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REGULATION OF THE FRICTIONAL CHARACTERISTICS DISC BRAKE

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Abstract: The article presents the results of an experimental study of the influence of external vibration disturbances on the working elements of the friction unit, which simulate the power circuit of the disc brake: brake pad and brake disc. The studies were carried out on the original friction machine reproducing the interaction conditions of the disc brake elements on a scale of 1:100. Vibration impact on the working elements of the friction unit was carried out by means of a drive, which allowed to implement a sequence of power pulses in the frequency range 5–100 Hz. It is shown that due to the vibration disturbances of the working elements it is possible to influence the friction characteristics within 20–25%.
Keywords: disc brake, friction machine, vibration perturbation.

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Introduction

Currently, the control of friction characteristics during braking of rolling stock is carried out according to the criterion of wheel slippage relative to the rails. Excessive slippage is leveled by reducing the braking torque on the axle of the wheel pair and (or) increasing the coefficient of adhesion in the contact zone of the wheels with the rails. However, the disadvantages of this method include the lack of effective tools for metered smooth regulation of the characteristics of interconnected tribological systems: "brake pad – brake disc" and "wheels – rail".

In this regard, the method of controlling the characteristics of the disc brake due to periodic pulsed force effects on the friction elements in the normal and tangential direction relative to the plane of their interaction is of interest [1, 2].

Experimental

In order to confirm the possibility of practical implementation of this method, an experiment was conducted on a specially designed stand [3, 4]. The basis of the stand was a friction unit, modeling the working elements of the disc brake, made in scale 1:100. The functional scheme of the stand is shown in the figure 1, 2.

Materials of friction unit pair: "structural steel 45 – bronze (Sn–10%, Zn–5%, Pb–5%)". The choice of materials for a pair of friction units was random in many respects situational in nature, since the aim of the experiment was a qualitative initial confirmation of the method. The DC motor 1 with a capacity of 0.1 kW through the reducer 2 rotates the steel disc 3. The load 4 using the lever system 6 presses the brake pad 5 to the surface of the rotating disc 3. Periodic impulse action on the working elements of the friction pair in the normal direction is carried out by means of a magneto dynamic drive 7. Pulses are generated by the oscillator 8 in the frequency range $10 \div 5 \cdot 10^2$ Hz.

The moment of friction forces is estimated indirectly on the basis of the measurement data of the voltage values applied to the armature of the electric motor by the compensation method. The speed of rotation of the disk 3 is maintained unchanged by including the electric motor in the automatic speed control system. Speed feedback is carried out by means of tachogenerator 9. The difference signal between the values of the set and actual speeds (1) from the output of the discriminator 11 is fed to the input of the regulator 12.

$$\Delta\omega = \omega_{preset} - \omega_{current} \tag{1}$$

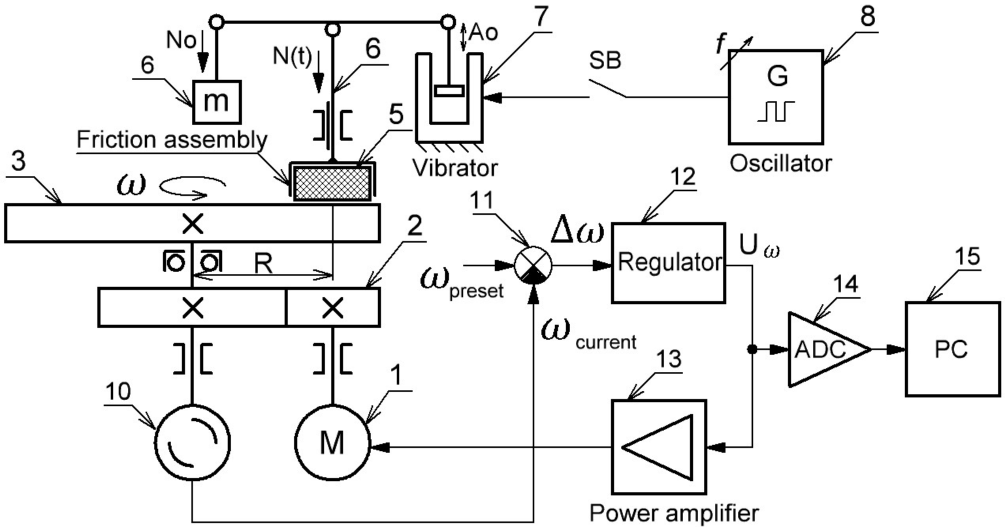

Figure 1. Functional diagram of the machine of friction.

