ELASTIC DEFORMATION OF DISC BRAKE FRICTION LINING

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Abstract. There are the analysis and estimation of disc brake friction linings deformation value and impact on the friction body (friction lining on steel base) loading given in this article. It allows to modulate the wear of friction lining in the future research. Such friction body construction was analyzed in this research – the friction body and brake lever are linked with a pin, the direction of this pin is perpendicular regarding the vector of the slipping velocity. The research is significant to the joint-stock company “Jelgava engineering plant” (A/S “Jelgavas masinbuves rupnica”) – they produce brakes with such construction. The theoretical research was done using two methods – solving differential equation for balk with finite length on elastic base (Winkler’s method) and the method of finite elements (FEM). After solving the differential equation for balk with finite length on elastic base the author found equations for vertical displacement *y*, turning angle θ, specific surface pressure *p*, bending moment *M* and transversal force Q_y of balk in any cross section. The author used the second calculation method (FEM) to test and evaluate the accuracy of the first method (Winkler’s method). Calculation with FEM was carried out by computer using software “Cosmosworks”, before it the 3 dimensional friction body model was built in computer software “Solidworks”.

Key words: disc brake, friction lining, ellastic deformation, friction pair, loading.

Introduction

The aim of the research is to analyze and estimate the disc brake friction linings deformation value and impact on the friction body (friction lining on steel base) loading to modulate the wear of friction lining in the future research.

The friction lining (Fig. 1) deforms under resultant pressing force *R* and the distribution of the specific surface pressure on friction surface is uneven, it causes the bending moment applied to the steel base, too.

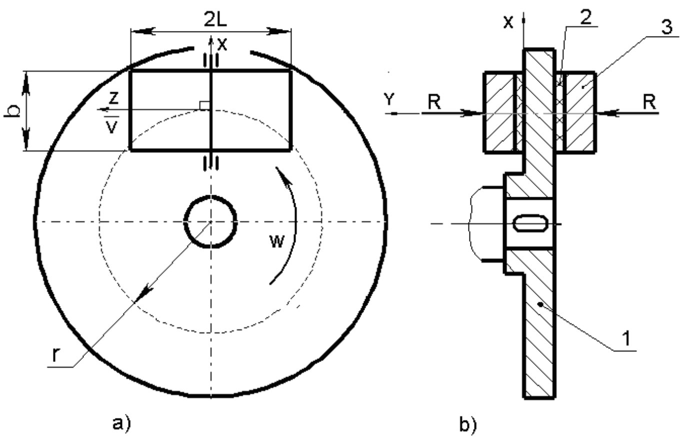

Fig. 1. **Friction pair:** 1 – disc, 2 – friction lining, 3 – steel base

The wear depends on the material, specific surface pressure (loading), friction coefficient, slipping velocity and temperature [1].

Such friction body construction was analyzed in this research – the friction body and brake lever are linked with a pin, the direction of this pin is perpendicular regarding to the vector of the slipping velocity $\bar{v}$ (Fig. 2). The research is significant to the joint-stock company “Jelgava engineering plant” (A/S “Jelgavas masinbuves rupnica”) – they produce brakes with such construction.

Theoretical research was done using two methods – solving differential equation for balk with finite length on elastic base (Winkler’s method) and the method of finite elements (FEM).

The wear of the friction lining depends, mainly, on the specific surface pressure *p*(*x,z,t*) and sliping velocity *v(x,z,t)* between the disc and the friction lining (Fig. 1). Usually the speed of the wear dw/dt is proportional to the specific surface pressure *p* [2]:

$$\frac{dw}{dt} = K'_W \, p^{\alpha}(x,z,t) v^{\beta}(x,z,t) \,, \tag{1}$$

The linear wear $w\ (x,z,t)$ in direction y:

$$w(x,z,t) = K_W \int_0^t p^{\alpha}(x,z,\tilde{t}) v^{\beta}(x,z,\tilde{\tau}) d\tau \,, \tag{2}$$

where K_W - the coefficient of the wear resistant and work regime;
τ - the moment of time;
α and b - eksponent ($\alpha \geq 1 ; b \approx 1$);
p – specific surface pressure;
v – sliping velocity;
x, y, z, t – coordinates of space and time.

There is information in the technical literature [6, 7, 8, 9] about researches of friction pair wear and temperature depending on the regime of work, physical and mechanical material properties. There is no information about friction pair deformation impact on the wear. The accuracy of friction pair wear modeling depends on elastic deformations caused by pressing force R.

