EXPERIMENTAL TEST RIG FOR VEHICLE BRAKE PAD EVALUATION

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Abstract: *The paper presents details regarding an original test rig conceived to evaluate the friction generated by the brake pads on steel flat surfaces. The experimental equipment consists of a vehicle brake system, driven with constant velocity by an electric motor. The friction magnitude can be adjusted by aim of a hydraulic pump and a pressure gauge. The friction torque is measured using an original device made from two parallel flanges; one of them is fixed to the driven shaft while the second one is connected to the driving shaft. The flange connected to the driving shaft has three helical springs placed between hub and cylindrical body. The friction torque forces the springs to deform. The magnitude of the displacement is proportional to the friction torque and was measured by aid of an optic transducer. Several measurements were conducted at different clamping forces and the results show that the experimental friction coefficient is close to those presented in literature.*

Keywords: friction, automotive, efficiency, wear, safety

1. Introduction

Brake pads are one of the most important brake system consumable parts of a road vehicle and play a vital role in vehicle safety and performance. Brake pads are essential for slowing down or stopping a vehicle. They create friction against the brake disc, which conducts to slow down the wheels. Properly functioning brake pads are crucial for safe and effective stopping, reducing the risk of accidents.

Brake pads endure substantial wear due to the heat generated during braking. High-quality brake pads can withstand this stress and last longer, reducing maintenance costs and enhancing the vehicle's overall reliability, [1].

Properly behavior brake pads can indirectly contribute to better fuel efficiency meaning that brake pads may lead to increased fuel consumption.

Well-maintained brake pads contribute to a smoother and more comfortable driving experience. They help prevent juddering, vibrations, or noise that can occur when the brake pads are worn or damaged, [2,3].

2. Vehicle's brake pads manufacturing

Brake pads are typically manufactured through a multi-step process involving various materials, [4,5]. The specific manufacturing process can vary among different brake pad manufacturers, but a series of steps is generally used.

The most common materials for the lining friction surface include: organic compounds, semi-metallic or ceramics materials, [4]. Usually, the backing plates are made of general-purpose steel. Depending on the brake pad type, various raw materials, including resin binders, abrasives, and reinforcing materials, are mixed, and blended together to form a uniform compound. This compound is what provides the necessary friction and wear resistance for the brake pad. The compound is then pressed and shaped into the desired form for the brake pad. This process can involve hydraulic presses to create the pad's specific size and shape, [6].

The shaped brake pad material is subjected to a curing process, which typically involves heat. This cures the resin binders and hardens the pad. It's during this step that the materials bond together to form a solid and durable brake pad, [6].

Brake pads are often slotted and chamfered to improve performance. Slots and chamfers help with heat dissipation, reduce noise and vibration and improve the initial bite of the brakes, [7].

The present paper presents an original test rig used to evaluate the friction generated by friction pads in contact with a steel brake disc.

3. The experimental test rig

The experimental device shown in Fig. 1 is used to perform the measurements. With its help, the moment of friction that occurs between the active elements of the braking system and disc can be determined.

