

Stiffness increasing device for the moving elements of the CNC milling machines

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Abstract. This work presents a new concept of automatic compensation of the ram deformation during its in-cantilever horizontal motion of the milling machines that feature a ram slide (headstock) on the column side. During the ram horizontal motion that is a CNC controlled axis of the machine, there is a permanent modification of the ram gravity center that makes the reaction forces in the guideways to permanently change their rates as well. Thus, an elastic variable deformation occurs on the ram supports along with the elastic deformation of the guideways and the elastic deformation of the ram slide. The new concept of automatic compensation of the elastic deformation presented in this work takes its compensation force from the feed kinematic linkage of the ram, having interdependence between the ram motion length and the rate of the compensation force. The rate of the compensation force taken over from the kinematic linkage is very low, then it is mechanically multiplied (>100 times), until the value needed for the quasi-total compensation of the elastic deformations is reached. The experimental application of this new concept has been done on a milling machine model BFK 1000 CNC, where an increase by 90% of the ram geometrical accuracy while going out in cantilever has been obtained, compared to the case when such a compensation device would be missing.

1. Introduction

Milling machines having the horizontal spindle and its headstock located on the column side feature a series of advantages in terms of design and several deficiencies in terms of achieving the geometrical accuracy, caused by the horizontal motion of the gravity center of the ram. The negative result of this situation may be found in the straightness values during the horizontal motion of the ram (Z axis of the machine). In other words, a “droop” of the ram occurs as long as its cantilever distance from the headstock increases. The straightness deviation of the ram will affect the quality of the work pieces processed on such a machine. Most of the machine builders are sizing “through measurement” the ram stiffness and, at the same time, they limit the ram travel in order to comply with acceptable rates of straightness on Z axis. In case of the large size milling machines where the ram (Z axis) travel is long and the ram weight is heavy as well, to obtain an acceptable straightness rate along Z axis will become a problem that cannot be sorted out just through sizing. of the ram elastic deformations during its motion in console is. In such cases, compensation solutions are applied. The hydraulic compensation system well known. Thus, during the ram motion a pressure increase is generated into the guideway pockets located on the ram front side that compensates the ram weight variation in such a way that the machine geometrical accuracy is little affected. Deformations are offset through a pressure regulator that modifies the pressure value into the guideway pockets. The actuator of the regulator takes its



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motion from ram kinematic linkage of the feed, providing interdependence between the ram position and pressure rate into the front side pockets. The disadvantage of this system consists of imposing the condition to provide hydrostatic lifting of the ram that leads to higher manufacturing and maintenance costs. Another well-known ram balancing compensation solution consists of a system of two wires (steel ropes) and fixed pulleys located on top of the column; one end of the wires is fixed to a counterweight located inside the column and the other end is fixed to the ram. During the ram motion a mechanical system provides the motion of the two wires as well, so that on the front side of the guideway weight variations will no longer occur. The disadvantage of this balancing-compensation system consists of the intricacy of the assembly composed of counterweight, pulley, wires and ram synchronous motion mechanism. The same problem concerning the Z axis straightness is encountered on the large size boring and milling machines where, instead of the ram, there is a quill performing a horizontal motion. In such a case, the presence of a compensation system is difficult because of the intricacy of the headstock assembly since its structure includes both the milling and boring spindles.

2. Structural elements that affect the ram stiffness on milling machines

As it may be noticed in figure 1, the headstock 3 moves vertically along the column guideways 4 (Y axis) and, at the same time, the headstock 3 provides the guidance of the ram 2 that moves horizontally along with the milling head 1. The travel of the ram 2 leads to increasing the ram cantilever distance and to modifying the position of the application point of the ram weight, G_c .

