Researches and constructive solution to enhancing the stiffness of the gantry type milling machines

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Abstract. For machining of large size work pieces, CNC milling machines with moving bridge (known as Gantry Type milling machines) that have a large distance between the bridge columns, i.e. 4 to 7 meters are frequently required. This leads to a difficult guiding, driving and stiffening of the moving bridge. At the same time, the moving bridge is required during the cutting process by a number of forces, some of them fixed, others variable. The manufacturers of such machine tools sort out this problem by a “suitable” sizing of the moving bridge that involves large overall dimensions of the bridge, with major implications on its guiding. This work presents an innovative solution concerning the increase of the stiffness of the Gantry type bridge, so that the geometrical accuracy of the machine is preserved regardless the force variations in the guiding system of the moving bridge. This work sets out an interdependent connection between the variation of the requiring forces generated at a high extent by the location of the moving elements on the cross rail and the compensating forces in the guiding of the two columns of the bridge.

1. Introduction

For obtaining high accuracy work pieces, the usage of high accuracy machine tools is necessary. The accuracy of a machine tool is conditioned at a high extent by its geometrical, kinematic and thermo-mechanical errors. In case of the geometrical accuracy of the machine, of major importance is the linearity of the kinematical axes and their mutual position [1].

As far as the machining of large size work pieces is concerned, the Gantry type milling machines with numerical control (CNC) are frequently used. The specificity to this type of machine tool consists of the moving bridge (portal), both in terms of its driving (Gantry system) and stiffness. It is known that machining of large size work pieces requires a moving bridge having a large clearance between columns (4 through 7 m) hence the difficulty in providing proper guiding, driving and stiffness [2].

The moving bridge is composed of two columns stiffened through one cross rail and the cross rail provides the horizontal motion of one, two or more cross slides that, on their turn, are fitted with a ram that moves vertically. Inside the rams the milling spindles are located. Through the motion of the moving elements along the cross rail, variable forces come up that require the moving bridge, especially in terms of the guiding of the two columns of the bridge along their guideways [3].

The moving bridge, along with the moving elements located on the cross rail, has large overall dimensions as well a heavy weight. For this reason the two columns of the bridge need to be guided by means of rolling elements or hydrostatic lifting that generate a very low friction coefficient. At the same time the guideways with rolling elements and hydrostatic lifting, do not assure a high stiffness when submitted to variable forces [4].
This affects negatively the machine geometrical accuracy and the quality of the work pieces being machined, respectively. In this context most of the builders of moving bridge machine tools are sizing the bridge guiding system such as to comply with the admissible values of the machine geometrical accuracy. The choice of such a version is easier and involves lower costs, but it cannot provide high performances [5].

The second version which provides a very good geometrical accuracy to the machine consists of compensating the elastic deformations of the rolling elements or of the hydrostatic supports when the forces acting on the guiding system are variable. This version involves higher costs compared to the first version but it improves considerably the accuracy of the work pieces to be machined.

2. Law of variation of the forces in the moving bridge guiding system

Two types of Gantry type milling machines are subject to the analysis. The structure of the first one includes one milling head, figure 1, and the second one has two milling heads, figure 2.

a) In case of the first version the moving bridge is composed of the two columns 1 that are stiffened through the cross rail 3, forming together the moving element, i.e. the kinematic axis \( X \) that moves along the guideways located on the two beds 2. Along the cross rail 3 the cross slide 4 moves horizontally as shown at figure 1, i.e. the kinematic axis \( Y \) and the ram 5 provides the kinematic axis \( Z \). The ram contains the milling spindle 6. The floor plate 4 assures the possibility to fix and set the work pieces to be machined.

![Figure 1. Forces in the structure of the Gantry type milling machine with one milling head.](image-url)

The two guideways 7 that provide the supporting and guiding of the moving bridge will be submitted to the variable forces \( V_1 \) and \( V_2 \) because of the variable position of the cross slide 4 along the cross rail 3. As shown at figure 1 the total weight of the moving element along the \( X \) direction/axis is given by the weight of the moving bridge \( G_p \) and the force \( F \) that represents the sum of the weights of the cross slide 4, i.e. \( G_s \) and the ram 5, i.e. \( G_c \), as shown in equation (1).

\[
F = G_s + G_c
\]  

By knowing, from the design conditions, the travel \( l \) of the kinematic axis \( X \), as well as the distance \( c \) from the origin of \( X \) axis to the guiding system of the moving bridge, the values of the forces that are requiring the two guideways 7 \( (V_{1\text{min}}, V_{1\text{max}}, V_{2\text{min}}, V_{2\text{max}}) \), can be calculated in equation (2) and (3).