Evaluating elastic deformations in friction pairs it is possible to explain the reasons of refuse of disc brakes, couplings, sealings and other friction mates.

Methods

Theoretical research was done using two methods – solving differential equation for balk with finite length on elastic base (Winkler's method) and the method of finite elements (FEM). It is assumed that the steel base of friction lining is a balk and the friction lining is an elastic base (Fig. 1), deformation of steel disc is ignored. In conformity with hypothesis of Winkler'sthere are proportionality between the normal reaction force and displacement (deformation) in any point of joint [3]. The deformation of steel base transfer on the friction lining – that is why the method is adopted to solve this problem.

The differential equation for balk with finite length on elastic base [4]:

$$EI_x y'' = M \ . \tag{3}$$

After modifying:

$$EI_x y^{(IV)} = M'' = q \,, \tag{4}$$

where q – intensity of normal reaction force (Fig. 2., b).

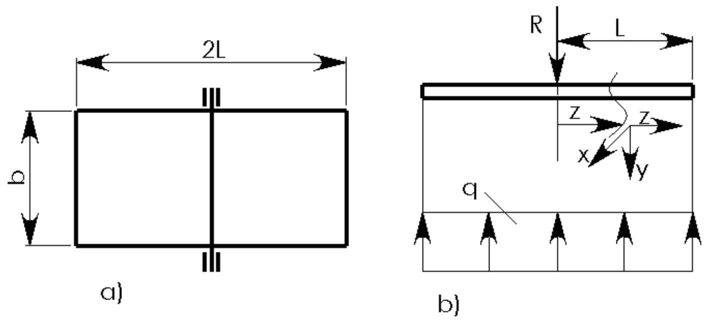

Fig. 2. **The balk on elastic base (steel base on friction lining):** a – steel base of friction lining; b – loading schema of steel base

The intensity of normal reaction force:

$$q = -\frac{c \cdot y}{2L} \,, \tag{5}$$

where c – the stiffness of friction lining;
$2L$ – the length of friction lining;
y – displacement in direction y.

After equation (5) insert in (4):

$$y^{(IV)} + \frac{c \cdot y}{2L \cdot E \cdot I_x} = 0. \tag{6}$$

It is assumed that

$$\frac{c}{2L \cdot E \cdot I_x} = 4k^4, \tag{7}$$

where k – constant coefficient:

$$k = \sqrt[4]{\frac{c}{L \cdot E \cdot I_x \cdot 8}}. \tag{8}$$

The differential equation for balk with finite length on elastic base is acquired:

$$y^{(IV)} + 4k^4 y = 0. \tag{9}$$

Solving equation (9), we can find displacement y and other derives [5]:

$$y = C_1 \cdot \sin(kz) \cdot sh(kz) + C_2 \cdot \sin(kz) \cdot ch(kz) + C_3 \cdot \cos(kz) \cdot sh(kz) + \\ + C_4 \cdot \cos(kz) \cdot ch(kz); \tag{10}$$

$$y' = (C_2 - C_3) \cdot k \cdot \sin(kz) \cdot sh(kz) + (C_1 - C_4) \cdot k \cdot \sin(kz) \cdot ch(kz) + \\ + (C_1 + C_4) \cdot k \cdot \cos(kz) \cdot sh(kz) + (C_2 + C_3) \cdot k \cdot \cos(kz) \cdot ch(kz); \tag{11}$$

$$y'' = 2C_1 \cdot k^2 \cdot \cos(kz) \cdot ch(kz) + 2C_2 \cdot k^2 \cdot \cos(kz) \cdot sh(kz) - 2C_3 \cdot k^2 \cdot \sin(kz) \cdot ch(kz) - \\ - 2C_4 \cdot k^2 \cdot \sin(kz) \cdot sh(kz); \tag{12}$$

$$y''' = 2(C_2 - C_3) \cdot k^3 \cdot \cos(kz) \cdot ch(kz) + 2(C_1 - C_4) \cdot k^3 \cdot \cos(kz) \cdot sh(kz) - \\ - 2(C_1 + C_4) \cdot k^3 \cdot \sin(kz) \cdot ch(kz) - 2(C_2 + C_3) \cdot k^3 \cdot \sin(kz) \cdot sh(kz). \tag{13}$$