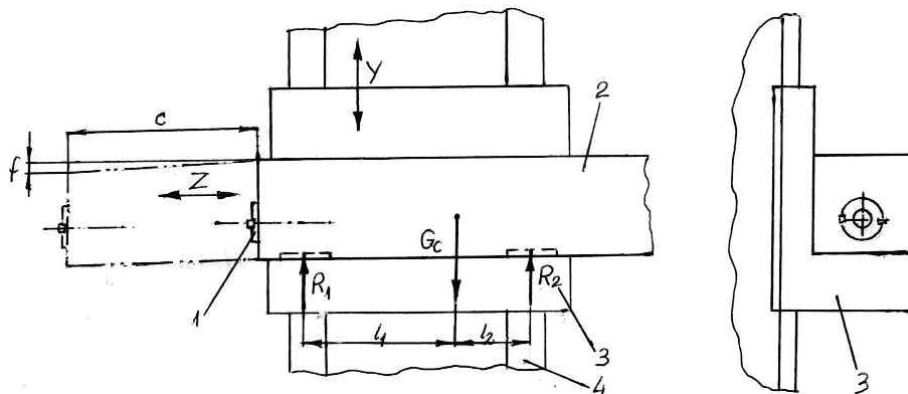


Figure 1. Constructive structure of a milling machine having the headstock on the column side.

In case when the ram is retracted, i.e. $c=0$, the reaction forces R_1 and R_2 into the guide ways of the ram 2 will be:

$$G_c = R_1 + R_2 \quad (1)$$

$$G_c \cdot l_2 - R_1(l_1 - l_2) = 0 \quad (2)$$

$$R_1 = G_c \frac{l_2}{l_1 + l_2}; R_2 = G_c \frac{l_1}{l_1 + l_2} \quad (3)$$

Where G_c means the ram weight, l_1 is the distance from the front guide way to the ram gravity application point and l_2 is the distance from the rear guide way to the ram gravity application point. In the case when the ram achieves the maximum travel c , the values of the two reaction forces R_1 and R_2 will be:

$$R_1 \frac{l_2 + c}{l_1 + l_2}; R_2 \frac{l_1 + c}{l_1 + l_2} \quad (4)$$

By comparing the values of the reaction forces R_1 and R_2 in case of the extreme positions of the ram, a substantial modification of these values will be noticed. Thus, at the maximum value of the ram travel, the reaction force R_1 will increase very much, leading to the increase of the elastic deformation of the front bearing support; this will be reified on the ram front side through the sag (deflection) f . There is a relation of dependence between the ram position in relation to the origin of Z axis and the value of the reaction forces R_1 and R_2 , so that the presence of a compensation system for the values of the reaction forces is strictly necessary in order to decrease (or even annul) the deformations represented through the sag f . For this purpose, the compensation system must take over its compensation action from an element of the feed kinematic linkage of the ram. This element of the kinematic linkage may be the moving element itself, consisting of the ram, whose horizontal motion is related to the origin of Z axis, in case when the ram is retracted.

3. Compensation system of the ram elastic deformation

The new compensation system of the ram elastic deformation has a mechanical structure, figure 2, consisting of a linear-tilted key 1, fixed by bolts to the ram 2 that moves horizontally (Z axis) along the guideways located into the ram slide 3 and that, on its turn, moves vertically along the guideways of the column 4. The bearing 5 is in contact with the tilted side of the linear-tilted key. This bearing is fixed to the tappet 6 through the shaft 7 and the tappet 6 can move vertically because it is guided inside the support 8 that is fixed to the ram slide by means of several bolts. The bottom of the tappet 6 comes into contact with an end of the lever 9 and the other end of the lever is solidly fixed to the end of the eccentric shaft 10. This shaft is borne onto two bearings 11, located into the ram slide 3. On the eccentric portion of the eccentric shaft 10 an oscillating roller bearing 12 is mounted; this bearing comes into rolling contact with the low side of the ram 2. The kinematic linkage of the ram 2 performs the cantilever motion of the ram on which the linear-tilted key 1 is fixed; the tilted surface of the key is in contact with the bearing 5, thus acting on the tappet 6 that, on its turn, rotates the lever 9 along with the eccentric shaft 10. Due to the eccentricity of the surface on which the roller bearing 12 is located, a force will be generated that acts upwards, thus compensating the additional weight of the ram on the front side guideway. In this manner, the sag f of the ram in cantilever will be compensated and the geometrical accuracy of the machine will improve. This compensation system has a high multiplication ratio of the force taken over from the feed kinematic linkage of the ram, so that when the linear-tilted key 1 acts on the tappet 6, the force value will be very low. Subsequently it will be highly multiplied, generating a high force for compensating the drop of the ram 2, sufficient to annul the sag f . Due to the high multiplication ratio of the force, it results that the mechanical compensation system needs a very low power rate from the feed kinematic linkage of the ram and will not affect negatively its capacity to provide the Z axis feed. The tilting value of the linear-tilted key 1 confers the possibility to optimize the interdependence between the cantilever position of the ram 2 and the value of the compensation force, so that the sag f can be significantly decreased. The scheme of the structure of the ram elastic deformation compensation system is shown at figure 3. The main parameters of the compensation system structure are: ε - compensatory deformation of the front guideway of the ram during the horizontal motion (Z axis); e - eccentricity of the surface on which the radial roller bearing is located; α - rotation angle of the eccentric shaft; m - arm of the lever that is solidly fixed to the eccentric shaft; c - ram travel; t - the small cathetus of the linear-tilted key. The main parameters of the compensation system structure are: ε - compensatory deformation of the front guideway of the ram during the horizontal motion (Z axis); e - eccentricity of the surface on which the radial roller bearing is located; α - rotation angle of the eccentric shaft; m - arm of the lever that is solidly fixed to the eccentric shaft; c - ram travel; t - the small cathetus of the linear-tilted key. The dependence between the compensatory deformation ε and the rotation angle α of the eccentric shaft may be written as below, figure 3(b):