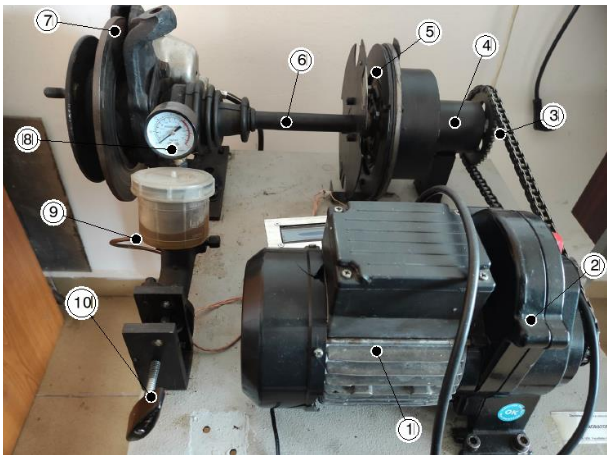

Figure 1: *Brake pad experimental test rig*

The experimental device consists of a series of components such as: electrical motor (1), gear transmission (2), chain transmission (3), drive shaft (4), elastic coupling (5), driven shaft (6), disc and pad brake mechanism (7), pressure gauge (8), hydraulic clamping force transmission system (9) and the actuator (10). In order to estimate the friction magnitude for different clamping forces, an optical torque monitoring device has been implemented. Using this device, the measurement friction torque and the calculus of the friction coefficient established between the active elements of braking system was possible. The structure of the optical torque monitoring device was represented graphically in Fig. 2. The elements highlighted in Fig.2 are: the driving shaft (1), support flange (2), helical springs (3), driven shaft hub (4), the support flange for the obturator element (5), driven shaft (6) and obturator elements (7) which is a metal strip.

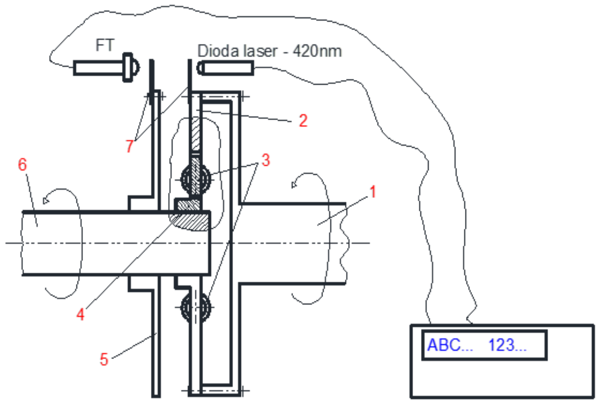

Figure 2: *Optical torque monitoring device structure.*

The driving shaft (1) has a fork at one end to which the support disc (2) is attached. This disc has a series of recesses into which one of the ends of the four helical springs (3) are inserted. The opposite ends of the helical springs are positioned in the seating surfaces of the hub (4). Thus, the transmitted torque (T) and rotational movement between the driving and the driven shaft is achieved by means of an elastic coupling. The support flanges (2 and 5) have fitted on the circumference, two metal strips (7) whit size of 40 mm wide and 30 mm high.

To obtain the friction torque generated by the friction pads and the disc, a correlation between relative displacement of the strips and torque was necessary. The correlation was obtained in static conditions. By attaching known weights on one of the ends of an arm in perfectly horizontal position and measuring the relative displacement between strips, the torque generated between the driving and driven shafts was evaluated. The experimental results were recorded and represented in Fig. 3.

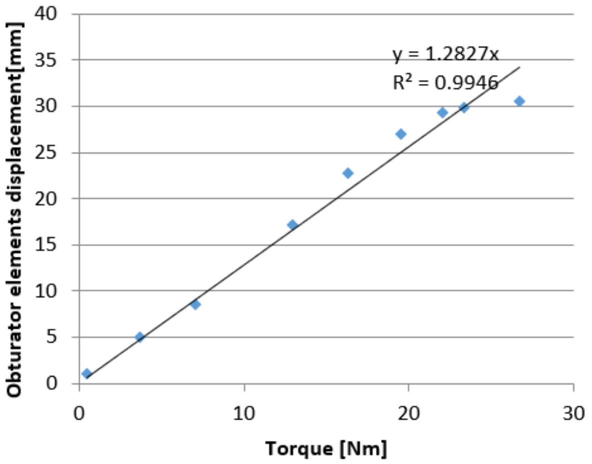

Figure 3: *Torque – strips relative displacement correlation*

The obtained results were interpolated using a linear function.

In moving conditions, the magnitude of the relative displacement between the two strips is measured using a Tektronix TDS3032B acquisition oscilloscope shown in Fig. 4 and an optical device.