The output signal of the regulator $U_{\omega}(t)$ through the amplifier 13 is fed to the power terminals of the DC motor 9. The analog-to-digital Converter 14 converts the analog signal $U_{\omega}(t)$ into a digital representation for its subsequent registration and processing in a personal computer 15.

The torque on the shaft of the motor with independent excitation M_{engine} under the condition of the constant speed of rotation of the shaft $\omega = const$ is a function of the voltage U_{ω} and determined by the gradient of its mechanical characteristics.

$$M_{engine} = k_m \cdot U_{\omega} \tag{2}$$

The purpose of the experiment is to study the effect of periodic force pulse effects on the characteristics of the friction elements of the stand, modeling the brake disc and brake pad.

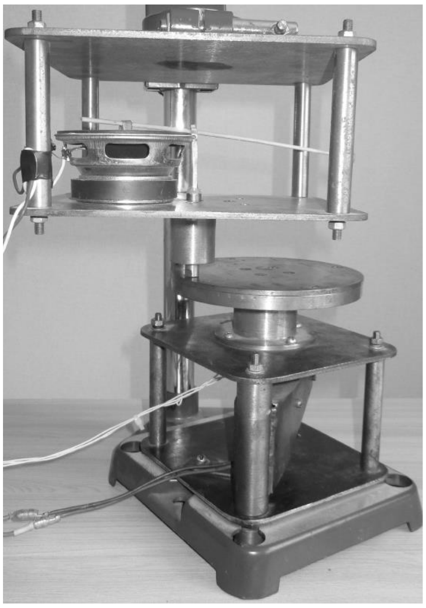

Figure 2. Appearance of the experimental stand.

Loaded pad disc spins up the motor before reaching the steady state rotation frequency of 150 rpm. the Static normal loading of the friction node elements in the range N_0=3÷5 N (site characteristics I, figure 3).

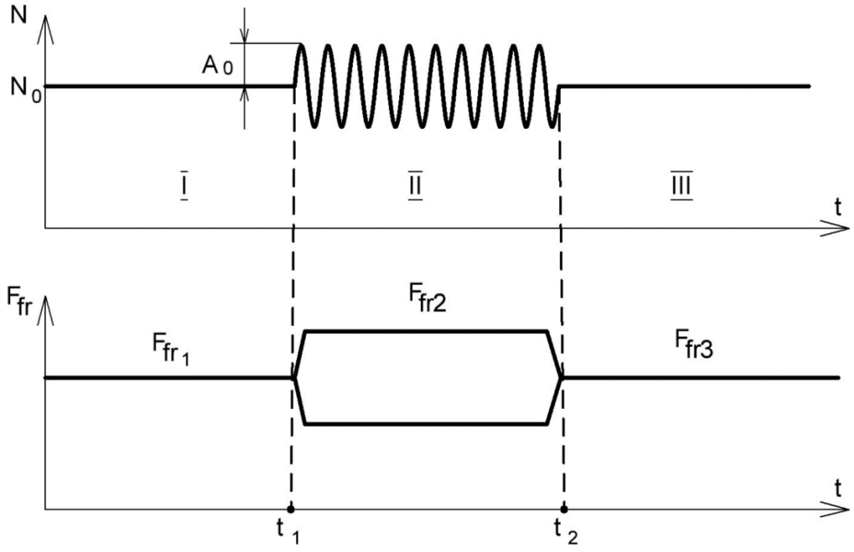

Figure 3. The experiment plan.

At the time t_1, a magnetodynamic drive is activated, which adds a dynamic component with amplitude A_0 and frequency f to the static value of the established normal force of pressing the friction elements N_0

$$N(t) = N_0 + A_0 \cdot \sin(2 \cdot \pi \cdot f \cdot t). \tag{3}$$

In this case, the amplitude of the dynamic component of the normal force is less than the static pressing force N_0 (section of characteristics II, figure 3).

At time t_2, the magnetodynamic drive is switched off. The normal loading of the friction node elements returns to the initial value in the range N_0 = 3÷5 N (characteristic section III, figure 3).

