PROCEEDINGS

53. IWK

Internationales Wissenschaftliches Kolloquium
International Scientific Colloquium

Faculty of
Mechanical Engineering

PROSPECTS IN MECHANICAL ENGINEERING

8 - 12 September 2008

www.tu-ilmenau.de

TECHNISCHE UNIVERSITÄT
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Published by
Impressum

Publisher Herausgeber	Der Rektor der Technischen Universität Ilmenau Univ.-Prof. Dr. rer. nat. habil. Dr. h. c. Prof. h. c. Peter Scharff
Editor Redaktion	Referat Marketing und Studentische Angelegenheiten Andrea Schneider
	Fakultät für Maschinenbau Univ.-Prof. Dr.-Ing. habil. Peter Kurz, Univ.-Prof. Dr.-Ing. habil. Rainer Grünwald, Univ.-Prof. Dr.-Ing. habil. Prof. h. c. Dr. h. c. mult. Gerd Jäger, Dr.-Ing Beate Schlütter, Dipl.-Ing. Silke Stauche
Editorial Deadline Redaktionsschluss	17. August 2008
Publishing House Verlag	Verlag ISLE, Betriebsstätte des ISLE e.V. Werner-von-Siemens-Str. 16, 98693 Ilmenau

CD-ROM-Version:

Implementation Realisierung	Technische Universität Ilmenau Christian Weigel, Helge Drumm
Production Herstellung	CDA Datenträger Albrechts GmbH, 98529 Suhl/Albrechts

ISBN: 978-3-938843-40-6 (CD-ROM-Version)

Online-Version:

Implementation Realisierung	Universitätsbibliothek Ilmenau ilmedia Postfach 10 05 65 98684 Ilmenau

Home / Index:
http://www.db-thueringen.de/servlets/DocumentServlet?id=17534

53rd Internationales Wissenschaftliches Kolloquium
Technische Universitat Ilmenau
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A. Pogosian / A. Bakhshyan

Vibrostability Calculation of Disk Brake in Automobile Anti-Lock Braking System

ENGINEERING DESIGN

To ensure controllability and safety in the braking process automobile is equipped with anti-lock braking system (ABS), which prevents wheel skidding relative to road by changing brake pad pressure. Anti-lock braking system provides maximal deceleration and reduces automobile brake distance. Improving automobile braking characteristics anti-lock braking system simultaneously increase brake loading, which impact on reliability of the brake operating [1]. One of the automobile brake device significant characteristics is the vibrostability, which influences on smooth motion and stable operation of brake. Conditioned by brake disk unevenness friction oscillations originated in the brake devices adversely impact on brake vibrostability and hinder the reliable functioning.

Essential influence on oscillations aroused in the automobile brake device at braking render not only the forces affecting in the brake device, but also the forces acting on the whole automobile. For this reason to calculate the brake device vibrostability the calculation model of the whole automobile, which describes the movement of automobile on plane surface during braking is chosen, taking into account the dynamic load transfer on wheels (fig. 1).

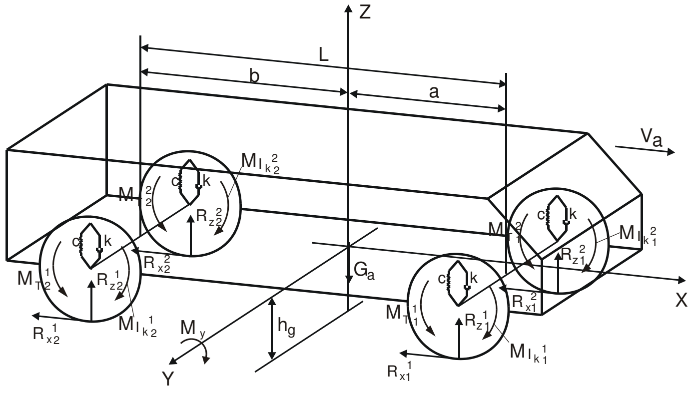


Fig. 1. Calculation model of automobile braking

Neglecting the rolling and air resistance forces that do not exceed 3-5 % in the common force balance, the equation of the automobile gravity centre movement is described by the following [2]:

$$M_a j_x = \sum_{i=1}^{2}\sum_{k=1}^{2} R_{xk}^{i} , \tag{1}$$

where M_a-the mass of the automobile, j_x-the automobile deceleration, R_{xk}-tangent reaction of the wheel to road engagement, k=1-front axle, the k=2-back axle, i=1-left wheel, i=2-right wheel.

Automobile gravity centre movement longitudinally axis z, which is perpendicular to the road surface is defined:

$$M_a j_z = \sum_{i=1}^{2}\sum_{k=1}^{2} R_{zk}^{i} - M_a g , \tag{2}$$

where j_z-the automobile deceleration regarding axis z, R_{zk}-normal reaction force on the automobile wheel, g-acceleration of gravity.