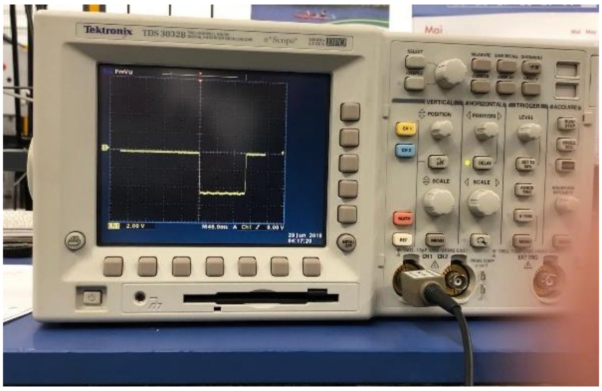

Figure 4: *Tektronix TDS3032B electronic oscilloscope.*

4. Experimental setup

The efficiency of the classic braking system is closely related to the value of the friction torque generated between the friction surfaces when the clamping force is applied. To evaluate the friction torque, the experimental test rig is driven by aid of an electric motor, respectively a chain transmission. The electronic oscilloscope was connected to the terminals of an optical sensor. The blocking time of the light generated by a laser diode was measured using a phototransistor based on the graph displayed by the oscilloscope.

The operating speed of the disc on the driving shaft was determined by calculus considering the motor nominal speed and transmission ratio.

Knowing the distance from the rotation axis of the driving shaft to the strip geometrical center (*r=0.125m*) and the angular velocity of the driving shaft (*ω=37rpm*), using equation (1), the circumferential velocity of the strips elements is obtained and has a value of 0.772m/s considering no-loading conditions.

$$v = r \cdot \omega \cdot \frac{\pi}{30} \left[m/s \right] \tag{1}$$

By loading the experimental test rig, the blocking time of the light received by the photoreceptor will increase proportional with the torque value, according to Fig. 3 due to the relative displacement between strips. Knowing the circumferential speed of the strips and the blocking time of the signal measured using the oscilloscope (Div), equation (2) can be used to determine the delay width of the strips (Δ*x*) with loading.

$$\Delta x = v \cdot \Delta t \left[m \right] \tag{2}$$

The initial width of the strips band (with no loading) was measured (*l*). Using equation (3), the relative displacement (φ) between strips elements, in loading conditions, can be calculated if at each new loading level, the blocking time of the signal recorded by the oscilloscope is measured.

$$\varphi = \Delta x - l \left[m \right] \tag{3}$$

Under no-loading conditions, the value of the blocking time was found 84ms.

Several loading levels were considered corresponding to different values of pressure in the hydraulic loading system. Were performed tests corresponding to: 1 bar, 1.5 bar, 2 bar and 2.5 bar respectively. For each of these values, the parameters were recalculated as a function of the shut-off time and the value of the delay between the two strips elements was determined.

The obtained values are summarized and shown in Table 1 and Fig. 5.

Table 1: *Experimental values for different braking regimes*

Div	Δt [ms]	Δ*x* [mm]	φ [mm]	P [bar]	F_s [N]
2.1	84	40.68	0.68	0.0	0.00
2.3	92	44.56	4.56	1.0	125.66
2.4	96	46.50	6.50	1.5	188.50
3.0	120	58.12	18.12	2.0	251.33
3.2	128	61.99	21.99	2.5	314.16

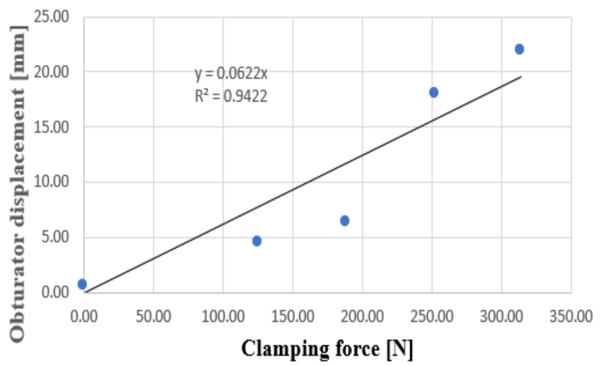

Figure 5: *The correlation between the clamping force and the displacement of the strips*

The clamping force (Fs) was calculated by evaluating the area of brake cylinder with 20mm radius and the hydraulic pressure value (P) shown on the pressure gauge installed on the test rig.

The blocking period of signal generated due to the light reaching to the photoreceptor (Δt) is shown in Fig. 6, obtained by screen capture of the oscilloscope. Several signals were captured corresponding to four values of pressure in the brake system: a) for 1bar, b) for 1.5bar, c) for 2bar and d) for 2.5bar.

