Simulation and Experimental Analysis of Quality Control of Vehicle Brake Systems Using Flat Plate Tester

A.I. Fedotov and M. Młyńczak

Abstract Paper describes simulation analysis of errors that arise in the control process of brake systems for vehicles using flat testers. Technical state of brake systems and their performance in operation are important aspects of safety. Method of numerical simulation of dynamic systems is described concerning vehicle braking capability. It is proposed dynamic model of the tire on the flat plate brake tester. Simulation model takes into account all factors influencing measurement error resulting from testing method. Methodology of brake testing on flat testers and measurement system is analyzed. There are discussed measurement errors observed on flat testers related to the dynamic properties of the tire, load on the wheel, mass of tester plate as well as speed and braking time and force applied by a driver. Calculations are illustrated with graphs. Square correlation coefficient of assumed functions is not less than 0.95.

Keywords Flat braking testers · Brake testing · Measurement error

1 Introduction

Vehicle brake testing is an obligatory periodic maintenance carried out in workshops equipped in special devices measuring brake parameters [2, 5]. Brake tests use two ideas based on rolling drums or flat plates suspended elastically [5–7]. Rolling drums are more popular and have more accurate measurements, though flat testers are more universal as can be used to test shock absorbers and suspension [6, 7]. A vehicle moving on plates engages brakes and causes forces and moments proportional to

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brake efficiency. Flat plate brake testers have usually two plates for wheel of the same axis with sensors measuring longitudinal and vertical forces. Computer application processes data and give an assessment about vehicle brake conditions. Problem with measurement accuracy appears due to wheels positioning on the plates, what is analyzed in the first part of the paper and on dynamic parameters of the system wheel—tester plate. Results depend on many dynamic parameters of braking process. Paper presents simulation approach to analysis of influence selected parameters on brake test results. The problem is concern because of its importance to the road safety. Brake testing is one of the elements of the conditional preventive maintenance. Safety depends in that case on: brake performance measured as ratio of total braking forces over vehicle weight, uniformity of braking forces of left and right wheel and dependence of braking force from the pressing force on the pedal. All that forces should bring a vehicle to stop at predefined distance and provide straightforward path of the move while braking. Brake systems are repairable objects, though not many accidents are caused by their failure [9]. Brake system was a cause of 18 out of over 40,000 road accidents what is 10^{-4}. High influence of that system on safety and strong requirements tested periodically makes those systems as highly reliable.

2 Basic Parameters of Flat Brake Testers

Flat brake tester consists of two plates to run onto it by two wheel of the same axis of a vehicle [2, 5–7]. Figure 1 shows a diagram of flat plate tester with the left—2 and right—3 plates with wheels of the car. The wheel may be located to the left or to the right side of the plate axis of symmetry. The best location is exactly on that axis but it is difficult to perform in real conditions, therefore it is observed distance

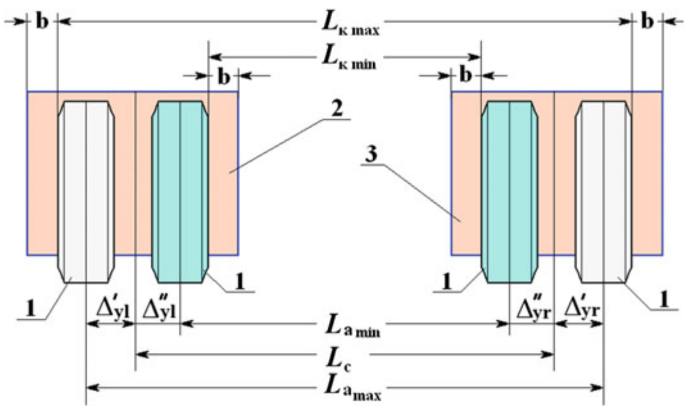


Fig. 1 Scheme for the geometric dimensions flat tester: *1* car wheel, *2 left* plate of the tester, *3 right* plate of the tester

between the plate axis of symmetry and current position of tire axis of symmetry, described as: Δ_{yl} and Δ_{yr} for left and right wheel respectively.

In order to determine the width of the pads and the distance between them it is necessary to know the following geometric dimensions:

1. $L_{\text{к max}}$—the maximum distance between the outer sides of the car wheels having broad wheel track,
2. $L_{\text{к min}}$—the minimum distance between the inner sides of the car wheels having narrow wheel track,
3. b—side margin to guarantee position of wheels on the surface of tester plate,
4. L_c—distance between the axes of symmetry of tester plates.

