VIRTUAL SIMULATION OF A GUIDING SYSTEM FOR ECCENTRICALLY LOADED SLIDE WAYS

Slobodan Mitrović
(University of Kragujevac, Faculty of Mechanical Engineering, Sestre Janjic 6, 34000 Kragujevac, Serbia & Montenegro, e-mail: boban@kg.ac.yu)

ABSTRACTS

Proper selection of slide ways of machine tools has the exceptional influence on machining precision and quality of the machine surface. All errors that arose as a consequence of clearance between the sliding surfaces or deformations in the contact zone directly affect the machining error.

In order to ensure mobility of slide ways, it is necessary that a clearance exist in sliding joints. Under the influence of eccentric load the twisting of the cross section of the slide way occurs, which can lead to appearance of self-braking, namely the blocking of slider's motion along the slide ways. To prevent that it is necessary that the length of a slider is as large as possible with respect to its width. Since the increase in size of the stand also leads to increase of the slide ways, the whole machine, and by that also its weight and price, that solution of this problem requires more complex analysis, especially from the tribological aspect.

This paper presents an attempt of solving the undesired effect of slide ways self-braking. Through theoretical analyses.

Keywords: *guiding, slide ways, eccentric load*

1. INTRODUCTION

Machine tools, according to their presence in industry, represent a significant group of complex technical systems. The most frequent failure to achieve accuracy and quality of the machined surface is a direct or indirect consequence of contact surfaces wear of the contact surfaces of the TMS guiding elements. Due to wear clearances appear between those elements, which then lead to deviations of real from the necessary trajectories of motion. This directly affects the accuracy of machining.

From the aforementioned reasons in all the machine tools the guiding systems have special importance.

In machine tools are significantly more presents slide ways, with respect to rolling ones. Present development in the area of machine tools does not point to a significant change in this ratio, except in some machines of high accuracy class.

The most frequently slide ways are exposed to eccentric loads.

Load of slide ways due to action of driving forces, is characteristics at universal lathe, different types of capstan lathe, eccentrically loaded broaching machine and other machine tools. With this type of guiding of the slider and loads, the so-called sealed guides, it is necessary to adequately select geometrical parameters of slide ways.

2. ANALYSIS OF THE STRESS STATE

The proper selection of slide ways of machine tools has exceptional importance for accuracy of machining and quality of the machined surface. All the faults that appear as a consequence of clearances between the sliding surfaces or deformations in the contact zone directly affect the machining error. Related to that, the adequate analysis of the stress state on the slide ways is of a special importance.

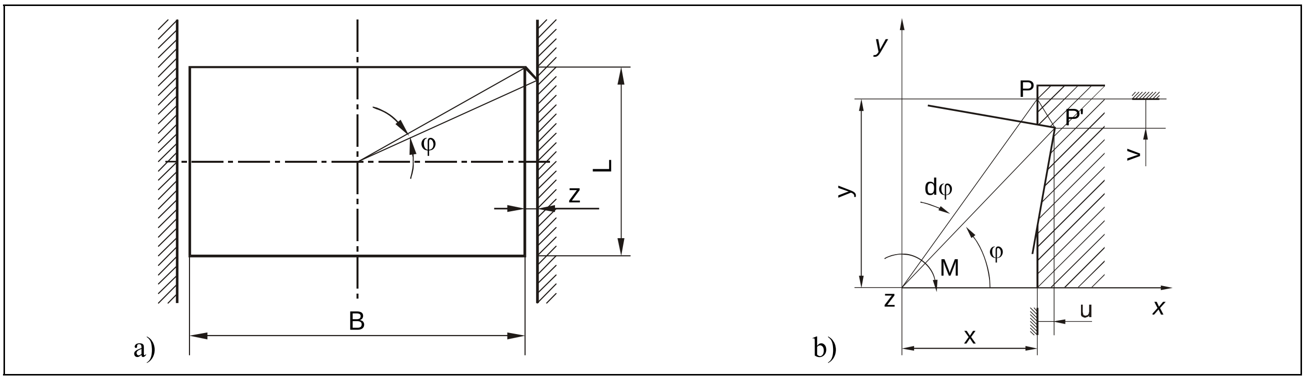

Figure 1. *Rotation of slide ways due to action of the torque load*

To ensure motion of the slide ways, it is necessary that clearance exist in sliding joints. Under the action of the eccentric (torque) load, the rotation occurs of the cross-section of the slide ways for a certain angle φ, Figure 1a. The total angle of free rotation of the cross-section can be expressed according to relation