Based on the data presented for the calibration curve of the device (Fig. 3), which estimates the transmitted torque (T) correlated with the relative displacement between strips it is possible to determine the braking torque (M_f) which is also the transmitted torque according to the principle of action and reaction.

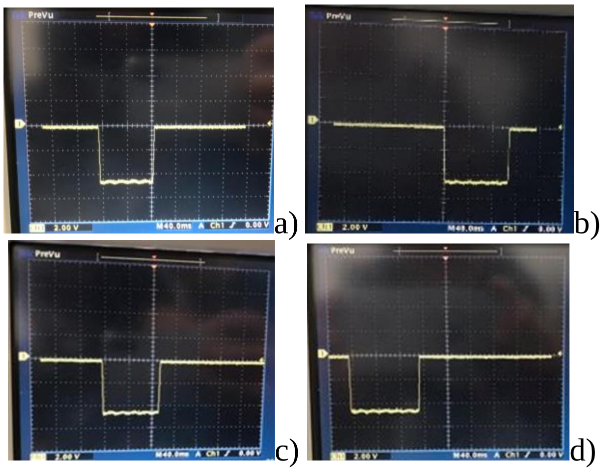

Figure 6: *Light signal blanking time.*

Using the correlation between the relative displacement between strips elements and the clamping force of the braking elements, a linear interpolation was made. The interpolation parameters were used to obtain a representation between the braking force (Ff) and clamping force as is shown in Fig. 7. The obtained data are summarized in Table 2.

Table 2: *Experimental values for torque and braking force*

Div	Δx [mm]	F_s [N]	M_f [Nm]	F_f [N]
2.1	40.68	0.00	0.55	4.41
2.3	44.56	125.66	3.68	29.43
2.4	46.50	188.50	5.24	41.94
3.0	58.12	251.33	14.62	116.99
3.2	61.99	314.16	17.75	142.01

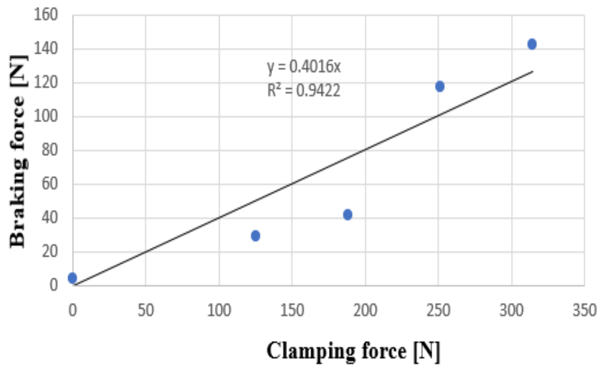

Figure 7: *Dependence between the braking and clamping force*

The relative displacement between the strip elements can be transformed into values of the transmitted torque by interpolation with the values shown in the calibration graph of the device through which the torque was measured. The value of the friction coefficient was obtained, and it has a value of 0.32.

5. Results and conclusions

The present paper presents an original test rig built and conceived to evaluate the friction generated between two brake pads and a brake disc. The experimental test rig was constructed by aim of a brake system extracted from a vehicle. The clamping force is generated by a hydraulic pump connected to the brake system. The instant hydraulic pressure is measured with a pressure gauge and controlled by a screw-nut mechanism. The equipment is driven by an electric motor. The friction torque is evaluated by an original device which consists in two parallel flanges. One of them is rigid and is fixed to the driven shaft, while the second one is mounted with springs on the conducting shaft. The friction torque leads to the compression of the springs and a delay between the flanges occurs. The magnitude of the delay is proportional to the friction torque.

Several measurements were conducted using the above presented experimental test rig and the magnitude of the friction torque corresponding to different clamping forces was obtained. The experimental results reveal that the friction torque has a magnitude of 17.75Nm, corresponding to 314N clamping force. The obtained results were used to determine the friction coefficient. According to [8] for urban vehicles the domain values of the friction coefficient can vary between 0.2 and 0.45. Our own results show that the coefficient of friction is 0.32. This value validates the experimental test rig and the testing methodology.

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