3 Computer Simulation of Flat Brake Tester Dynamics During Measurement Process

To analyze the process of interaction of the vehicle wheel with braking flat tester was drawn a scheme shown in Fig. 2. The model allows for analytical study of dynamic processes from the moment of run onto tester at point A until it stops. Vehicle of the weight M moves under the force of inertia F_{jx}^{M}. In the point of tire contact with the supporting surface reaction R_x arises. The scheme allows us to

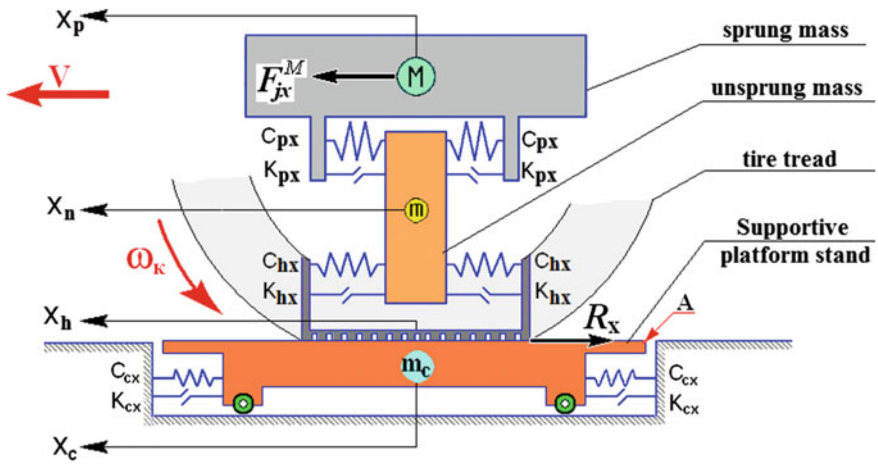


Fig. 2 Scheme of the interaction of a vehicle wheel braking pads with tester where: M part of the sprung mass at the wheel of the car, m unsprung weight, m_{sh} the mass of the tire tread contact patch; m_c mass of the plate tester; K_{px}, K_{hx}, K_{cx} damping coefficients, respectively, suspension, tires and tester [N s/m]; C_{px}, C_{hx}, C_{cx} respectively suspension stiffness, tires and tester [N/m], X_p, X_n, X_h, X_c coordinates of longitudinal displacement, respectively of the sprung mass (M), unsprung mass (m), the mass of the tire tread at the contact patch (m_h) and the weight of the tester plate (m_c)

analyze the influence of the longitudinal oscillations of the masses of the car and flat tester by the uncertainty of measurement of brake forces.

To calculate the vehicle deceleration we write the differential equation of its dynamic equilibrium in the general form (1):

$$\frac{d^2X}{dt^2} = \frac{\sum_{i=1}^{n} R_x}{(M + m \cdot n)} \tag{1}$$

where: n—number of wheels.

Dynamic equilibrium equation of the braking wheels can be written as (2):

$$\begin{cases} M\frac{d^2X_p}{dt^2} = C_{px}(X_{p2} - X_p) - C_{px}(X_p - X_{p1}) + K_{px}(\dot{X}_p - \dot{X}_{p2}) + K_{px}(\dot{X}_{p1} - \dot{X}_p); \\ m\frac{d^2X_n}{dt^2} = C_{hx}(X_{n2} - X_n) - C_{hx}(X_n - X_{n1}) + K_{hx}(\dot{X}_h - \dot{X}_{h2}) + K_{hx}(\dot{X}_{n1} - \dot{X}_n) \\ \qquad - \left[C_{px}(X_{p2} - X_p) - C_{px}(X_p - X_{p1}) + K_{px}(\dot{X}_p - \dot{X}_{p2}) + K_{px}(\dot{X}_{p1} - \dot{X}_p)\right]; \\ m_h\frac{d^2X_h}{dt^2} = C_{hx}(X_n - X_{n2}) - C_{hx}(X_{n1} - X_n) + K_{hx}(\dot{X}_{n2} - \dot{X}_n) + K_{nx}(\dot{X}_n - \dot{X}_{n1}) - R_x; \\ m_{\mathrm{c}}\frac{d^2X_{\mathrm{c}}}{dt^2} = C_{\mathrm{cx}}(X_{c2} - X_c) - C_{cx}(X_{\mathrm{c}} - X_{\mathrm{c}1}) + K_{cx}(\dot{X}_c - \dot{X}_{c2}) + K_{cx}(\dot{X}_{c1} - \dot{X}_c) - R_x. \end{cases} \tag{2}$$