$$\varepsilon = e \sin \alpha \quad (5)$$

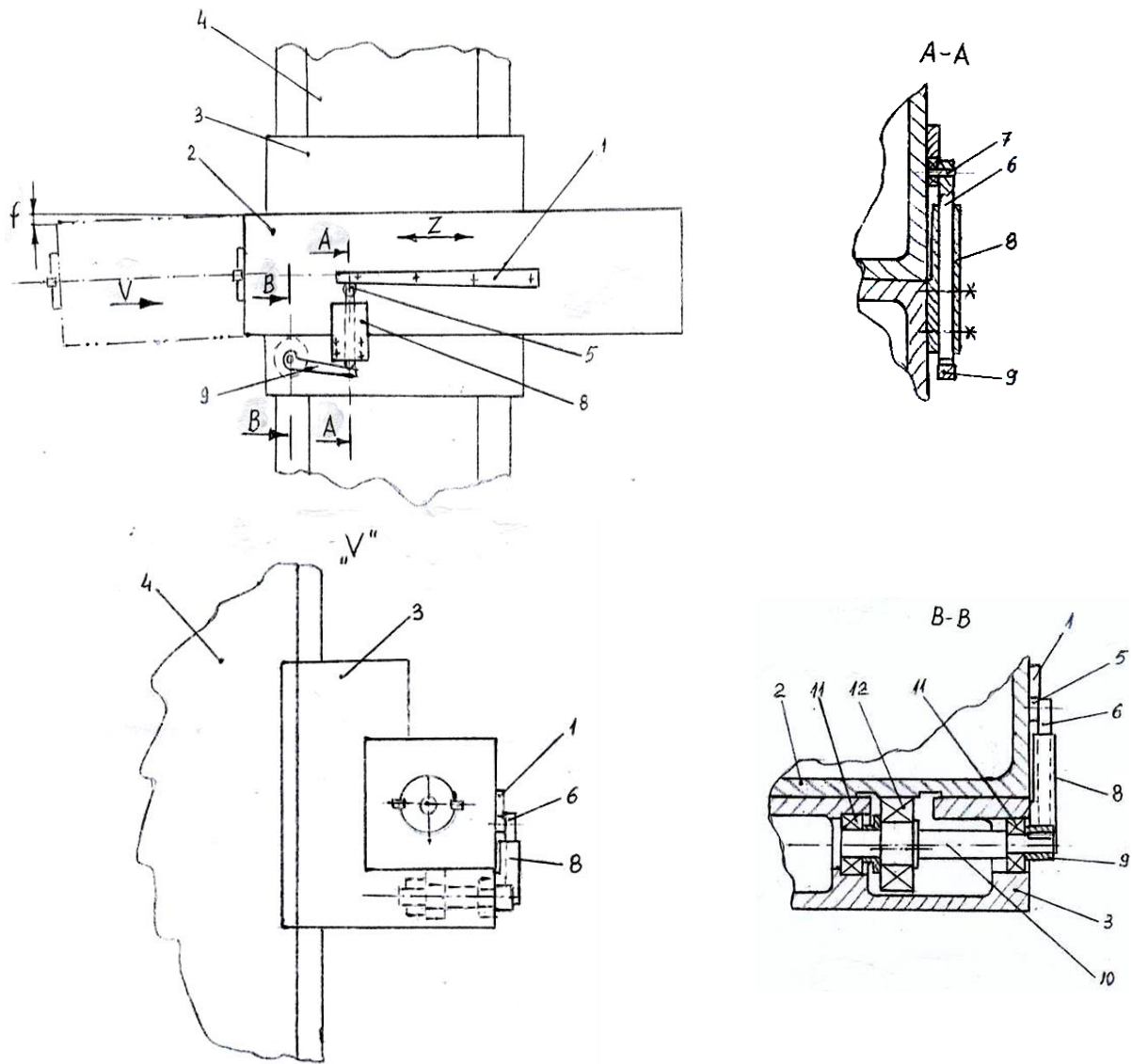


Figure 2. Ram (Z axis) compensation system.

At the same time, from figure 3(c), it will result:

$$\sin \alpha = \frac{t}{\sqrt{m^2 + t^2}} \quad (6)$$

By replacing the relation (6) into (5), it will result:

$$\varepsilon = \frac{et}{\sqrt{m^2 + t^2}} \quad (7)$$

The relation (7) represents the gradient of the compensatory deformation in function of the small cathetus t of the linear-tilted key when the ram is moving along the travel c . By considering the tilting angle β , of the linear-tilted key, it will result:

$$\tan \beta = \frac{t_{max}}{c_{max}} \quad (8)$$