\[
V_{2\text{mi}} = \frac{F_c}{1+\frac{c}{z}} + \frac{G_p}{2}
\]  

(2)
\[ V_{2_{\text{max}}} = \frac{F(c+1)}{1+2c} + \frac{G_p}{2} \]  

(3)

The variation of the forces in the guideways 7, i.e. \( V_1 \) and \( V_2 \) in relation to the position of the cross slide 4 on the cross rail 3 is shown at figure 2.

\[ \Delta V = \frac{F-1}{1+2c} \]  

(4)

b) In the case of the second version, on the moving bridge cross rail two milling heads are moving horizontally, corresponding to the kinematic axes \( Y_1 \) and \( Y_2 \), as shown at figure 3. It is to be noticed that the axes \( Y_1 \) and \( Y_2 \) have the same travel \( l \) and the distance \( c_1 \) is considered from the origin of each kinematic axis to the guiding system and the distance \( c_2 \) is from the maximum limit of the travel \( l \) to the opposite guiding system.

\[ F = G_{s1} + G_{c1} \]  

(5)

Figure 2. Variation of the forces \( V_1 \) and \( V_2 \) in the guiding system in relation to the position of the milling head on the cross rail.

Figure 3. Forces in the structure of the Gantry type milling machine equipped with two milling heads.

In this case the forces on the column can be calculated by using the equations (5) and (6).
The above symbols mean: \( G_1 \) - weight of the cross slide 1; \( G_2 \) - weight of the ram 1; \( G_3 \) - weight of the cross slide 2; \( G_4 \) - weight of the ram 2; \( F_1 \) - force of the milling head 1; \( F_2 \) - force of the milling head 2. In this case the values of the forces \( V_1 \) and \( V_2 \) that are requiring the guiding system of the moving bridge, figure 3, will be variable in function of the positions on the cross rail of the two milling heads. In order to find out the maximum value of the forces \( V_1 \) and \( V_2 \), the maximum and minimum values \( V_{1\text{max}} \), \( V_{1\text{min}} \) and \( V_{2\text{max}}, V_{2\text{min}} \) respectively will be determined by using the equations (7) and (8), and then their difference \( \Delta V \) by using the equation (9).

\[
V_{2\text{max}} = \frac{F_2(c_1+1)+F_4(c_2+1)}{1+c_1+c_2} + \frac{G_p}{2}
\]

(7)

\[
V_{2\text{min}} = \frac{F_1c_1+F_2c_2}{1+c_1+c_2} + \frac{G_p}{2}
\]

(8)

\[
\Delta V = \frac{|(F_1+F_2)|}{1+c_1+c_2}
\]

(9)

Further to the analysis of the relations (4) and (9) of the two constructive versions of the Bridge type milling machines it results that the variation of the forces in the guiding system will generate an elastic deformation of the rolling elements or of the hydrostatic support, with direct effect on the geometrical accuracy of the machine. Practically, the moving bridge subassembly will tilt along with the milling heads, but the machine floor plate will remain fixed; this will directly affect the machining accuracy of the work pieces to be machined. The more the difference \( \Delta V \) is higher, the more the geometrical accuracy of the machine will be affected; in other words, the parameters that are directly influencing the machine geometrical accuracy are the travel \( l \) of the kinematic axis \( Y \) and the weights of the cross slide \( G_3 \) and the ram \( G_4 \).

3. Design solution to increasing the stiffness of the Gantry type milling machines

With a view to diminishing the negative effects on the geometrical accuracy of the Gantry type milling machines presented above, the manufacturers are using guideways of increased stiffness that only limit the phenomenon and lead to increasing the friction coefficient into the guideways. A design solution to annul the effects presented above will be described further on; it refers to a mechanism to compensate the elastic deformations in the guiding system of the moving bridge.

The basic law that settles the variation of the forces into the guiding system has, as a variable, the position of the milling head along the travel \( l \) of the kinematic axis \( Y \). In case of the version with two milling heads there will be two variables given by the position of each milling head along its travel, corresponding to the axes \( Y_1 \) and \( Y_2 \) respectively. In case of the machine equipped with one milling head a compensation mechanism will be inserted into each guiding system, right side and left side. These will generate the compensating forces, variable according to the same variation law shown at figure 3.

Each compensation mechanism will be controlled by means of a signal taken from the measuring system of \( Y \) axis that indicates the location of the milling head on the cross rail. As such, in each one of the two columns of the bridge, a compensation mechanism will be placed at the level of the guiding system. The compensation mechanism is composed of two floating supports 1, see figure 4 that generate a variable force from the bed \( B \) towards the column \( C \) of the moving bridge, in function of the position of the milling head on the bridge cross rail.