To relate the resulting equations system (3) with tangential reaction R_{x} we use a dynamic model of a vehicle wheel braking process [4] based on the characteristics of the stationary characteristics of elastic tires and the normalized slip function [8]:

$$f(s) = \sin\{A \cdot arctg(B \cdot s)\} \tag{3}$$

Dynamic equilibrium equation of the braking wheels can be written as (4):

$$\frac{d\omega_k}{dt} = \frac{R_x \cdot r_{ko} - M_t - M_f}{J_k} \tag{4}$$

where: M_{t}—current value of the braking torque supplied to the wheel [N m], M_{f}—moment of resistance to rolling wheel [N m], r_{ko}—the radius of the rolling wheels in the braking mode [m], J_{k}—wheel moment of inertia [kg m^2].

The current values of brake torque M_{t} applied to the wheel will be given in the form of a linear relationship (5):

$$M_{ti} = M_{t_{i-1}} + \frac{dM_t}{dt} \cdot \Delta t \tag{5}$$

where: $\frac{dM_t}{dt}$ is the rate of increase of the braking torque [N m/s].

The braking force (tangential reaction R_{x}) fulfils the formula (6) [4, 8]:

$$R_x = R_z \cdot \varphi_{max} \sin\{A \cdot arctg(B \cdot s)\}[\mathrm{N}] \tag{6}$$

where: $\varphi_{\max}$—the maximum value of the friction coefficient, A, B—the stationary slip coefficients, S—tire slippage relative to the tester area, R_z—normal reaction at the contact tire patch [N].

It is obvious, that normal reaction R_z during vehicle deceleration changes. To determine reaction R_z the scheme shown in Fig. 3 is proposed. On that basis mathematical description of the process of mass vibrations of the car and tester is made.

To move the masses M, m and m_c along the OZ axis, a system of dynamic equilibrium equations, which is solved with respect to the higher derivatives, is proposed (7):

$$\begin{cases} M\frac{d^2Z_1}{dt^2} = C_n(Z_2 - Z_1) + K_n(\dot{Z}_1 - \dot{Z}_2) - M\cdot g \\ m\frac{d^2Z_2}{dt^2} = C_n(Z_1 - Z_2) - C_h(Z_2 - Z_3) - K_n(\dot{Z}_2 - \dot{Z}_1) + K_h(\dot{Z}_3 - \dot{Z}_2) - m\cdot g \\ m_c\frac{d^2Z_3}{dt^2} = C_h(Z_2 - Z_3) - K_h(\dot{Z}_3 - \dot{Z}_2) - C_c\cdot Z_3 + K_c\cdot \dot{Z}_3 - m\cdot g \end{cases} \tag{7}$$

where: K_n, K_h, K_c—suspension damping coefficients, respectively, suspension, tire and tester [Ns/m], C_n, C_h, C_c—stiffness of, respectively, suspension, tire and tester [N/m], Z_1, Z_2, Z_3—coordinates of the vertical movement, respectively, of the sprung mass (M), the unsprung mass (m) of the tester plate and the mass (m_c) [m].

Static deflection of the elastic elements of the suspension, tires and stiff plate were calculated relatively to the steady state of car mass according to Hooke's law using the following formulas (8):

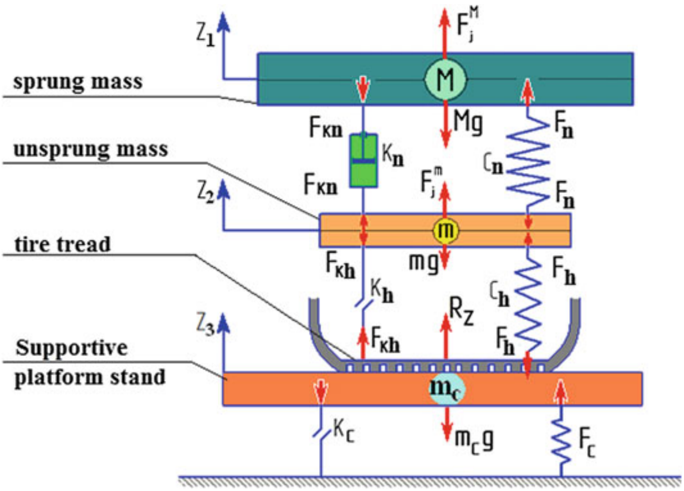


Fig. 3 Scheme to determine the normal reaction of wheels R_z during braking on flat tester

$$\Delta l_n = \frac{M \cdot g}{C_n}; \Delta l_k = \frac{(M+m) \cdot g}{C_h}; \Delta\, l_c = \frac{(M+m+m_c) \cdot g}{C_c} \tag{8}$$

