Backlash Decrease System of Reducers/Gearboxes in Feed Kinematical Linkage Structure of CNC Machine Tools

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Abstract. Feed kinematic linkages of the CNC machine tools are built in canned cycle (closed loop) and their structures include reducers/gearboxes, as well. Because of the mechanical backlash between the flanks of the teeth of gearings, disturbances occur in the performance of a kinematic linkage as a whole (in terms of speed and position loops), thus leading to instability. This work presents a new system for decreasing the mechanical backlash that is based on a new type of cylindrical gearings with tapered teeth, allowing the backlash adjustment between the flanks of the joint teeth. The modality of acting in terms of decreasing the backlash on a reducer in the structure of a feed kinematic linkage is also detailed.

1. Introduction
The nowadays progress in the manufacture of CNC machine tools is mainly determined by the continuous improvement of the feed kinematic linkage performances, providing better and better dynamic behaviours, positioning accuracy and wide ranges for speed variation. Such requirements impose a permanent adaptation of the machine tool designer to the new performance conditions especially in case of the contouring numerical controls where a strict correlation is imposed for the simultaneous motion along two or more kinematic axes, in order to obtain more and more intricate plane and spatial profiles [1, 2]. The problematic of the feed kinematic linkages of the CNC machine tools consists mainly of improving the static and dynamic parameters, reified through the positioning accuracy, path error and stability of the kinematic axis. The static parameters are different in function of the type of kinematic linkage, type and purpose of the machine tool. In achieving the static parameters a major role is held by the quality of the mechanical reducer belonging to the feed kinematic linkage structure, located, in most of the cases, inside of the position loop, directly influencing the loop behaviour. The response time characteristic to a canned cycle control system for a feed kinematic linkage is influenced by the value of the mechanical backlash in the reducer, so a time difference results between the input and output signals [3]. In correlation with the response time, the second factor in the dynamic of a feed kinematic linkage affected by the presence of the mechanical backlash is the stability of the system as a whole. A feed kinematic linkage with the speed and position circuits is stable further to a variation on the input when, after leaving the stable balance condition, the system returns to the stationary mode after the variation disappearance on the input [4]. The presence of the backlash in a feed kinematic linkage causes either the deviation of the slide from the programmed value or permanent oscillations around the stationary value of the programmed positioning point. The size of the mechanical backlash in the reducer depends on the number of gearings inside it and, eventually, on the transmission ratio of the reducer. Feed kinematic linkages
may be linear or circular, where the presence of a mechanical reducer is indispensable. Thus, in case of circular feed kinematic linkages, especially on the rotary tables of milling, boring and milling and drilling machines, reducers of a high transmission ratio \(i \geq 100\) are used, that leads to a high value of the mechanical backlash. In case of linear feed kinematic linkages using the rack and pinion mechanism as the last element to transform the rotation motion into linear motion, a reducer is placed after the servomotor. This reducer has a medium transmission ratio \(i \geq 20\) where the value of the mechanical backlash is also high. A lower transmission ratio \(i \geq 2\) is met on the reducers of feed kinematical linkages that have as final transformation element a ball screw mechanism. Because the accuracy of CNC machine tools has reached values of microns, the mechanical backlash of the reducers is imposed to very low or, as much as possible, even eliminated.

2. Block Structure of a Feed Kinematic Linkage of CNC Machine Tools
The location of the reducer into the block structure of the linear feed kinematic linkage is shown at figure 1, where it is integrated inside the position loop \(BP\). The control signal \(e_1\) that means the preset part of the command is compared in the block \(C\) to the signal \(e_0\) emitted by the position encoder \(TP\), having as result the difference signal \(e_1 - e_0\) that is amplified into the frequency converter \(CF\), so that the output current \(i\) is applied to the A.C. motor \(MCA\). This motor develops the torsion torque \(M_t\) at its output that further on is applied to the input shaft of the reducer \(R\). The rotation motion \(\theta\) obtained at the output shaft of the reducer \(R\) is transmitted to the ball screw \(S_b\) then converted into linear speed of the slide \(S\). It may be noticed that the speed loop \(BV\) picks up its signal from the rotary encoder \(TV\) located on the rotor of the motor \(MCA\). This signal is compared to the input signal and their difference is applied to the motor until the error is annulled, thus keeping the rpm constant on the motor shaft when the resistant torque is variable. From figure 1 it may be noticed the architectural location of each component of the feed kinematic linkage: the comparator \(C\) belongs to the numerical control; the converter \(CF\) is located in the electric cabinet; the motor \(MCA\), the reducer \(R\) and the ball screw \(S_b\) are located on the machine tool.