$$\varphi = \arccos \frac{\frac{B}{2}}{\sqrt{\left(\frac{B}{2}\right)^2 + \left(\frac{L}{2}\right)^2}} - \arccos \frac{\frac{B}{2} + z}{\sqrt{\left(\frac{B}{2}\right)^2 + \left(\frac{L}{2}\right)^2}} .$$

Through analysis of this expression one comes to indicators which relate the influence of the ratio of the slide ways sizes (B/L) and clearance magnitude, to magnitude of the free rotation angle φ. Results of the analysis, which is not presented here, point to increase of the angle φ with increase of the B/L ratio within the frame of the constant clearance. It was also shown that for the constant ratio B/L, when the clearance is increased, the angle of free rotation increases. When the angle φ reaches the critical value φ_{cr}, the self-braking of slide ways occurs.

Due to action of the torque load elastic displacement occurs of points P and P' (Figure 1b), in material in the vicinity of the contact of the slider with the guiding elements. In Figure 1b is presented the position of the sliding pair slide ways-slider, subjected to torque load M.

Stresses on the contact surfaces represent the complex functions of the material characteristics and coordinates x and y. Starting from the assumption that elementary displacements in the contact zone occur on the circular paths, one can find, through the theory of elasticity, to the stress distribution over the contact surfaces.

Since stresses in the direction of the z axis are negligibly small, the state of plane strain will be considered of an element of the unit thickness. The value of displacement of the elementary parts along the x and y axes amount to:

$$u = y \cdot d\varphi ; \quad v = -x \cdot d\varphi , \qquad d\varphi = \frac{-y}{x^2 + y^2} \cdot dx + \frac{x}{x^2 + y^2} \cdot dy .$$

Values of strains ε_x i ε_y are determined as derivatives of displacements with respect to respectful coordinates according to expressions:

$$\varepsilon_x = \frac{\partial u}{\partial x} = \frac{y^3 - x^2 y}{\left(x^2 + y^2\right)^2} \cdot dy ; \qquad \varepsilon_y = \frac{\partial v}{\partial y} = \frac{2x^2 y}{\left(x^2 + y^2\right)^2} \cdot dy .$$

The total normal strain is:

$$\varepsilon = \varepsilon_x + \varepsilon_x = \frac{y^3 + x^2 y}{\left(x^2 + y^2\right)^2} \cdot dy ,$$

while the shear strain (sliding) can be determined according to expression:

$$\gamma = \frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} = \frac{x^3 - 3xy^2}{\left(x^2 + y^2\right)^2} \cdot dy .$$

According to Hooke's law the relations can be established between stresses and strains as:

$$\sigma_x = \lambda \cdot \varepsilon + 2G \cdot \varepsilon_x , \qquad \tau_{xy} = G \cdot \gamma_{xy} ,$$

where:

$$\lambda = \frac{\nu \cdot E}{(1+\nu)(1-2\nu)} , \quad G = \frac{E}{2(1+\nu)} .$$

With given expressions are determined the linear stresses as functions of coordinates. After substituting the expression that defines strain, these relations obtain the form:

$$\sigma_x = \frac{\left[(1-\nu)y^3 - x^2 y(1-3\nu)\right] E \cdot dy}{(1+\nu)(1-2\nu)\left(x^2 + y^2\right)^2} , \quad \tau_{xy} = \frac{\left(x^3 - 3xy^2\right) E \cdot dy}{2(1+\nu)\left(x^2 + y^2\right)^2} .$$