Normal reaction R_z of supporting plate is determined by the formula (9):

$$R_z = K_c \cdot \dot{Z}_3 - C_c \cdot Z_3 \tag{9}$$

Numerical study was based on solutions of Eqs. (1)–(9) is performed together with the equations of the braking process of a car wheel [4]. As a result, mathematical model of the braking wheel process beginning from run onto tester is obtained and suitable graphs of two trials are shown in Fig. 4.

The graphs show that the longitudinal oscillations of the sprung and unsprung mass of the car, as well as areas of the tester having largest amplitude. They make significant changes in the dynamics of a vehicle wheel braking. This process can be divided into two stages: 1st phase is when the wheel runs onto plate of a tester and 2nd phase while blocking the wheel.

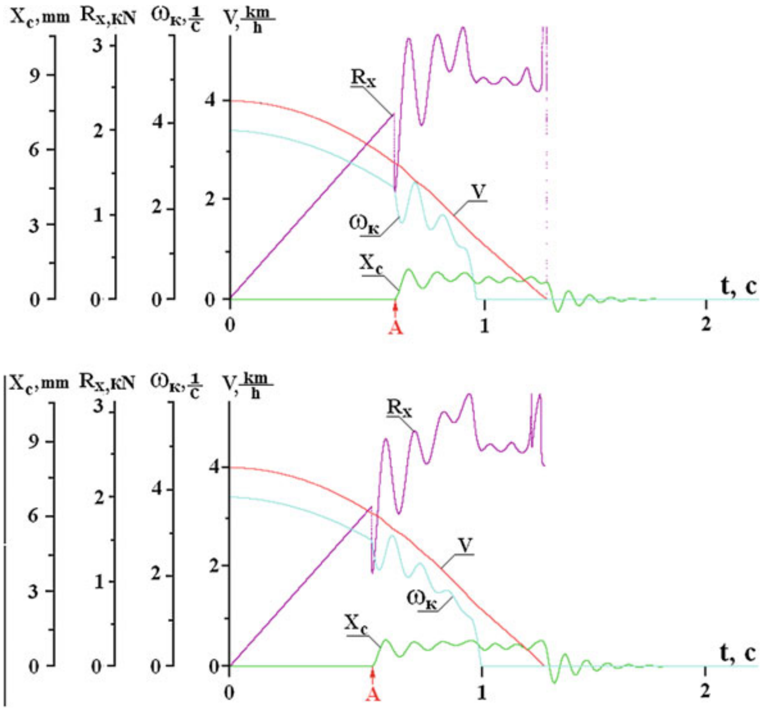


Fig. 4 Graphics car wheel braking process with collisions on a plate tester at point A

3.1 1st Phase—Wheel Runs onto Plate of a Tester

At the moment of running a wheel onto plate of a tester (Fig. 4, point A), under the influence of the braking force, the longitudinal displacement of the plate occurs and subsequent damped oscillations excite longitudinal vibrations of the tire, as well as fluctuations in the sprung and unsprung mass of the vehicle. The graph clearly shows the braking force of the amplitude of these oscillations. Calculated error at this stage according to formula (10) is 51 ÷ 57 %.

$$\delta = \frac{\Delta F_t}{F_{tmax}} \cdot 100\,\% \tag{10}$$

where: F_{tmax}—the maximum braking force,
ΔF_{t}—oscillation amplitude of the braking force.

3.2 2nd Phase—Blocking the Wheel

At the time of wheel blocking ($\omega_{\mathrm{k}} = 0$), the wheel loses contact between tyre and a plate of the tester and reduced frictional properties of the tire are observed. This leads to a new cycle of oscillation, which is accompanied by periodic joining and detaching of both surfaces. Moreover, when decelerating the vehicle, this process is accompanied by the resonance, which causes an increase of oscillation amplitude of the braking force. Margin of error when locking a wheel is 24 ÷ 29 %.

4 Experimental Study of Fluctuations of Braking Properties on Flat Tester

4.1 Analysis of Initial Vehicle Speed Influence on Measurement Error

Experimental investigation of the influence of parameters of the car and tester by the uncertainty of measurement of brake forces was performed in the diagnostic laboratory of the Transport Department of Irkutsk State Technical University.

