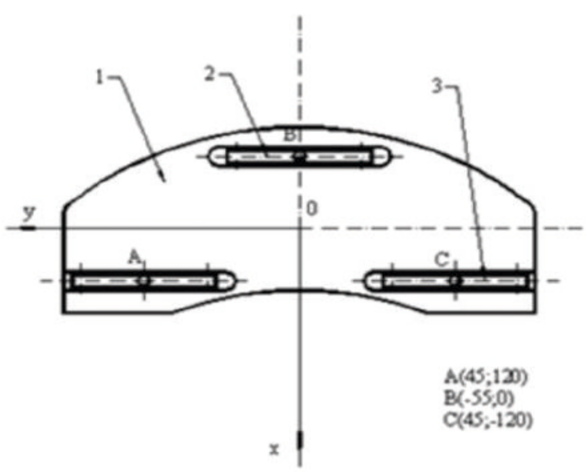

Figure 2. Force cell view

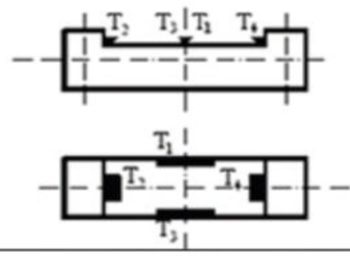

Figure 3. The location of the transducers in the flexible element

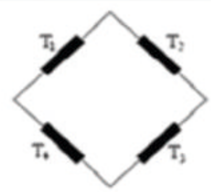

Figure 4. The connection of the electrical variable resistance transducers onto the electrical balance

The force cell determining the pressure and pressure distribution of the friction linings on the brake disk consists of a *plate* 1 that has a shape similar to the friction lining used for the respective brake and *flexible elements* 2 adequately shaped, fitted with electrical variable-resistance transducers T1……T4 *and* displaying also *supports* 3 which during braking lean against the brake disk.
The measuring of the component forces in points A, B and C, i.e. : F_A, F_B, F_C is done by determining the signal obtained at three electrical balances made of electrical variable-resistance transducers connected as per Figure 4, for each flexible element and for each measured component force respectively.
If a suitable shape is chosen for the plate and the flexible elements, these elements assembled together form a force cell which can be placed instead of the existing friction lining, thus allowing for experimental determinations of the braking pressure and of its distribution on the surface of the friction lining under steady- state conditions.

The force cell designed for the determination of the pressure distribution of the friction lining on to the brake disk has the following advantages:

- It allows for the measurement of the pressure exerted by the friction lining on the disk of the disk brake;
- It allows for the measurement and determination of the pressure distribution of the friction lining on the brake disk in the three points considered;
- Depending on the values of the pressure in the three points considered , it can be established whether the pressure is evenly distributed, and the necessary technical measures can be taken in order to generate an equalizing of the values of the three components of the friction lining pressure;
- The shape of the force cell allows for an easy mounting/dismounting, observing the same technology as the one used for the mounting/dismounting of the friction lining.

4. EXPERIMENTAL MEASUREMENTS

Using the existing force cells, experimental tests have been carried out upon various passenger car bogies, currently running on the railroads in Greece. These bogies have been provided with disk brakes driven by two 254 mm cylinders.
The measurement chain was composed of 4 force cells connected to UMK 10 switch boxes and a KWS 3050 Hottinger Baldwin Messtechnik amplifier. The supply of the two bogie brake cylinders has been made via a KEs Dü 21C/1.27 air sparger and a mobile braking stand fitted with a KD2 (Knorr D2) valve tap. The trial pressure was of 3.8 daN/cm^2.
The results of the experiments are shown in Table 1 for the bogie that was used in traffic and in Table 2 for the same bogie after the application of the measures designed to make the necessary corrections for the pressure distribution, so as to maintain an uneven wear below 22%. To fulfill this aim it is necessary that $y_F < 5.7$ mm and $x_F < 2.33$ mm.

Table 1. The results of the experiments for the bogie that was used in traffic

Bogie no.	Shaft neck	Pick-up	F_A [daN]	F_B [daN]	F_C [daN]	F_T [daN]	x_F [mm]	y_F [mm]
0081/1980	1	I	562	689	631	1882	8,39	-4,39
		E	216	931	638	1785	-7,15	-28,36
	2	I	530	863	463	1856	-1,49	4,33
		E	390	648	781	1819	9,37	-25,79
	3	I	505	992	340	1837	-9,00	10,77
		E	633	718	478	1829	5,74	10,16
	4	I	74	687	688	1449	-2,28	-50,84
		E	676	825	155	1656	-4,81	37,75

1, 2, 3, 4 – the shaft neck number of the two axles of the bogie (every axle has two shaft necks)

Values x_F and y_F represent the coordinates of the point in which the resultant pressure applies against the reference system. The denotations in the tables stand for:

- F_A, F_B, F_C - the determined forces in point A,B,C;
- F_T the total pressure force on the friction lining;
- I and E – internal and external positioning of the force cell to the brake disk.

Analyzing the results presented in Table 1 it can be noticed that the distribution of force F_T is

uneven; the point where the overall force applies being displaced considerably.
Following the corrections applied, coordinate y_F was brought to values below 5.7 mm.

The values higher than 2, 3 mm of coordinate x_F were considered to have a very small influence on the uneven wear of the friction lining.

Table 2. The results of the experiments for the same bogie after application of the measures designed to make the necessary corrections for the pressure distribution

Bogie no.	Shaft neck	Pick-up	F_A [daN]	F_B [daN]	F_C [daN]	F_T [daN]	x_F [mm]	y_F [mm]
0081/1980	1	I	576	573	599	1748	12,22	-1,58
		E	382	924	455	1761	-7,47	-4,97
	2	I	534	677	576	1787	7,12	-2,82
		E	456	871	473	1800	-3,39	-1,13
	3	I	542	755	590	1887	4,99	-3,05
		E	376	1029	445	1850	-10,62	-4,47
	4	I	468	730	404	1602	-0,57	4,79
		E	372	890	416	1678	-8,04	-3,15

5. CONCLUSIONS

A great variation in the distribution of the pressure force of the unevenly worn friction linings has been found. This fact comes as a consequence of the geometrical and kinematical deviation of all of the brake component elements. It has also been found out that there is a close dependence between the values of pressure forces F_A, F_B, F_C on the surface of the lining and the values of the wear.

The lack of parallelism between the surface of the friction linings and the brake disks leads to uneven wear oriented along the friction force.
The favorable results found at bogies after a period of service during which correction measures for the pressure force have been applied, prove the usefulness and efficiency of the experimental measures carried out via the force cell devised by the authors of the paper.

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