The constants C_1, C_2, C_3 and C_4 are calculated by formulas:

$$C_1 = \frac{R}{8E \cdot I_x \cdot k^3} \cdot \frac{sh^2(kL) + \sin^2(kL)}{sh(kL) \cdot ch(kL) + \sin(kL) \cdot \cos(kL)}; \tag{14}$$

$$C_2 = -\frac{R}{8E \cdot I_x \cdot k^3}; \tag{15}$$

$$C_3 = +\frac{R}{8E \cdot I_x \cdot k^3}; \tag{16}$$

$$C_4 = -\frac{R}{8E \cdot I_x \cdot k^3} \cdot \frac{ch^2(kL) + \cos^2(kL)}{sh(kL) \cdot ch(kL) + \sin(kL) \cdot \cos(kL)}. \tag{17}$$

There are correlations between derives of displacement y and turning angle θ, bending moment M and transversal force Q_y:

$$\theta = y', \tag{18}$$

$$M = E \cdot I_x \cdot y'', \tag{19}$$

$$Q_y = E \cdot I_x \cdot y'''. \tag{20}$$

The correlation between the specific surface pressure p and displacement y:

$$p = \frac{y \cdot c}{2L \cdot b}. \tag{21}$$

Author used the second calculation method (FEM) to test and evaluate accuracy of the first method (Winkler's method). Calculation with FEM was carried out by computer using software "Cosmosworks", before it three dimensional friction body model was built in computer software "Solidworks".

Results

Four kinds of friction linings with different modules of elasticity (E_1 = 50 GPa, E_2 = 100 GPa, E_3 = 150 GPa, E_4 = 200 GPa) were analyzed using two methods (Winkler's and FEM). The value of resultant pressing force R = 10 kN, length of friction lining $2L$ = 200 mm, width – b = 100 mm and thickness – h = 10 mm. The modules of elasticity of steel base E = 200 GPa. The assumed values correspondent to the actual dimensions and modules of elasticity of friction linings.

The results prove that maximal values of the deformations y (Fig. 3, a, b), the specific surface pressure p (Fig. 5, a, b) and bending moment (Fig. 4) are located under resultant pressing force R.

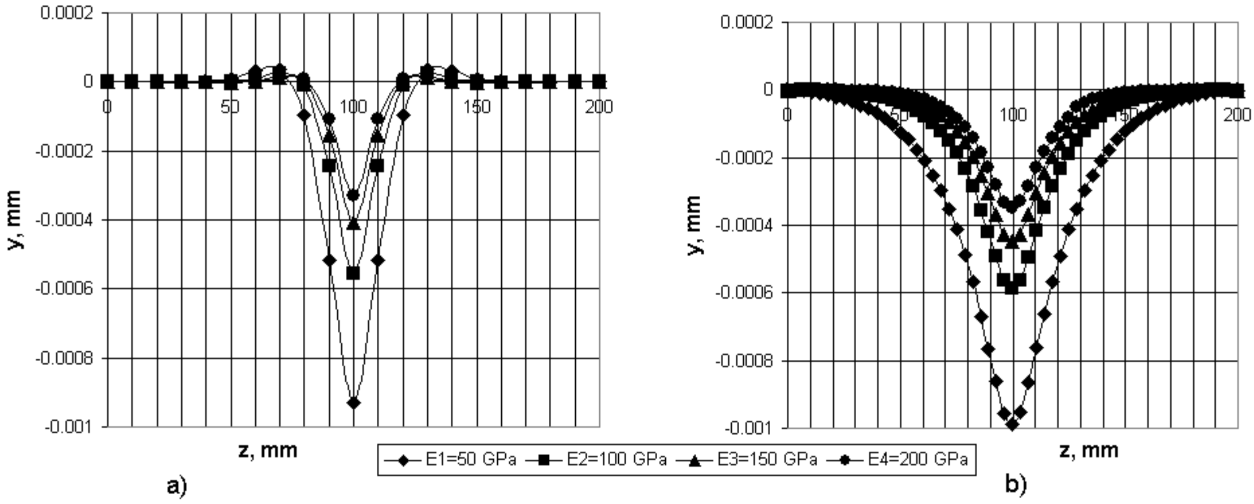

Fig. 3. **The elastic deformations *y* of friction lining depending on resultant pressing force *R* in different split *z* of friction lining:** a – using Winkler's method; b – using finite element method (FEM)

There are no linear correlation between module of elasticity E and the maximal deformation y_{max} of friction lining (Fig. 7). The maximal deformation reduces 2.8 times, if module of elasticity increase 4 times.

There is linear correlation between module of elasticity E and the maximal pressure p_{max} of friction lining (Fig. 8). The maximal pressure increase 1.4 times, if module of elasticity increase 4 times.

The bending moment (Fig. 4), caused by elastic deformations of friction lining, is too small to generate notable tension in steel base with a thickness 10 mm. Depending on module of the friction lining elasticity the maximal value of bending moment reach 18 ... 27 Nm.