Firstly, initial values of vehicle speed V were varied in the range of 2–20 m/s (Fig. 5).

It is seen that with increasing initial speed of braking vehicle V, braking force measurement error is significantly reduced by the logarithmic dependence. This is due to the increase of the inertial moment of braking wheel at higher speed.

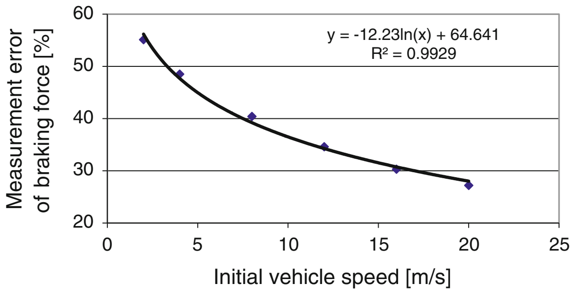


Fig. 5 Measurement error of the braking force in function of car speed

Therefore, to reduce the measurement error of the braking force initial deceleration rate should be increased. But in case of braking flat tester it is very difficult to stop the car just on the tester with limited length (in inertial testers). On power flat testers high speed is technically difficult to implement.

4.2 Analysis of Breaking Time Influence on Measurement Error

It is known that the brake pedal pressing takes some time before the wheel comes over to the tester plate. Therefore, the next phase of the study was the analysis of measurement error depending on the magnitude of the braking force interval t_n from the beginning of braking before the wheel is touching the plate (Fig. 6).

Test was conducted for the initial speed of $V = 4$ m/s and the rate of braking torque increase is 1497 N m/s.

The results indicate that with increasing time t_n braking force measurement error first increases, then reaches an extremum, and then decreases. This dependence is expressed by a quadratic form of a parabola.

The initial growth of the error is due to the fact that for a small braking time the braking force is not large and its reaction impact on the area of the tester is also not

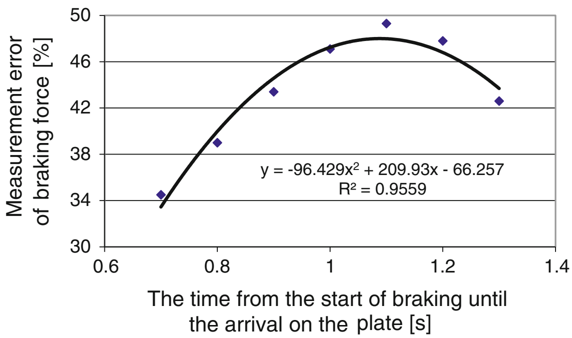


Fig. 6 Graph of the measurement error of the braking force at time t_n varying from the start of braking until hitting the tester plate

very large. Consequently, the oscillation amplitude of the braking force is relatively small. With increasing time t_n braking force measurement error increases as the braking force increases. Extremum occurs at a time when run onto the plate coincides with the wheel lock. At this point, there are the greatest oscillations of braking force amplitudes.

4.3 Analysis of Breaking Force Influence on Measurement Error

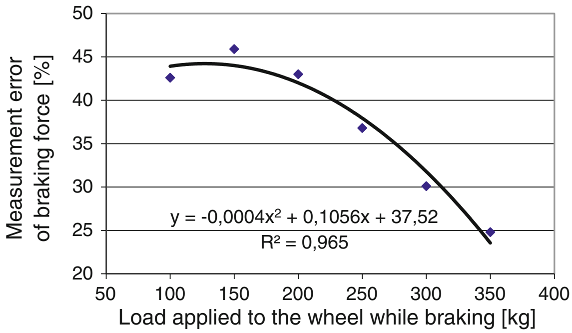
To analyse an impact of load applied to the wheel it is working-out an application simulated brake force in function of mass weight. Figure 7 shows the results of the analysis of impact of that load on the wheel R_z on braking force measurement error.

The graph in Fig. 7 discovers that with increasing load on the braking wheel braking force measurement error is significantly reduced by the law of a quadratic function. This is explained by the fact that with increasing load reduced is amplitude oscillation in the elements of the car and tester. The reason for this is that, firstly, load on tester is increasing and secondly, contact of the tire with the surface of the tester becomes stable.

4.4 Analysis of Plate Mass Influence on Measurement Error

Another element influencing brake force and measurement error is a mass of the plate. Weight of tester plate is made of steel and has a significant impact on the accuracy of the braking force measurement. As the results of the study (Fig. 8) with increase in mass of the plate from 5 to 40 kg the measurement error of the braking force decreases more than twice.

