Influence of the ball screw stiffness on the positioning accuracy of the CNC machine tools

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Abstract: This work is researching the ball screw stiffness influence on the positioning accuracy of the feed kinematic linkage of a CNC machine tool fitted with an indirect type measuring system for positioning the moving element. Because the machine tool is equipped with an indirect measuring system, backlashes and elastic deformations in the kinematic linkage cannot be offset by the position loop, consequently these are directly affecting the positioning accuracy. In this context, methods are used for diminishing their influence as well as compensation methods with a view to improve the positioning accuracy. Ways to preloading the ball screws and bearings of the ball screw are presented as well, by focussing on the optimisation, through calculations, of the preloading force value. An important component, analysed in this work, consists of the research of the ball screw–nut stiffness that depends on the strategy being used on the axial bearing of the two ends of the ball screw. The experimental tests have been performed at idle running on the CNC milling machine (CPFH 500), fitted with three controlled axes (X, Y and Z). Based on the experimental results, it may be noticed that, once increased the ball screw stiffness, within the limits stated by the ball screw supplier, the positioning accuracy will increase as well.

1. Introduction

The most used mechanisms for transforming the rotation motion from a servomotor into the translation motion of a moving element are the ball screws [1]. These subassemblies are to provide a position as accurate as possible of the moving element slide and to assure a working speed rate as high as possible for an increased productivity and high quality. An important role for reaching a high positioning accuracy is played by the bearing system as well, without which the stiffening of the ball screw would not be possible. The design of the feed kinematic linkages of the CNC machine tools features two types of bearing: thrust (axial) bearing at an end of the ball screw and free at the other end, or thrust bearing at both ends. In case of the preloaded ball screw nuts there are researches that show that these have a low effect on the system stiffness, but a too high preloading of them will have a negative effect because this would lead to increasing the friction force between the ball screw and the nut [2]. This will affect the positioning accuracy of the moving element. In this respect researches have been carried out with reference to the ball screw–nut system stiffness variation caused the temperature rising; for this purpose the method of the finished 3D FEM has been used [3]. This method of analysis can exactly anticipate the variation of the preloading force in function of the temperature of the ball screw-nut subassembly. A new research method highlights the preloading of a ball screw based on the vibration signals of the servomotor when the ball screw is preloaded at various preloading rates, of 2%, 4%, and 6% [4]. The
characteristics of the signals having various preloading rates have been categorised based on the SVM (support vector machine) calculation algorithm and for the vibration signal the algorithm GA/KNN (genetic algorithm k – the closed neighbour) has been used. Based on the related results, the state of functioning of the feed kinematic linkage can be determined.

2. Assembly diagram of the feed kinematic linkage with the speed and position loops in case of indirect measuring

Figure 1 shows the block diagram of a feed kinematic linkage of a CNC machine tool that uses the balls screw-nut mechanism to transform the rotation motion from the servomotor into the translation motion of the moving element, having an indirect type measuring system of its position.

Figure 1. Block diagram of the feed kinematic linkage with indirect measuring system of the position.

The feed kinematic linkage is driven by the A.C. motor 3 that develops a torque to drive the ball screw 4. For measuring the position of the moving element 5 there is the position loop Bp that includes the position rotary encoder care Tp coupled to the ball screw. The position rotary encoder can be mounted in two modes: on the output from the motor shaft (Tp’) or on the output from the feed kinematic linkage, opposite to the motor (Tp). The location of the rotary encoder at the end of the ball screw (Tp) can provide a more accurate positioning of the moving element because the position loop includes the angular deformations of the ball screw, as well. The indirect measurement of the slide position is performed through the emission of a signal Sr, of the ball screw rotation rate that will be compared to the signal Si that is obtained through the conversion into the numeric-analogue convertor CNA of the input rate Mi. The comparator 1 will analyse the input and output signals and will settle the error “Si – Sr”. This error will be conveyed to the frequency convertor 2 that will amplify the output current a supplied to the A.C. motor 3, until the error will be annulled. In case of the speed loop Bv, the speed is measured by means of the speed encoder Tv, whose signal is proportional to the rpm and the rpm error is offset at the level of the comparator C in the structure of the frequency convertor 2. The speed feedback loop provides the obtaining of an imposed rpm of the A.C. motor, regardless the rate of the driving force Fa, thus avoiding the acceleration at light loads or deceleration at heavy loads of the moving element. The speed loop is independent from the position loop because each loop has its own input and output parameters.

3. Influence of the ball screw stiffness on the positioning accuracy of the feed kinematic linkage

The ball screw stiffness, associated with the use of indirect measuring has a major influence on the positioning accuracy of the moving element. In order to increase the ball screw stiffness and for eliminating the radial-axial backlash, two preloaded ball screw nuts are used and the ball screw bearings are preloaded, as well.
Figure 2. Preloading of the ball screw double nut with enlarged pitch of 20 mm.