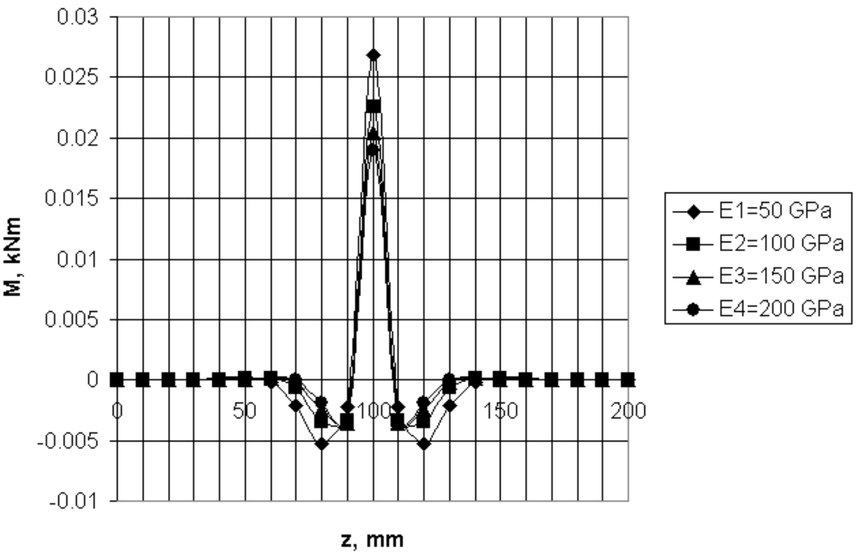

Fig. 4. **The bending moment *M* (applied to the steel base) depending on resultant pressing force *R* in different split *z* of steel base:** a – using Winkler's method; b – using FEM

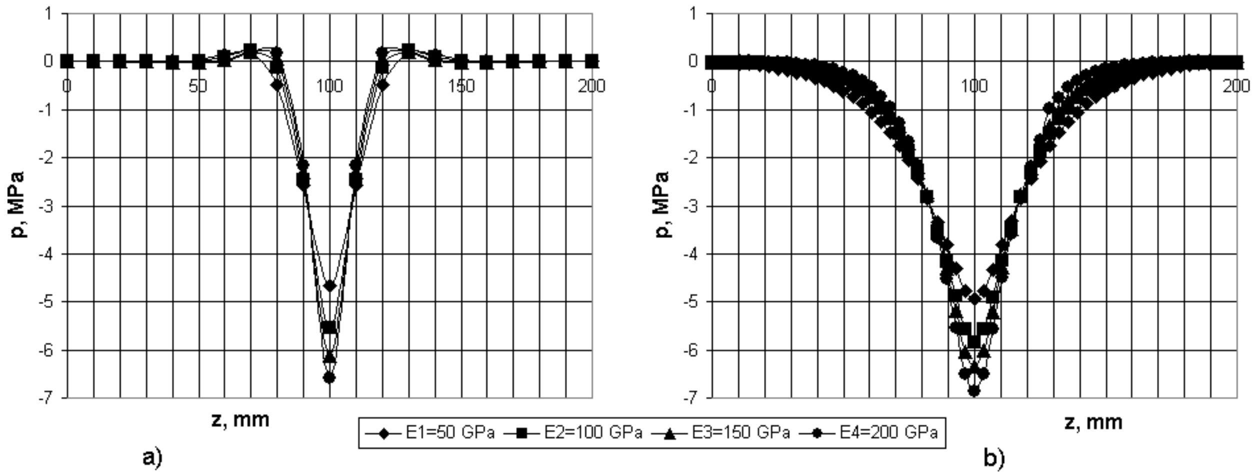

Fig. 5. **The specific surface pressure *p* depending on resultant pressing force *R* in different split *z* of friction lining:** a – using Winkler's method; b – using finite element method (FEM)