Fig. 7 Graph of the measurement error of the in function of load applied to the wheel while braking



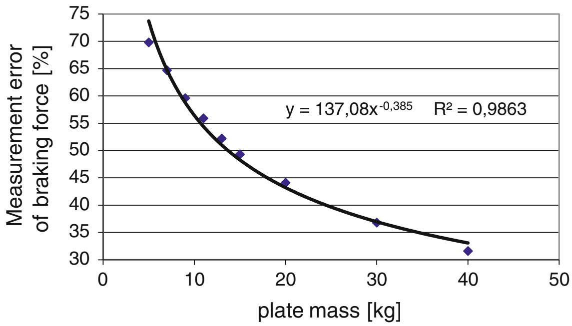


Fig. 8 Graph of the measurement error of the braking force of the weight of the tester site

4.5 Analysis of the Error of Braking Force Lateral Displacement Influence on Measurement Error

Figure 9 shows studies on dependence of measurement error of the braking force of the wheel due to relative lateral displacement of the longitudinal axis of symmetry of the tester. Increasing lateral displacement of the wheel relatively to the axis of symmetry of the tester increases measurement error of the braking force. As noted earlier (1–3), lateral displacement affects the accuracy of measurements of the braking force due to friction forces caused by moment twisting a plate. In lateral displacement, effect of disturbing vibrations only increases the error, but not because of growth of the amplitude fluctuations but by reducing (by the amount of friction force) the maximum value of the braking force.

4.6 Analysis of Influence of Tire Dumping Properties on Measurement Error

It is logical that in the conditions of mass oscillations of the car tires and pads on the margin of error of the braking force measurement affect the damping properties of the tire. This clearly demonstrates the resulting graph in Fig. 10.

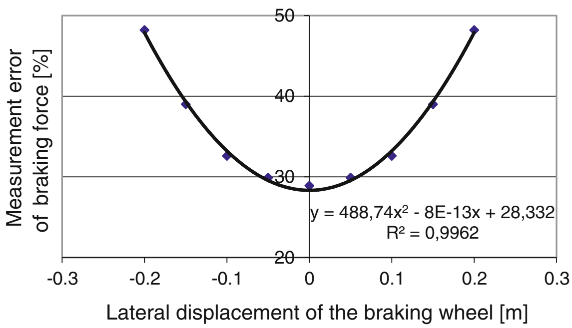


Fig. 9 Measurement error of the braking force of the lateral displacement of the wheel relative to the longitudinal axis of symmetry of the tester plate

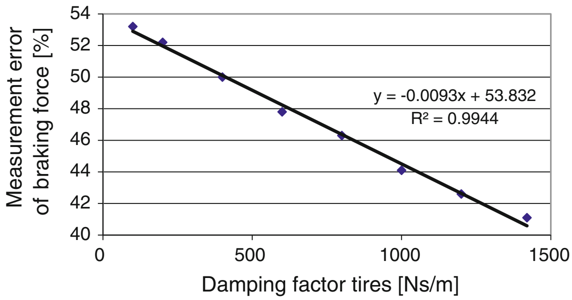


Fig. 10 Graph of the measurement error of the braking force of the damping properties of tires

5 Conclusions

Numerical simulation presented in the paper shows great advantage of that tool in dynamic process analysis. Proposed dynamic model of the system: wheel-flat plat tester is validated. Simulation results reasonably confirm doubts concerning measurements of braking force using simple testers not equipped with advanced compensating sensors.

Obtained simulation results strongly suggest that besides some benefits of the flat brake testers, they have number of drawbacks. The most significant of which are:

- longitudinal vibrations of tester plate cause disruptions of the contact between the wheel and tester plate at the moment of running onto tester plate as well as at the time of blocking the wheel,
- high complexity of braking wheels positioning into the centre of the tester plate and in consequence arising moments rotating tester plate,
- instability of test performance (variability of force and speed of pressing the brake pedal).

Collectively, all above reasons cause methodological mistake of the brake forces measurement on flat testers resulting the measurement error exceeding 50 % or more [1, 4].

This is not an exhaustive list of problems to be overcome in the operation of flat (plate) brake testers. Therefore, in some countries they are banned as a means of control of the brake systems.

The comparison of the metrological properties of flat brake testers with similar properties of brake roller dynamometers [2, 6, 7] shows a clear advantage of the latter.

Presented simulation analysis and experimental research confirm usability of the obtained results in safety analysis of dynamic systems.

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