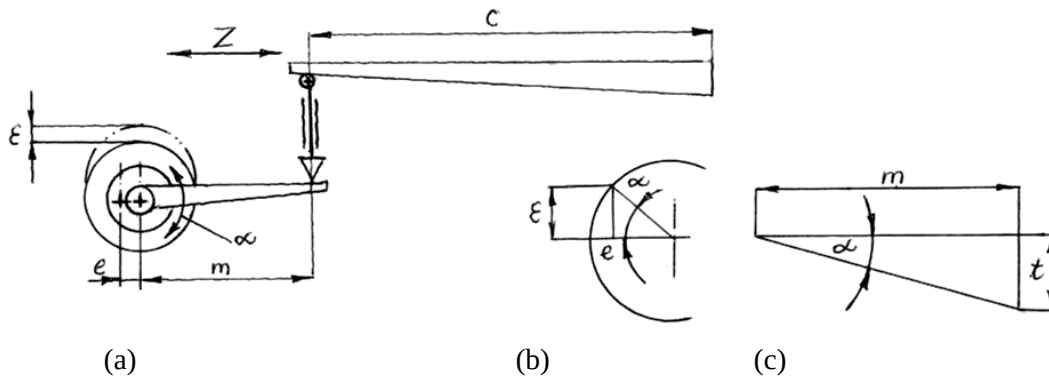


Figure 3. Scheme of the structure of the ram elastic deformation compensation system; (a)- structure of the compensation system; (b) - position of the eccentric shaft; (c) - angle of the linear-tilted key.

where c_{max} is the maximum travel of the ram and t_{max} is the small but maximum cathetus of the linear-tilted key. In this case, the relation (7) will become:

$$\varepsilon = \frac{e c \tan \beta}{\sqrt{m^2 + c^2 (\tan \beta)^2}} \quad (9)$$

Figure 4 shows the diagram of the dependence of ε in function of the ram position at various points of the travel c .