Force at the surface $dy \cdot 1$, balances the portion of the torque load M or the whole torque load, what depends on the magnitude of the load itself, Figure 2, so the momentum equation can be established as:

$$\left(\left|\sigma_x \cdot y\right| + \left|\tau_{xy} \cdot x\right|\right) \cdot dy \cdot 1 = M$$

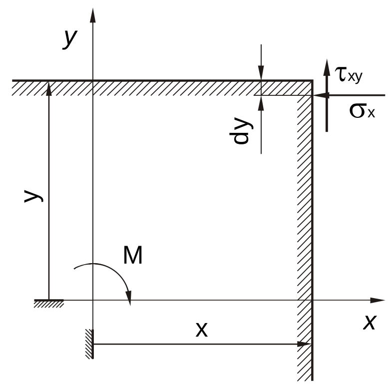

Figure 2. *Normal and shear stresses at the slide ways*

By solving the system of the last three equations, one obtains the final expressions for calculating stresses and the contact surface $dy \cdot 1$ as a function of coordinates and the torque load:

$$1 \cdot dy = \left[\frac{2M(1+\nu)(1-2\nu)}{E}\right]^{\frac{1}{2}} \cdot \left\{\frac{(x^2+y^2)^2}{2y^2\left[\left|(1-\nu)y^2 - x^2(1-3\nu)\right|\right] + x^2(1-2\nu)\left[\left|x^2 - 3y^2\right|\right]}\right\}^{\frac{1}{2}}$$

$$\tau_{xy} = \left[\frac{M \cdot E(1-2\nu)}{2(1+\nu)}\right]^{\frac{1}{2}} \cdot \frac{1}{x^2+y^2} \cdot \left\{\frac{(x^2+y^2)^2}{2y^2\left[\left|(1-\nu)y^2 - x^2(1-3\nu)\right|\right] + x^2(1-2\nu)\left[\left|x^2 - 3y^2\right|\right]}\right\}^{\frac{1}{2}}$$

$$\sigma_x = \left[\frac{2M}{(1+\nu)(1-2\nu)}\right]^{\frac{1}{2}} \cdot \frac{1}{x^2+y^2} \cdot \left\{\frac{(x^2+y^2)^2}{2y^2\left[\left|(1-\nu)y^2 - x^2(1-3\nu)\right|\right] + x^2(1-2\nu)\left[\left|x^2 - 3y^2\right|\right]}\right\}^{\frac{1}{2}}$$

Considering theoretically, the tangential stresses have minimum values for the zero value of coordinate x, namely the zero value for the slide ways width b.

This means that the design solution for the guiding system that is to be searched for is the one where the slide ways size B tends to zero, in order to minimize the tangential stresses, namely to ensure the lesser value of the friction force.

The appearance of the eccentrically loaded slider that is to be optimized is presented in Figure 3.

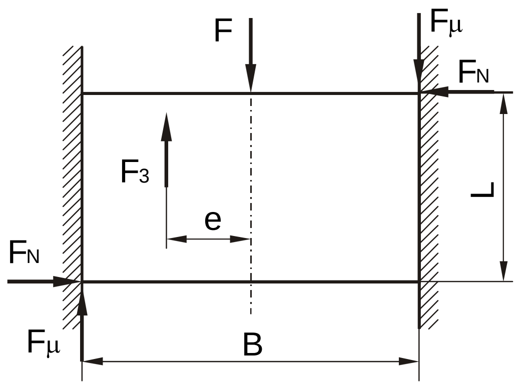

Figure 3. *The chosen model of eccentrically loaded slider*

3. PROPOSITION FOR THE DESIGN SOLUTION FOR IMPROVEMENT OF THE TRIBOLOGICAL CHARACTERISTICS OF THE ECCENTRICALLY LOADED SLIDE WAYS

Based on previous theoretical analyses, one now enters into the phase of conceiving the new construction of the eccentrically loaded slide ways whose properness, and accordingly validity is to be confirmed experimentally on a device specially designed for this purpose.