In figure 2, it may be noticed the mode of preloading the double nut with enlarged pitch of 20 mm. The value of the preloading force $F_{pr}$ may be controlled by adjusting the dimension of the spacer $2$ between the nut bodies $1$ and $3$ of the ball screw $4$, so that, when applying an external force, like the feed force $F_a$, the radial-axial backlash between the ball screw and nut to be eliminated.

Figure 3. Variable stiffness of a single nut.  
Figure 4. Stiffness of two preloaded nuts.

The allure of the force-deformation diagram in case of a single nut is shown at figure 3, where two areas may be noticed: the first one is an axial backlash $I$, and the second one is the axial elastic deformation $II$ generated in the contact zone of the balls. The profile of the diagram of the elastic deformation may be noticed: in the contact area, at low values of the load $F$, the deformation is high and when the force $F$ increases, the elastic deformation decreases (The position of the diagram versus the horizontal modifies). This aspect shows the fact that the stiffness of a single nut is variable, having a low stiffness (high elastic deformation) at low values of the load $F$ and increased stiffness at high values of the load. A relation of proportionality to express the relation of above between load $F$ and deformation $d$ is given by (1):

$$d = CF^{2/3}$$

Where: $C$ – constant of proportionality.

The consequences of using a single nut on the ball screw and an indirect measuring system would lead to a major influence on the positioning accuracy since the two components $I$ and $II$ would be found in the positioning error. In order to improve the positioning error two preloaded nuts are used. These two nuts are located into a box. This solution will eliminate the axial backlash $I$, as well as the
first part of the elastic component of the zone $II$, where the stiffness is low, as it may be noticed in figure 4. The two nuts $I$ and $2$ are preloaded at a value $F_{p}$, thus annulling the backlashes $I_{1}$ and $I_{2}$, as well as the elastic deformations $II_{p1}$ and $II_{p2}$, in the areas where the stiffness is low. In this situation, where the nuts are preloaded at the value $F_{p}$, there will be a stiffness increase, explained by the fact that when acting with a thrust (axial) force $F_{a}$, the elastic deformation will begin, as on the diagram shown at figure 4, at the point $A$. In this context, upon the requirement with the force $F_{a}$ of the two preloaded nuts there will be a decrease of this force with the value $F_{1}$, resulted from preloading, as shown at figure 4 and the driving force becomes $F_{a1}=F_{a}-F_{1}$. If the ball screw had a single nut (diagram at figure 3) the driving force would be $F_{a}$, whilst if there were two preloaded nuts, $F_{a}$ would become $F_{a1}$. At the same time, in the second case the deformation is generated in the area $II$ beginning with the point $A$, see figure 4, from where the stiffness of a single nut is high. The determining of the preloading force $F_{p}$ is also studied by other authors and ball screw manufacturers but there is no consensus. A solution being applied with good results consists of determining the value of the preloading force at idle running of the respective axis that is approximately 20% of the available force. This solution is fundamental through the fact that the positioning of the axis is performed at idle running and for the maximum loading of the axis the elastic deformations of the nuts are on the area where the nut has and increased stiffness. Additionally, during heavy cutting duties and low quality requirements, the work piece accuracy will also be affected by other deformations of the machine, work piece, setting devices and tool.

Figure 5. Thrust (axial) bearing system by means of ball bearings of the ball screw.

Figure 5, presents the assembly of the ball screw of diameter $d=50$ mm, having the increased pitch $p=20$ mm and fitted with a thrust bearing system at both ends. This system confers a good stiffness to the ball screw 2, regardless the motion sense of the nut box 3 that is driven by the feed force $F_{a}$, both towards the bearing 1 and the bearing 4. In this case, any driving force axially oriented on the box containing the two preloaded nuts will require axially the ball screw 2. If the driving force $F_{a}$, has the sense shown at figure 5, then the part of the ball screw at the right side of the nut box 3 will be required at stretching, whilst the left side part will be required at compression. For this assembly, very frequently used, the axial stiffness of the ball screw can be written as per the relation (2):

$$R_{s} = \frac{\pi d_{0}^{2}E}{41000} \left( \frac{L}{L-I} \right)$$

(2)

Where: $d_{0}$ – ball screw diameter measured at the interior of the thread (mm); $E$ – elasticity module of the steel the ball screw is made of (21000 daN/mm$^{2}$); $L$ – unsupported length of the ball screw between its bearings (mm); $I$ – intermediary distance measured from the unsupported end of the ball screw up to the half of the nut (mm). When the ball screw nut is located at the middle of the ball screw unsupported length ($I=L/2$), the ball screw stiffness will be minimal, $R_{s, min}$ and it will be written as per the relation (3):
In this case, the maximum deformation of the ball screw will be exactly at this point at half distance between the bearings 1 and 4 and it will be written as per the relation (4):

$$ \Delta e_{\text{max}} = \frac{F_a}{R_{s\text{ min}}} $$  \hspace{1cm} (4)

Where: $\Delta e_{\text{max}}$ – maximal elastic deformation of the ball screw (μm); $F_a$ – driving force (N); $R_{s\text{ min}}$ – minimal stiffness of the ball screw (N/μm).