![Figure 1. Block diagram of the linear feed kinematic linkage with the speed and position loops](image1)

![Figure 2. Block diagram of the circular feed kinematic linkage with the speed and position loops](image2)
on the output shaft of the reducer $R$. In this case the ratio between the output rpm of the reducer that is
the rapid rpm of the table rotation and the maximum rpm developed by the motor determines the
maximum value of the transmission ratio of the reducer $R$. Each element in the structure of the two
block diagrams has its own transfer function based on which the transfer function of the entire feed
kinematic linkage can be obtained. The response of the feed kinematic linkage is determined by the
values of its components. In making a decision on the most suitable response of the feed kinematic
linkage, two criteria may be applied:

a) The kinematic linkage should respond rapidly a given input signal and the oscillation around a
static condition should be minimized. This criterion is suitable for machine tools that only
perform positioning, such as drilling machines and lathes etc. In such a case the reducer quality is
important but not very demanding in terms of the mechanical backlash. Major manufacturers of
machine-tools that achieve only positioning, use speed reducers having a mechanical backlash
greater than 6 minutes.

b) The kinematic linkage should respond rapidly a given input signal and the oscillation around a
static condition should be eliminated. This criterion applies to contouring kinematic linkages
where the presence of a light over-oscillation is unacceptable because during contouring any
oscillation will modify the required profile, as well as the roughness of the machined surface. In
this case it is required the backlash to be as low as possible, even eliminated if possible. The
reducers used on such contouring feed kinematic linkages have very low backlash rates, i.e. 2-4
minutes [5].

3. Influence of the Mechanic Backlash on the Stability of the Feed Kinematic Linkage
The presence of the mechanical backlash in the reducer that is a component of the canned cycle of the
feed kinematic linkage makes the output signal to occur after a certain time from the occurrence of the
input signal. Figure 3 shows the ideal response of an element with idle time where upon the signal $X_i$, the output $X_e$ will be:

$$X_e = X_i(t - \tau) \quad (1)$$

where: $\tau$ - is the idle time and $t$ - is the time.

![Figure 3. Ideal response of an element with idle time](image)

![Figure 4. The transfer place of the feed kinematic linkage: U-real part and V- imaginary part of the transfer function](image)
The mechanical backlash in the reducer makes the input signal to come up at the output after a time \( \tau \) whose size depends on the value of the mechanical backlash. In this case the reducer may be considered an element with idle time and has the output rate \( X_{\tau}(t) \) of the form:

\[
X_{\tau}(t) = X_{e}(t - \tau)
\]  

(2)

The influence of the mechanical backlash in the reducer on the stability of the feed kinematic linkage with speed and position loops may be approached in three ways:
- To make sure that the feed kinematic linkage is stable.
- To determine the critical value of the idle time for which the feed kinematic linkage keeps its stability.
- To modify the parameters of the transient duty until the feed kinematic linkage is brought to the status of stability.

The first way may be applied by determining the transfer function of the feed kinematic linkage with the speed and position circuits, then by determining the response of the kinematic linkage. Practically, this algorithm is costly and after knowing the transfer function of the feed kinematic linkage, various stability criteria will be applied for determining the system behavior. By supposing that the feed kinematic linkage is stable, it has the zero backlash in the reducer and it has a transfer place \( Y(i\omega) \) like that one shown at figure 4. The transfer place of the feed kinematic linkage having a reducer with mechanical backlash, in other words a feed kinematic linkage having an idle stroke, \( Y(\tau,i\omega) \), from the transfer place \( Y(i\omega) \) can be obtained through the simple clockwise rotation of each one of its points at the angle \( \alpha = \tau\omega \), where \( \tau \) means the time corresponding to the mechanical backlash in the reducer and \( \omega \) means the pulsation. Thus, the point \( A \) in figure 4 will reach the position \( B \) and after such a deformation of \( Y(i\omega) \), this may intersect the sector \((-1 \div i\infty)\) that is instability. The increase of the mechanical backlash in the reducer makes \( \tau = \tau_{cr} \), case when \( Y(i\omega) \) will pass through the point \((-1, i0)\), situation when it enters the instability zone. The critical value of the mechanical backlash that is the value for which the feed kinematical linkage is at the limit of stability, is determined by the critical value of the angle \( \alpha \). It may be noticed from the figure 4 that the system will be stable if:

\[
\tau < \tau_{cr} = \frac{\alpha_{cr}}{\omega_{cr}}
\]  

(3)

where \( \tau_{cr} \) means the time corresponding to the maximum backlash in the reducer. Because many factors contribute to performing the feed kinematic linkage such as the elastic deformations of the reducer components and the ball screw mechanism, whose effect superposes the idle time of the mechanical backlash in the reducer, the real value of the idle stroke will be established experimentally. In this case the stability problem of the feed kinematic linkage will be sorted out by passing to a compromise between establishing the value of the transient duty and stability, by modifying the parameters that define the acceleration and deceleration.

4. Cylindrical Gearing with Tapered Teeth

This new type of gearing is used for decreasing the mechanical backlash between the tooth flanks. The current solutions with possibilities for decreasing the backlash between the tooth flanks are known; they are using two kinematic branches so that, during their relative angular rotation, the backlash is decreased. The disadvantage of this kind of solutions consists of the double sized dimensions of the kinematic transmission (one size for one rotation direction, another size for the other direction). Eventually, a voluminous kinematic transmission will result, with double inertia.

The cylindrical gearing with tapered teeth is composed of two gears 1 and 2 whose teeth are tapered; their two flanks \( a \) and \( b \) respectively are different, in terms that the flank \( a \) is straight whilst the flank \( b \) has a tilting angle within \( 2^\circ \) to \( 5^\circ \) having the tilting on the gear 1 in one sense and on the gear 2 in the opposite sense, figure 5. Through the axial motion \( I \) of one of the gears 1 and 2, the
backlash between the tooth flanks of the wheels will be decreased. The presence of the two flanks \( a \) and \( b \) of a tooth leads to performing two gearings with two different distances between the axes of the same gearing and, for making the gearing possible, i.e. to keep the same distance between axes, one gearing will be corrected in terms of angle. When establishing the sum of the values of the correction indicators, the difference of the distances between axes has to be considered and the values of each correction indicator will be obtained according to the stages of the following reasoning, based on obtaining the same geometric elements of the two gear \( I \) and \( 2 \) for the flank \( a \) and \( b \): outer diameter, inner diameter and tooth height.

**Figure 5.** Cylindrical gearing with tapered teeth

**Reasoning:**

1. The sum of the correction indicators \( \xi_s \) will be calculated; the correction of the flank \( a \) with straight teeth has been randomly chosen and the correction corresponding to the flank \( b \) remains uncorrected. With the help of the variation coefficient of the distance between axes \( \lambda_0 \), where:

\[
\lambda_0 = \frac{A_0 - A}{A}
\]

\( \xi_s \) will be determined.

\[
\xi_s = \xi_1 + \xi_2
\]

The symbols mean: \( A_0 \) – distance between axes at uncorrected gearing; \( A \) - distance between axes at corrected gearing; \( z_1 \) - number of teeth of the gear \( I \); \( z_2 \) - number of teeth of the gear \( 2 \); \( \xi_1 \) – correction coefficient of the gear \( I \); \( \xi_2 \) – correction coefficient of the gear \( 2 \).