Let us consider the basic model of eccentrically loaded slider of straight line slide ways, with the coordinate system x, y, z, placed at point 0, where acts the pulling force F, that drives the slider, as it is shown in Figure 4.

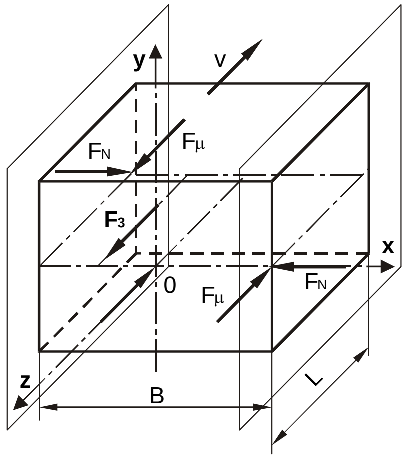

Figure 4. *Basic model of eccentrically loaded slide ways*

Slider is moving with velocity v along the slide ways due to action of the external pulling force F, to which opposes the momentum of the force F_3, through its reactions F_N and $F_{N\mu}$.

By decreasing the slide ways length L, with other value kept constant, at certain moment the self-braking occurs. The pulling force F depends on mutual ratios of geometrical parameters L and B, as well as the friction coefficient μ, what has to be experimentally checked. For the pulling force to be lower than the threshold of the pulling force of the driving engine, an attempt is made to redesign the sliding surfaces of the slider, where one expects the decrease of the slide ways size B influence on the magnitude of the pulling force F for any slider length L, Figure 5a.

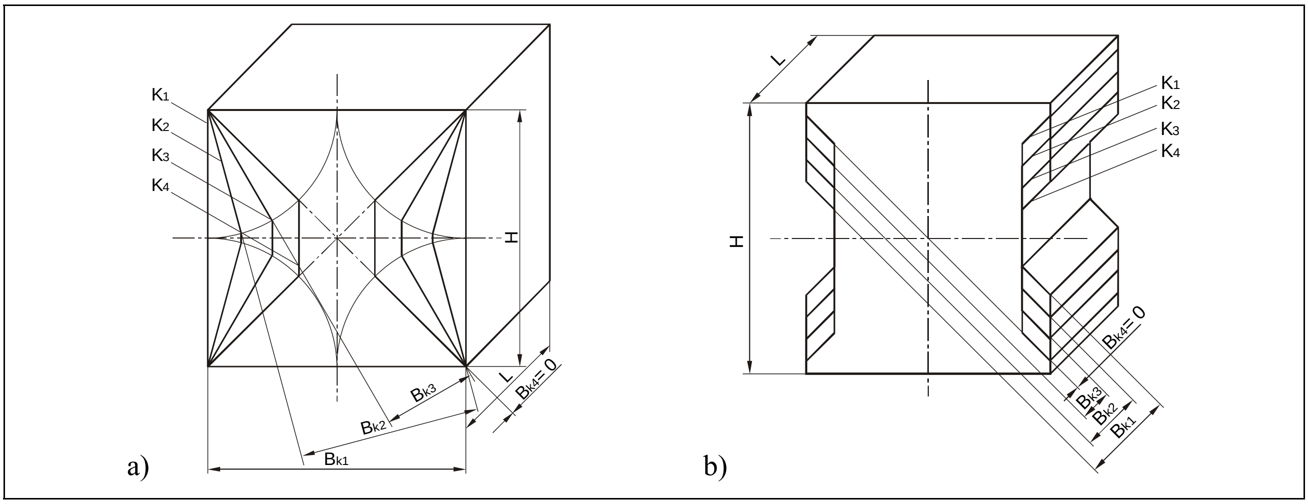

Figure 5. *Schematics of the redesign of the slider shapes - the second version of the redesign schematics of the slider shape in the sense of decrease of the slide ways size B influence on the magnitude of the pulling force F*

On the shown schematics K_1 denote the contact surfaces of sliders with slide ways, whose mutual distance is B_{k1}. Through redesign of slider along slide ways whose contact surfaces would be K_1, K_2 and K_3, mutual distance of contact surfaces decreases in such a way that:

$B_{k1} > B_{k2} > B_{k3} > B_{k4}$Ç$0$.