4. Experimental trials

The experimental trials have carried out on the milling machine model CNC (CPFH 500) at idle running. For measuring the position of the moving element the machine is equipped with a position encoder with indirect measuring located on the driving motor shaft and the measurement results has been compared in tandem with the calibrated measuring system, consisting of the laser interferometer of the Mechatronic Laboratory of the University of Bacau. The indirect measuring system of the machine does not have the capacity to offset the errors caused by the axial backlashs and deformations of the ball screw and, in this case, the improvement of the positioning accuracy $P$ is imposed through the improvement of the ball screw stiffness. The bearing system of this CNC machine tool is axially stiffened by thrust ball bearings and the translation motions of the axes are performed by the ball screw double nuts. The diameter of the ball screws is 50 mm and their increased pitch is 20 mm. The machine travels $L$ are: $X=700$ mm; $Y=600$ mm and $Z=600$ mm. Three sets of trials have been performed, by following up the effect of modifying the preloading forces in the ball screw nuts and in the thrust (axial) bearings of the ball screw. For preloading the thrust bearings the initial force was 320 daN. Afterwards, by modifying the size of the inner spacing ring between bearings, the preloading forces have had values higher than 450 daN. They reached up to 550 daN. For the double nut the initial preloading force was 380 daN. This preloading force has been modified by replacing the spacing ring between the ball screw nuts in two stages, so that higher values of the preloading force could be reached, i.e. 490 daN, respectively 570 daN. Through these modifications of the preloading forces in the ball screw bearings and double nut, the improvement of the positioning accuracy of the moving element has been followed up. Figure 6 shows the preloading diagram of the ball screw bearings where it may be noticed that preloading the bearings at higher rates of the preloading force leads to an improvement of the positioning accuracy of the moving element. For the ball screw, the preloading force rate of 550 daN renders the best improvement of the positioning accuracy, up to the value of 0.0061 mm. This peak is located in the median area of the ball screw. Under these conditions the motor current consumption (torque) at idle running has increased.

**Figure 6.** Preloading the ball screw bearings.  \hspace{1cm} **Figure 7.** Preloading the ball screw nuts.
In case of preloading the ball screw nuts, as per figure 7, the loading at a force of 570 daN, has brought the better improvement of the positioning accuracy at idle running. It reached 0.0058 mm. Instead, after a running time interval, an increase of the temperature of 8ºC has been noticed on the ball screw-nut assembly that will generate thermo-mechanical deformations. These deformations will be affecting the positioning accuracy obtained at the preloading force value 570 daN. For optimisation, in case of stiffening the bearings, the preloading force value of 450 daN has been chosen and, for stiffening the double nut, this value has been 490 daN. These values, correlated with the software corrections in those 1000 points, improve the positioning accuracy by up to 0.005 mm, and the machine will comply with the admitted tolerance limits.

5. Conclusions
In this work the influence of the ball screw-nut assembly and its bearings on the positioning accuracy of the moving element has been studied. It has been established that the stiffening of the ball screw assembly can contribute to improving the positioning accuracy of the moving element by preloading the double nuts and bearings of the ball screw with suitable forces. When high preloading forces are applied, the positioning accuracy can be improved considerably, but it has as negative aspects the loss in torque of the driving motor and the temperature increase of the ball screw-nut assembly that may generate thermo-mechanical deformations. For the ball screw assembly stiffening to have positive aspects only, it is necessary an optimisation of the preloading forces that, corroborated with the electronic corrections in the 1000 points, would lead to improving the machine positioning accuracy, so that it complies with the admitted tolerance limits.

6. References
[1] Mauro S, Pastorelli S, 2015, Influence of Controller Parameters on the Life of Ball Screw Feed rives, Advances in Mechanical Engineering 7(8), 1–11.
[2] Fuhua L, 2017, An Improved Dynamic Model of Preloaded Ball Screw Drives Considering Torque Transmission and its Application to Frequency Analysis, Advances in Mechanical Engineering 9(7) 1–11.
[3] Oyanguren A, Larranaga J, 2018, Thermo-mechanical Modelling of Ball Screw Preload Force Variation in Different Working Conditions, Int J Adv Manuf Technol 97, 723–739.
[4] Huang Y, 2018, Diagnosis of the Hollow Ball Screw Preload Classification Using Machine Learning, Appl. Sci. 8, 1072.