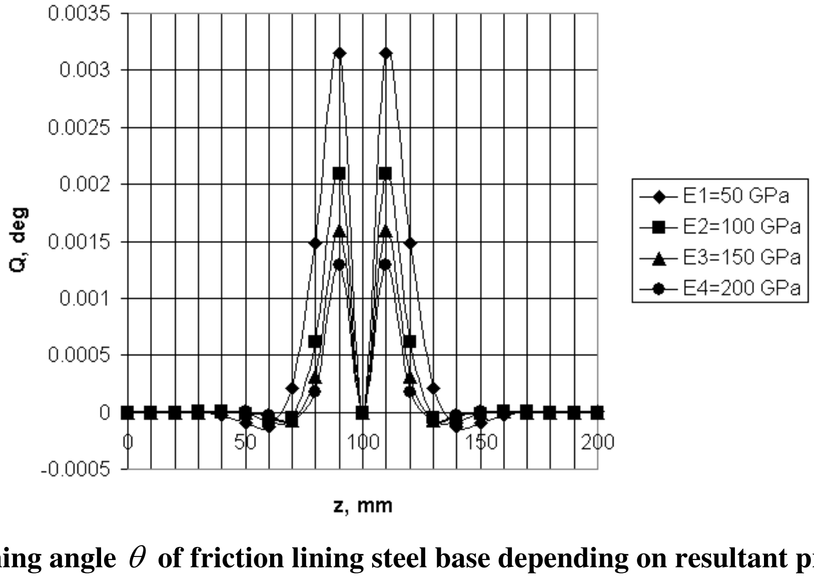

Fig. 6. **The turning angle θ of friction lining steel base depending on resultant pressing force *R* in different split *z* of steel base:** a – using Winkler's method; b – using FEM

The turning angle θ of friction lining steel base is zero under resultant pressing force *R* (Fig. 6), but the maximal value of turning angle (0.0013 … 0.0032 deg.) depends on module of friction lining elasticity.

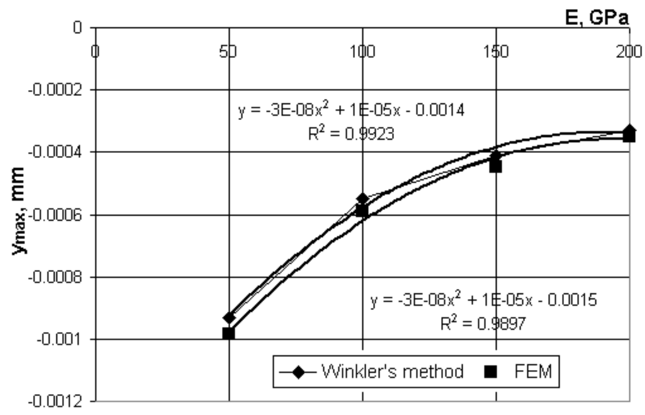

Fig. 7. **The maximal elastic deformations y_{max} depending on model of friction lining elasticity *E***

Both calculation methods give similar results (Fig. 2, 5), the difference fit in 5 % (Fig. 7, 8) – that testify validity of these results.

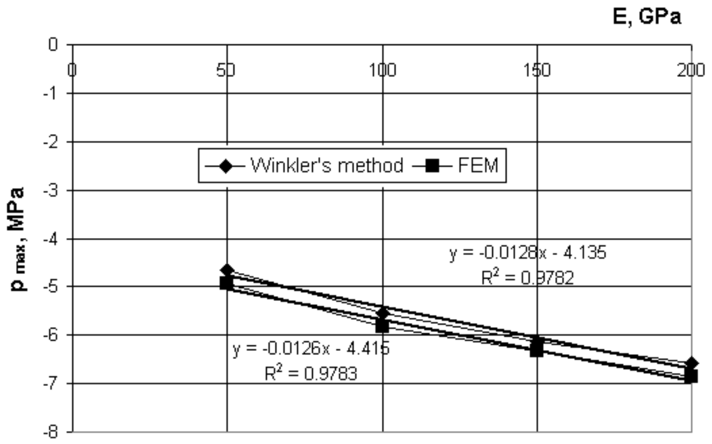

Fig. 8. **The maximal specific surface pressure p_{max} depending on the module of friction lining elasticity *E***

Conclusions

1. The bending moment, caused by elastic deformations of friction lining, is too small to generate notable tension in steel base with a thickness 10 mm.
2. Both calculation methods give similar results, the difference fit in 5 %.
3. There are no linear correlation between module of elasticity *E* and the maximal deformation y_{max} of friction lining. The maximal deformation reduces 2.8 times, if module of elasticity increases 4 times.
4. There are linear correlation between the module of elasticity *E* and the maximal pressure p_{max} of friction lining. The maximal pressure increase 1.4 times, if module of elasticity increases 4 times.
5. The maximal values of the deformations *y*, the specific surface pressure *p* and bending moment are located under resultant pressing force *R* – it contribute uneven wear of friction lining during braking process.
6. It is predictable that the wear of friction lining will reduce if the value of maximal specific surface pressure was lower - friction linings with smaller module of elasticity are recommended from this position.

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