Redesign of sliding surfaces can be performed also in another way, as shown in Figure 5b. As with the first version in this case also is decreased the value B_{ki} starting from B_{k1} towards B_{k4}, when is achieved that $B_{k4}'=0$.

Estimates are that the second version of the slider shape redesign is more convenient for realization, so the experimental part was done with the distribution according to this version, where for the case $B_{k4}'=0$ is expected that the slider's length L can be arbitrarily decreased without increase of the pulling force F. This statement is confirmed by experiment which is the subject of this work.

The appearances of redesigned straight-line slide ways with sliders K_1, K_2, K_3 and K_4 are presented in Figure 6.

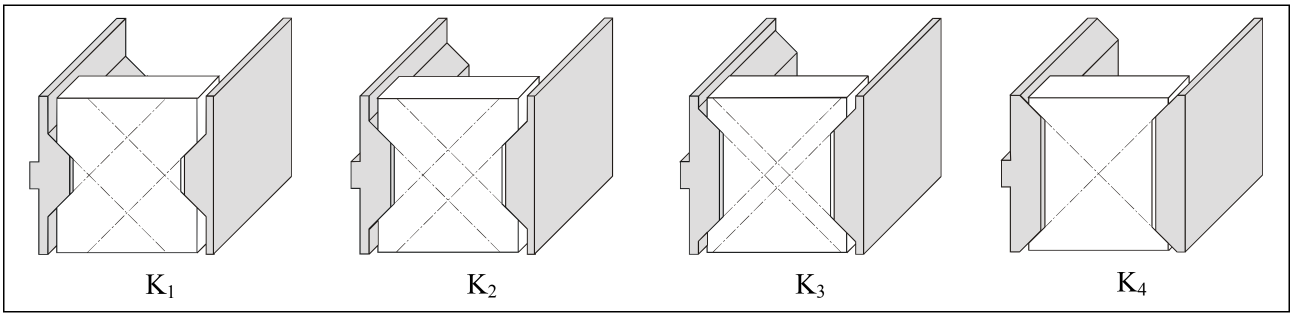

Figure 6. *Characteristic examples of redesigned eccentrically loaded slide ways, obtained by studying the problem of tribo-design*

4. DEVICE FOR TESTING OF SLIDE WAYS

For needs of experimental testing of different geometric shapes of slide ways, a special device was design and realized. Characteristic elements, as well as the way of functioning of the designed device for testing the slide ways are shown in Figure 7.

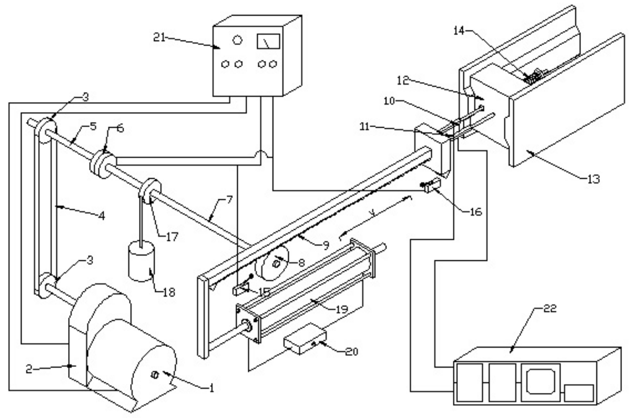

Figure 7.

Prior to realization, the complete device for testing the slide ways was three-dimensionally modeled in programming packages AutoCad 2000 and 3D Studio MAX. The animation was also performed of the operation of the complete device. By modeling the motion of the most essential functional parts, it was established that there is no collision of the moving parts and that the realization of the designed device can be performed. In Figure 8 is shown the model of the device, while in Figure 9 is shown the appearance of the realized device for testing the eccentrically loaded slide ways.