The angle of different is defined by the momental equation with regard to horizontal axis y, which passes through the automobile centre of gravity:

$$I_y \varepsilon_y = \sum_{i=1}^{2}\sum_{k=1}^{2} R_{xk}{}^{i}(h_k - z) - \left(R_{z1}{}^{1} + R_{z1}{}^{2}\right)a + \left(R_{z2}{}^{1} + R_{z2}{}^{2}\right)b , \tag{3}$$

where I_y-the automobile moment of inertia relative to axis y, ε_y- automobile angular deceleration relative to axis y, h_k-height of the automobile gravity centre, b-distance from the automobile gravity centre to forward and back axes accordingly.

The angular speed of a wheel is defined from a condition of forces balance:

$$I_к \frac{\partial \omega_к}{\partial t} = M_T - R_x r_? , \tag{4}$$

where I_k-the wheel moment of inertia, M_T-brake moment.

For the calculation of the automobile brake device vibrostability the oscillatory model of disc brakes [3] which represents a rotating disk with reduced moment of inertia I, on which the pads with mass m is considered as a solid elastically contacting with the disk and interacting with the motionless frame of brake device by elastic link simulated as a spring (fig. 2).

The oscillatory process of the automobile disc brake device is described by a differential equations system as follows:

$$\begin{cases} I_k \cdot \ddot{\varphi} = -T_f \\ m \cdot \ddot{x} = -c \cdot x + F_f , \end{cases} \tag{5}$$

where T_f and F_f -brake torque and friction force in coupling, cx –elastic link force between pad and frame, $\ddot{x}$-acceleration of the pad, $\ddot{\varphi}$-angle acceleration of the brake disk.

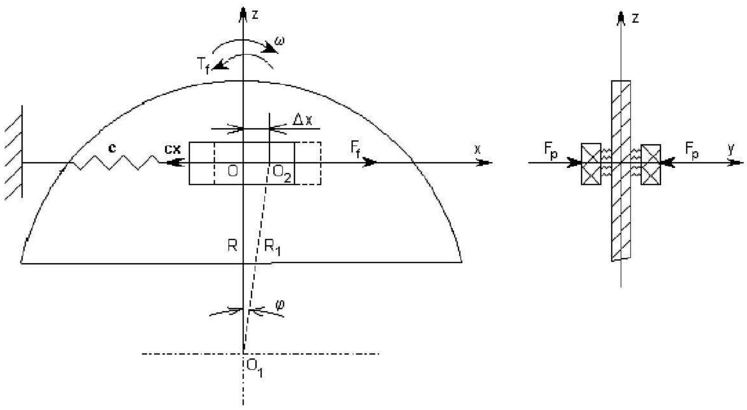


Fig. 2. Calculation model of disk-pad brake oscillatory process

Acting pressure on brake pad is assumed as a constant in the given system of the equations (5), i.e. the influence of anti-lock braking system is not taken into account, the basic purpose of which is to prevent the wheel blocking by means of changing the pressure. In this connection the equations (1)-(5) are systematized, with condition of wheel to road coupling force and brake disk angular speed interdependency, to investigate the oscillations aroused in brake device during braking. The considered model of brake dynamics establish interaction between braking and tire-road coupling moments:

$$\begin{cases} M_a \ddot{X} = R_{x1}^{1} + R_{x2}^{1} + R_{x1}^{2} + R_{x2}^{2} \\ M_a \ddot{z} = R_{z1}^{1} + R_{z2}^{1} + R_{z1}^{2} + R_{z2}^{2} - M_a g \\ I_y \ddot{\varphi}_y = \left(R_{x1}^{1} + R_{x2}^{1} + R_{x1}^{2} + R_{x2}^{2}\right)\left(h_k - z\right) - \left(R_{z1}^{1} + R_{z2}^{1}\right)a + \left(R_{z1}^{2} + R_{z2}^{2}\right)b \\ I_k \ddot{\varphi} = T_f - R_x r_? \\ m\ddot{x} = -cx + F_f \end{cases} \tag{6}$$

The solution of the system of equations (6) is carried out by a numerical method with a package of special software MathCAD.

During the braking of the automobile equipped with anti-lock braking system, the amplitude of pad oscillations increases, but high-frequency component of the vibrations decreases. The frequency of the vibrations in the initial moment of braking is varied within the range of 100-200 Hz, and at the final moment reaches up to 1000 Hz. Oscillations of the brake moment are the consequence of the pads pressure adjustment to prevent the wheels blocking by anti-lock braking system, which certainly raises efficiency of automobile braking, reducing its braking distance and increasing deceleration.

The calculation model is capable to research the brakes vibrostability with applying of anti-lock braking system and at exploitation of the automobile in different road conditions.

Appropriate test bench experimental investigations of the disc brake equipped with the simulator of anti-lock braking system are carried out. Investigations are carried out on asbestos-free friction materials such as Bastenit [4]. Calculation values of vibrodisplacements do not exceed the experimental results more than 15 %. Thus curves almost repeat each other during a braking cycle (fig. 3).