2. The values of each correction indicator \( \xi_1 \) and \( \xi_2 \) for the gearing corresponding to the flank \( a \) with straight teeth having imposed the outer diameters of the gears \( I \) and \( 2 \) from the gearing of the flank \( b \).

\[
D_{\eta b} = m[z_1 - 2(f_0 + c_0 - \xi_1)] + 2h
\]

\[
D_{\epsilon_{2b}} = m[z_2 - 2(f_0 + c_0 - \xi_2)] + 2h
\]
Where the symbols mean: $D_{1b}$ – outer diameter of the gear 1 for the gearing corresponding to the flank $b$; $D_{2b}$ – outer diameter of the gear 2 for the gearing corresponding to the flank $b$; \(m\) - module; \(f_0\) – reference coefficient of the tooth head height; \(c_0\) – reference coefficient of the bottom backlash; \(h\) – tooth height.

After making the difference between the two relations (6) it will lead to an equation that, along with the relation (5) are forming a resolving system, that has the unknown rates $\xi_1$ and $\xi_2$.

3. After having known $\xi_1$ and $\xi_2$ the outer diameter will be determined as well as the inner diameter and the tooth height for the corrected gearing that has the flank $a$, values that are close to the values of the uncorrected gearing that has the flank $b$.

5. **System of Mechanical Backlash Decrease by Using Cylindrical gearings with Tapered Teeth**

Figure 6 shows a constructive system of a reducer that uses three cylindrical gearings with tapered teeth. Each shaft on which one of the cylindrical gears with tapered teeth is located has an axial or radial-axial bearing, built into a flange that can move axially through the adjustment of a spanner. Thus, the decrease of the mechanical backlash begins with the gearing composed of the gears 2 and 3 that, by adjusting the spanner 4, allows the axial motion of the flange 5. For the gearing composed of the gears 6 and 7 the spanner 8 will be adjusted that allows the axial motion of the flange 9 and for the gearing composed of the gears 10 and 11 the spanner 12 will be adjusted that allows the axial motion of the flange 13. In this way, the motion form the motor 1 can be transmitted through the three cylindrical gearings with tapered teeth having a decreased mechanical backlash, whose value has been adjusted from the outside of the reducer housing.

![Figure 6. Reducer with cylindrical gearings with tapered teeth](image)

6. **Conclusions**

The system for decreasing the mechanical backlash that uses cylindrical gearings with tapered teeth has several advantages in comparison to the systems using two kinematic branches:
- It allows a realistic sizing of the cylindrical gearing without over sizing, that involves an adequate inertia and small overall dimensions.
- It provides a safe and easy adjustment of the backlash between flanks through the controlled axial motion of one of the two gears that compose the gearing.
-It allows the adjustment of the backlash between flanks by intervening from the outside of the reducer housing, without being necessary to access this from the inside of the housing, as constructively imposed by most of the reducers.

The new cylindrical gearing with tapered teeth can be built with tilted teeth, so that both flanks of a tooth may have a tilting angle that differs by $2^\circ$ to $3^\circ$ from a flank to the other. The usage of the cylindrical gearings with tapered teeth in the kinematic transmissions of the mechatronic systems renders mechanical backlashes of 1 to 2 minutes on the output shaft of the reducer. At the same time, after one number of running hours, further to the wear, the backlash between flanks may be recalibrated through interventions form the reducer outside only.

References

[1] Abrudan I, Ungureanu V, 2013, The influence of side mechanical factors on the positioning accuracy of CNC Machine Tools, Expert Coference – Trends in the Development of Machinery and Associated Technology, Barcelona, pp. 793-796.

[2] Bell Y, 2011, Compensation system of backlash and pitch errors of the feed kinematical linkages of the numerical control machine tools, Progresivie Tehnologii I Sistemi Masinocstroenia, Donet, pp. 165-173.

[3] Bezier P, 2009, Optimizing the transmission ratio on mechanical systems for decreasing the response time, Buletinul BIP Iasi, Sectia Constructii de Masini, Iasi, pp. 555-558.

[4] Stan G, 2007, Transmisii mecanice in bucla inchisa, Editura Junimea, ISBN 973-37-0490-3, Iasi.

[5] Ungureanu V, 2006, Researches concerning the influence of the cutting forces on the transient period of the kinematical linkages of machine tools, Annals of DAAAM for 2003 & Proceedings, “Intelligent Manufacturing & Automation: Learning From Nature”, Tuzla, pp. 529-530.