The torque given to the motor shaft is defined by the expression

$$M_{engine} = \frac{M_d + M_{idl} + M_{fr}}{j_{rd} \cdot \eta_{rd}}, \tag{4}$$

where: M_d – the dynamic moment, M_{idl} – static torque loss at idle, M_{fr} – static moment of forces of friction in the tribological system, j_{rd} – gear ratio, η_{rd} – efficiency gearbox.

We transform expression (2) to determine the moment of friction forces in a tribological system

$$M_{fr} = M_{engine} \cdot j_{rd} \cdot \eta_{rd} - M_d - M_{idl} \tag{5}$$

The values j_{rd} and η_{rd} is constant and known. Under the condition of constant speed of rotation of the disk, the dynamic component of the torque on the shaft of the motor 9 can be neglected $M_d = 0$. The static moment of the load losses M_{idl} is constant and can be determined experimentally by the condition $N(t)$=0.

The friction force in the tribological system is determined by the expression

$$F_{fr} = \frac{M_{fr}}{R}, \tag{6}$$

where: R – average radius of friction force application to the disc surface 3, F_{fr} – the friction force in a tribological system.

To obtain the calculation formula, convert the expression (6) taking into account (5) and (2)

$$F_{fr} = \frac{M_{engine} \cdot j_{rd} \cdot \eta_{rd} - M_{idl}}{R} = \frac{k_m \cdot U_\omega \cdot j_{rd} \cdot \eta_{rd} - M_{idl}}{R} \tag{7}$$

Results and discussion

Figure 4 shows the results of three measurement cycles.

As a result of the experiment, an increase in the friction force in the studied friction node was observed (section of characteristic II, figure 4).

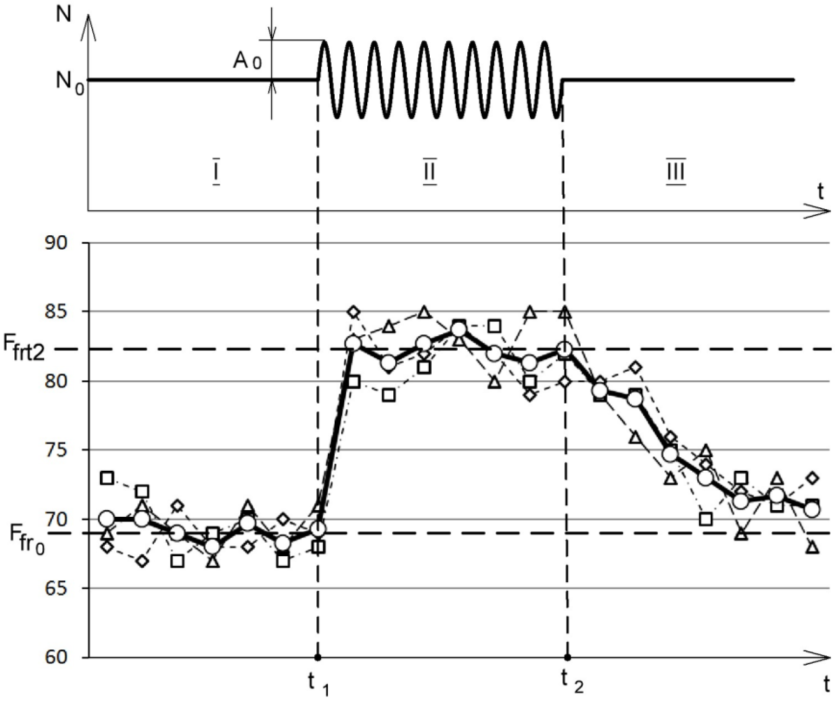

Figure 4. Experimental result.

After switching off the magnetodynamic drive and removing the dynamic component of the pressing force at time t_2, the value of the friction force returned, with some delay, to the original values (section characteristics III, figure 4).

Conclusions

The experiment allowed us to establish that the periodic impulse force action on the friction element, simulating the brake pad, can be used as a tool for selective, smooth control of the friction coefficient arising in the tribological pair "brake pad – brake disc".

The predicted value of the regulation of the coefficient of friction in the actual tribological pair "brake pad-brake disc" is expected to be 20–25%.

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