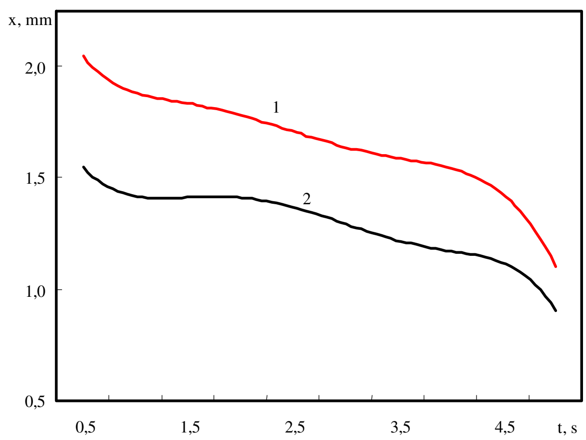
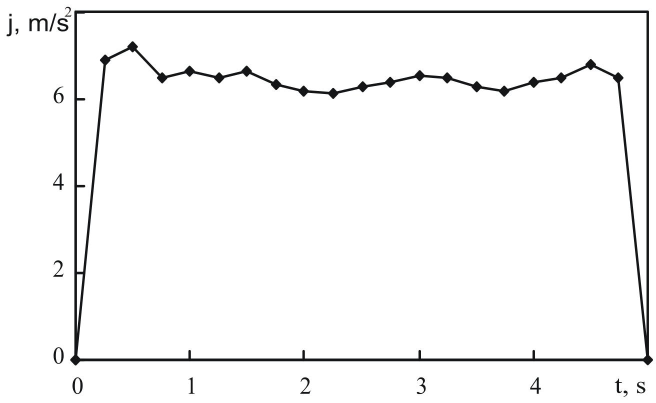
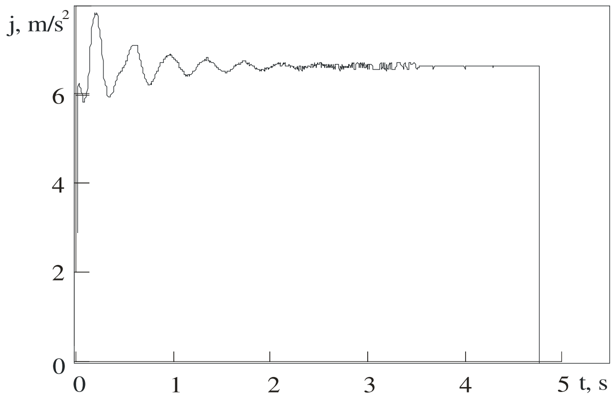


Fig. 3. Analytical (1) and experimental (2) curves of the pad displacements

One of the main characteristic of brake device is deceleration, which experimental and analytical dependences are shown in fig. 4.



a)

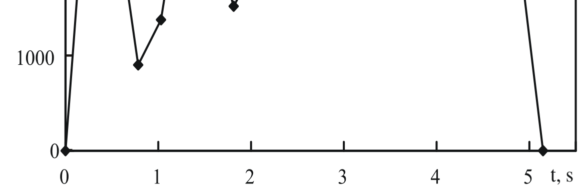


b)

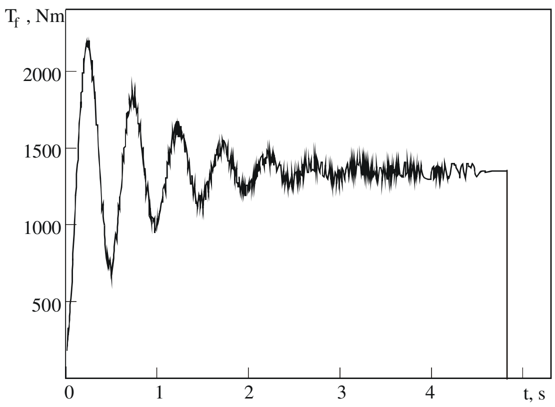
Fig. 4. Experimental (a) and analytical (b) dependences of deceleration upon time

The maximum divergence of experimental and analytical values of deceleration does not exceed 13%.

Experimental and analytical dependences of braking moment upon time are shown in fig. 5. Oscillations of braking moment during whole experiment are conditioned by working of anti-lock braking device, which adjusts the pressure on braking pad by changing it. The divergence of experimental and analytical values is conditioned by functioning of hydraulic drive of braking system and does not exceed 10%.



a)



b)

Fig. 5. Experimental (a) and analytical (b) dependences of braking moment upon time

The analysis of investigations results shows that the frictional oscillations aroused in brake devices, equipped with anti-lock braking system, differ from conventional brake devices oscillations, the amplitude of which is up to 4 times exceed the amplitudes of conventional brake devices oscillations. Simultaneously the high frequency component of the oscillations is less up to 3 times, thus the highest frequency appeared at the stopping moment of the brake disk. The calculation data of the brake pad vibrodisplacements magnitude with anti-lock braking system are 20 % higher.

The mathematical model sufficiently represents the dynamics of the oscillations during braking and the divergence of the calculation data with the experimental results does not exceed 15 %. It enables to investigate dynamics of the oscillatory spectrum during braking under different road conditions. The calculation method can be used for rating the vibrostability of both developed and existing brakes. In the result the existing brake devices structure can be modified and new brake materials with appropriate tribological characteristics developed.

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