A detailed description of the floating support is presented at figure 5. Its main components are: the piston 1 that is contact with the bed \( B \) by means of an O ring 2 made of teflon and its upper side has a rubber O ring 3 that provides the tightness with the plate 4. The floating support will move along with the moving bridge, condition that requires the provision of tightness between the bed \( B \) and the
floating support and, at the same time, the generation of the compensating variable force. For this purpose, the piston 1 assures the formation of two rooms, $a$ and $b$ that have different surfaces so that, the difference between these two surfaces multiplied by pressure will generate a low but sufficient force to assure the tightness with the bed $B$.

![Figure 4. Location of the floating supports.](image1)

![Figure 5. Cross section into a floating support.](image2)

The variable compensating force is generated through the modification of the pressure in the floating supports by means of proportional reduction valves $S_p$, figure 6. Thus, each one of the two compensation mechanisms located into the left side guideway $G_s$ is supplied with pressure by means of a pilot valve that modifies the pressure into the floating supports in function of the signals $e_1$ and $e_2$ respectively, taken from the position of the milling head on the cross rail.

![Figure 6. Hydraulic diagram of the compensating solution.](image3)

The position of the milling head is given by the position encoder of $Y$ axis and materialised into the signal $i$ that is input to the controller $c$. The controller will process it and generate the two output signals $e_1$ and $e_2$, diametrically opposed, assuring in this manner the law of variation of the compensating forces in the two compensation mechanisms. The two compensation mechanisms are hydraulically driven by the hydraulic unit on the machine that provides the pressure $P$ that further on will be modified by means of the proportional reduction valves.

In case of the machine equipped with two milling heads, each compensation mechanism has four floating supports and two proportional reduction valves. One valve will receive the control signal from
Y₁ axis and the other one will receive it from Y₂ axis. In this manner, the compensation mechanism will make the sum of the compensating forces generated by the position of each milling head on the cross rail.

4. Experimental results
The solution presented above has been applied on a Gantry type milling machine, equipped with one milling head, at the company REM Machine Tools. The moving bridge is guided by means of both carrying and directional roller packs and the clearance between the moving columns is 5.5 meters. The travel of the milling head along the cross rail is 4 meters (Y axis) and the vertical travel of the ram is 1.6 meters (Z axis). The measuring encoder of Y axis is linear with direct measuring and the signal that indicates the position of the milling head on the cross rail is taken over from the numerical control and afterwards processed by means of a controller.

The geometrical accuracy of the machine has been measured with and without the activation of the compensation mechanisms. Thus, in the case of inactive compensation mechanisms the horizontality of the moving bridge cross rail during the motion of the milling head along its entire travel has been recorded by means of dial gauges. A variation of the column positions towards the bed in value of 0.18 mm has been recorded. The same measurement has been performed with active compensation mechanisms and a variation of 0.02 mm has been recorded.

Even though roller packs have been used with the purpose to increase the stiffness of the guiding system, the effects of the force variations upon the rollers reach unacceptable values for the machine geometrical accuracy. It is recommended that the Gantry type milling machines having a large clearance between columns (larger than 4 meters) to be equipped with compensation systems, thus providing the improvement of the accuracy of the work pieces.

5. Conclusions
The solutions presented in this paper are recommended for all Gantry type milling machine with mobile central columns and with distances between columns greater than 2.5 meters. By using this method on Gantry type milling machines, a greater geometric stability is achieved and assures the possibility of using them on milling casings or frameworks up to class 6 precisions. The same mechanical solutions can be used on other types of machines, that have a central column system, as planers or grinding machine. These methods of analysis and the solution presented can be utilised on three-dimensional measuring machines with high distanced between columns that need a higher geometric precision.

6. References
[1] Catrina D, Totu A, Croitoru S, Carutasu G and Dorin Al 2005 Sisteme flexibile de prelucrare prin aschiere (Bucharest: Matrix Publishing House)
[2] Feng W, Yao X, Azamat A, Yang J 2015, Straightness error compensation for large CNC gantry type milling centers based on B-spline curves modeling International Journal of Machine Tools & Manufacture 28 165-173
[3] Lei W, Haitao L, Lei Y, Jun Z, Wanhua Z, Bingheng L 2015 The effect of axis coupling on machine tool dynamics determined by tool deviation International Journal of Machine Tools & Manufacture 88 174-182
[4] Gheorghe S, Ciobanu R, Pal A 2007 Balancing, Compensation System for the Vertically Moving Elements of the Machine Tools with Numerical Control Meccanica 43(6) 515-529 Springer Netherlands.
[5] Bell Y 2005, Compensation system of backlash and pitch of the feed kinematical linkages of the numerical control machine tools, Progresivie Tehnologii Sistemi Masinocstroenia, 165-173, Donet