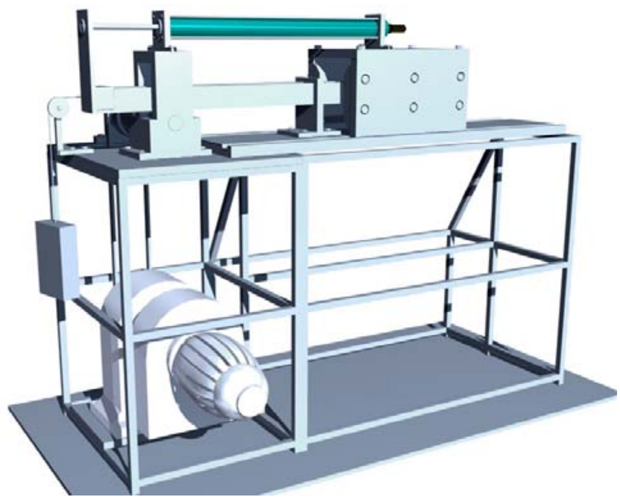

Figure 8. *Model of device for testing the slide ways*

Figure 9. *Device for testing the slide ways*

5. EXPERIMENTAL TESTING

By plan of experiments it was predicted to perform testing of the four versions of geometrical shapes of the coupled sliders and slide ways (K_1, K_2, K_3, K_4), with varying the three values of eccentric loads (F_{e1}, F_{e2}, F_{e3}, F_{e4}) and three sliding velocities (v_1, v_2, v_3, v_4). All parameters were varied according to the complete experiment plan.

As the valid parameter for comparison of slide ways would be used the total friction force F_t.

The total friction force ΣF_t is determined as a difference between the pulling force F and eccentric force F_e, as shown in Figure 10. If, at each moment, are known values of forces F and F_e, then is also known the total friction force F_t, which is obtained by computations.

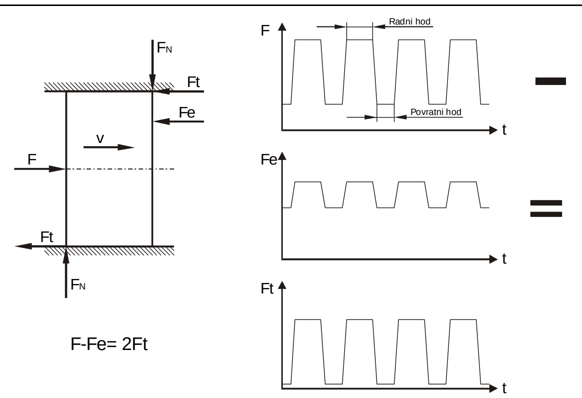

Figure 10. *Determination of the friction force*

6. TEST RESULTS

Based on recorded signals the medium values were calculated of the friction force $F_{t\,sr}$ (Table 1). Then the corresponding functional relations $F_{ti} = F_t\,(v_i)$, obtained experimentally, were formed, Figures 11, 12, 13, as well as $F_{tj} = F_t\,(K_j)$, where K_j represents the corresponding version of geometric shapes of slide ways.

From Figures 11, 12, 13 it is obvious that, for all the cases of varied shapes of slide ways K_j $(j = 1, 2, 3)$, the friction force F_t has tendency of increase with increase of eccentric load F_e , and it also changes with change of geometry of slider and slide ways, what leads to a conclusion that through change of geometry of slider and slide ways the friction force can be decreased.

As from the presented results can be clearly seen, the friction force F_t varies with variation of geometry of slider and slide ways, as well as with variation of eccentric load F_e and sliding velocity v, what leads to a conclusion that the friction coefficient does not have a constant value, what again confirms the importance of optimization of the TMS slider/slide ways shape.