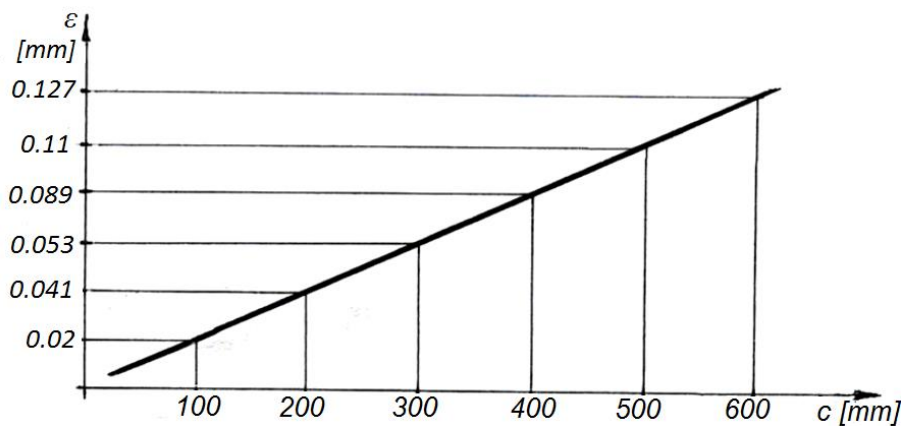


Figure 4. Values of ε of the compensation system in relation to the ram position c .

The following constant values have been considered: $c_{max} = 600 \text{ mm}$; $e = 1 \text{ mm}$; $m = 120 \text{ mm}$; $t_{max} = 15 \text{ mm}$. It may be noticed that the compensatory values ε are at the level of the front guideway of the ram and these values are different from those recorded at the level of the ram front side, i.e. the values of f . In order to correlate the values of ε and f the stiffness of the eccentric shaft has to be analyzed as well, especially the stiffness of its bearings. A careful analysis of the dependence of the two values ε and f , completed with the necessary tests and adjustments, may even annul the value of the sag f . Based on the structure of the elastic deformation compensation system, the value of the transmission ratio i of the system can be settled:

$$i = \frac{\varepsilon}{t_{max}} \quad (10)$$

The value of the transmission ratio i is helpful for settling the ratio between the compensatory force generated by the oscillating roller bearing located on the ram front guideway and the input force that acts upon the tappet.

4. Experimental trials

The new compensation system of the ram elastic deformation has been applied on a milling machine model BFK manufactured by WMW Bacău, Romania. This machine has three CNC controlled axes (X, Y and Z) and the table surface is 800 x 1200 mm. The travels are: X axis = 1400 mm, Y axis = 900 mm and Z axis = 650 mm.

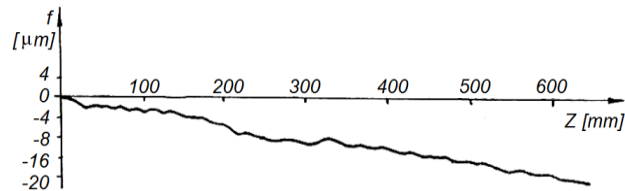


Figure 5. Sag f of Z axis without compensation system.

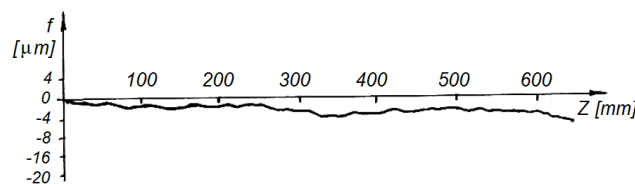


Figure 6. Sag f of Z axis with the compensation system being activated.

The diagram of the elastic deformation at the ram front level (the sag f) in case when the compensation system is not activated, is shown at figure 5. The diagram of the elastic deformation in case when the new compensation system is activated is shown at figure 6. An improvement of the Z axis straightness by 90% has been obtained.

5. Conclusions

The new compensation system of the ram elastic deformation has led to improving the straightness of Z axis at rates close to its annulling. At the same time, the feed kinematic linkage of Z axis, from where the compensation force has been taken over, has not been affected by this. The compensation system presented in this work has a simple structure and has been built at low costs. Its reliability is high as well, because it is made of well-known mechanical elements. The compensation system allows the performance of over adjustments on the interdependence between the sag f and the ram position along its travel on Z axis, by adjusting the tilting the linear-tilted key fixed on the ram.

6. References

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