For all the four cases of tested slide ways K_j, results of the friction force F_t are given by histograms according to variation of F_t with variation of sliding velocity v for all the three values of eccentric force F_e, Figures 14, 15, 16.

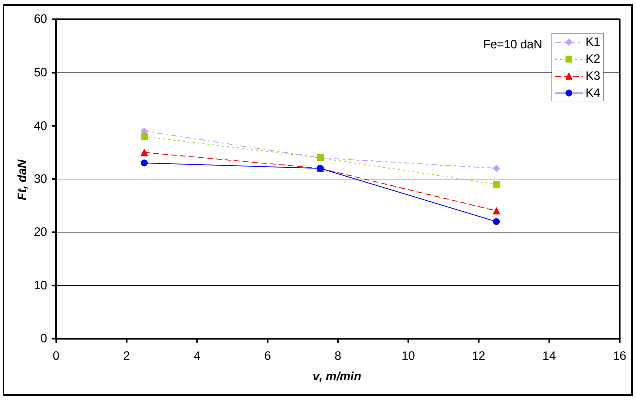

Figure 11. *Dependence F_t=F_t(v) for F_e=10 daN*

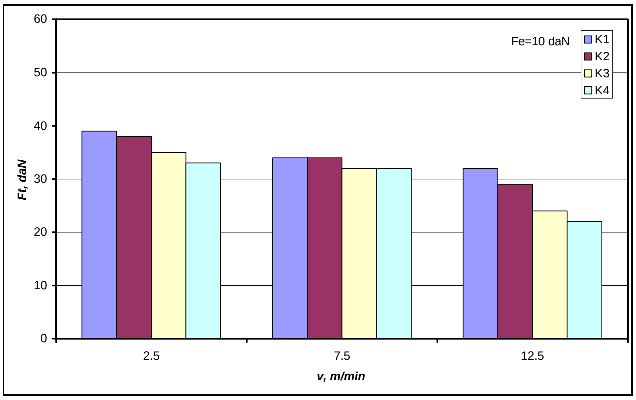

Figure 14. *F_t=F_t(v, K_j) for F_e=10 daN*

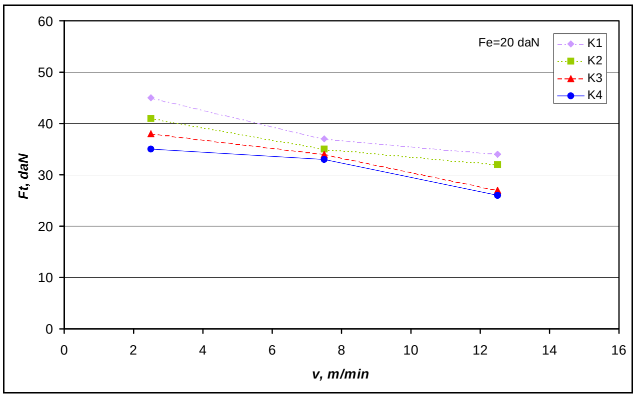

Figure 12. *Dependence F_t=F_t(v) for F_e =20 daN*

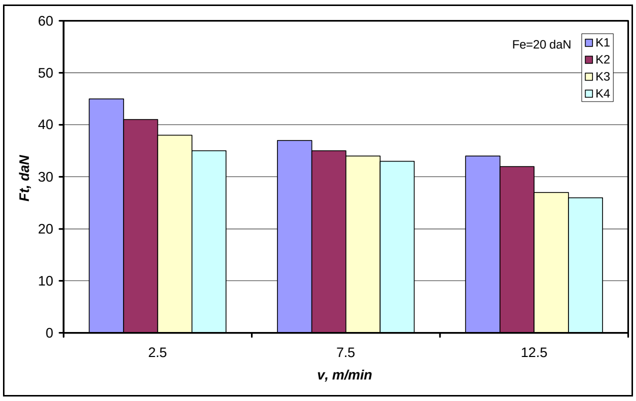

Figure 15. *F_t=F_t(v, K_j) for F_e =20 daN*

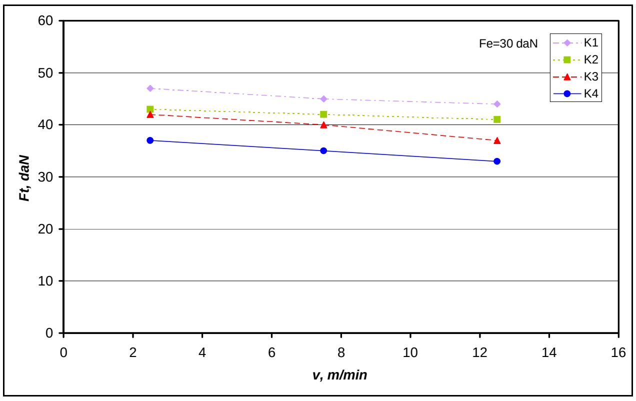

Figure 13. *Dependence F_t=F_t(v) for F_e =30 daN*

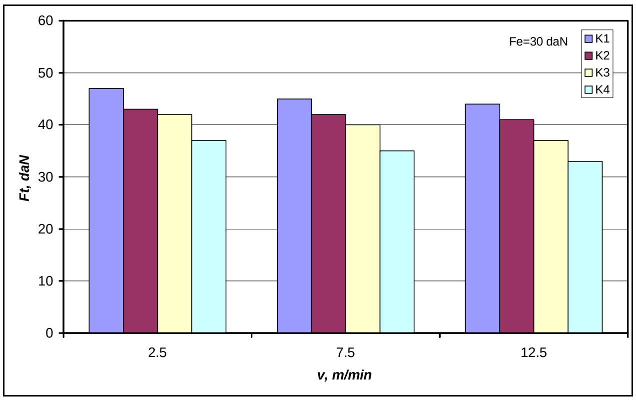

Figure 16. *F_t=F_t(v, K_j) for F_e =30 daN*

Table 1.

Varied shapes of slide ways		Eccentric load, Fe								
		Fe_1=10 daN			Fe_2=20 daN			Fe_3=30 daN		
		Sliding velocities: v_1=2.5 m/min, v_2=7.5 m/min, v_3= 12.5 m/min								
		v_1	v_2	v_3	v_1	v_2	v_3	v_1	v_2	v_3
Ft_{sr}	K_1	39.31	34.14	32.83	45.08	37.27	34.42	47.63	45.88	44.13

da N	K_2	38.87	34.19	29.25	41.46	35.22	32.71	43.13	42.02	41.38
	K_3	35.58	32.41	24.84	38.86	34.25	27.13	42.09	40.11	37.44
	K_4	33.75	32.12	22.37	35.45	33.46	26.58	37.12	35.78	33.42

7. CONCLUSIONS

The general conclusion could be that for all the four cases of varied geometries of eccentrically loaded slide ways K_1, K_2, K_3 and K_4, the friction force F_t has the minimum value for the shape K_4, where the opposite sliding surfaces belong to the same plane.

The conducted experiment has shown the noticeable improvement of the slide ways shape K_4 with respect to shape K_1 for more severe working regimes, for cases of higher eccentric load. Thus, for work with eccentric force of F_e = 30 daN, decrease of the friction force is achieved within limits 28.2 to 32 %. With decrease of eccentric force F_e, the convenient shape of geometry of slide ways K_4 with respect to other slide ways is still noticeable, only it is somewhat less prominent.

For different cases of varied values of the sliding velocities form 2.5 to 12.5 m/min and eccentric load from 10 to 30 daN, percentage of decrease of the friction force F_e with respect to the most inconvenient shape of the slide ways K_1, and in favor of the slide ways shape K_4, lies within limits 1σ, from 25 to 32 %.

By designing eccentrically loaded slider K_4 and corresponding slide ways, it is enabled to shorten the slider and extension of the useful working path in machine, with diminishing the self-braking phenomenon of the slider along the slide ways.

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