Prediction of the insensitivity zone in the design of the feed drive CNC lathe

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Abstract. The processing on CNC machines of parts of a complex profile with multipass shaping schemes is accompanied by multiple changes in the direction of movement of the working bodies (reverse) during the cutting process and at idle running. When reverse, loss of information is possible due to the presence of gaps in the chain of feed drive and elastic deformations determined by constant loads overcome by the drive. Permanent loads consist of cutting forces and friction in the feed drive chain. Information loss is defined as a position insensitive zone at a position. To determine the insensitive zone, a technique is proposed that takes into account the differential accounting of friction losses in the mechanical feed drive chain. To estimate the elastic deformations in the drive chain, design schemes are proposed and analytical dependencies obtained. Differentiated consideration of friction in determining the insensitive zone allows you to outline the main ways to improve feed drives. It is shown that the magnitude of the preload on the thrust bearings and the tensile forces of the lead screw during assembly do not have a significant effect on the size of the insensitive zone. Even less impact on the dead zone has a preload change in the transfer screw-nut rolling. It was determined that the main ways to reduce the insensitivity zone is to increase the rigidity of the screw-and-nut rolling transmission anchor to the machine carriage and reduce friction in the guides. In addition, an analysis of the layout decisions of the lead screw assembly (one-sided and two-sided termination) according to the insensitive zone criterion showed that the use of one-sided termination of the lead screw is more rational.

1. Introduction
The use of metal-cutting equipment with CNC is an objective necessity, due to the ever-increasing demands for accuracy, productivity, concentration of operations and multi-nomenclature during metalworking.

Effective use of CNC machines is possible only if they ensure parametric reliability, when the main operating characteristics of the machine meet the established requirements.

The operational characteristics of CNC machines are formed from two components: accuracy standards for machines of this type (for example, for lathes [1]) and additional requirements related to the operation of the machine in automatic mode from the CNC system [2].

Evaluation of the performance characteristics of the created machine structures has proven procedures and clearly defined requirements for metrological support [3-5]. However, in predicting these characteristics, at the design and manufacturing stage of the machine, very generalized dependencies are used that do not take into account the design features.

CNC machines are most effective when processing parts of a complex profile with multipass shaping schemes. The processing according to such shaping schemes is accompanied by multiple changes in the direction of movement of the working bodies (reversals) during the cutting process and
at idle strokes. In case of reverse, information loss is possible due to the presence of gaps in the chain of feed drives and elastic deformations determined by constant loads overcome by the drive. In this regard, much attention is paid to the design of feed drives of CNC machines [6-12].

To estimate the possible loss of information during the reverse, GOST 27843-2006 provides for such an indicator as a position insensitive zone (IZ) at a position: the difference between the average unilateral positional deviations obtained for the two directions of approach to the position.

2. Determination of the insensitive zone

In the general case, the IZ can be determined by the formula [13]:

\[ \Lambda = \Delta \Sigma + \frac{2F}{c}, \]  

where \( F \) is a friction force in the chain of feed drive; \( c \) - stiffness of the chain of feed drive; \( \Delta \Sigma \) - total clearance in the chain of feed drive.

Due to the fact that in modern designs of feed drives, elements with gaps are rarely found, it is possible to accept in formula (1) \( \Delta \Sigma \approx \Delta \). However, formula (1) does not take into account the distribution of friction forces on the elements of the mechanical drive chain.

The typical design of the CNC machine feed drive includes the following elements in which friction losses occur: gearbox, the transfer screw-nut rolling (SNR), lead screw supports, guides.

The mechanical drive system can be represented as a chain multi-mass system with concentrated masses connected by elastic inertia-free sections. The value of IZ can be determined from the system of equations describing the steady-state motion of the drive, as doubled the twist angle of the drive (the angle of rotation of the first mass before starting off from the last mass):

\[ \begin{align*}
  c_1 (\varphi_1 - \varphi_2) &= M_1; \\
  c_1 (\varphi_1 - \varphi_2) &= c_2 (\varphi_2 - \varphi_3) + M_2; \\
  \vdots \\
  c_{n-1} (\varphi_{n-1} - \varphi_n) &= M_n,
\end{align*} \]  

where \( \varphi_1, \ldots, \varphi_n \) are angles of rotation of concentrated masses; \( c_1, \ldots, c_{n-1} \) are stiffness coefficients of the inertia-free sections; \( M_1 \) is the driving moment (the moment developed by the driving engine); \( M_2, \ldots, M_n \) are moments determined by friction losses, given to the corresponding concentrated masses.

In the general case, the rigidity of the inertia-free sections is not constant, but depends nonlinearly on the deformation.

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\[ c_n = a_n (\varphi_n - \varphi_{n+1}) b_n^{-1}, \]  

where \( a_n \) and \( b_n \) are coefficients depending on the site design, manufacturing technology and assembly of its parts.

Solving the system of equations (2) with regard to (3), we determine the magnitude of the IZ:

\[ \Lambda = 2 \sum_{i=1}^{n-1} \left( \frac{M_j}{a_i} \right)^{\frac{1}{b_i}}. \]  

In the absence of the drive gearbox, the formula (4) will look like:

\[ \Lambda = 2 \left( \frac{M_2 + M_3}{a_1} \right)^{\frac{1}{b_1}} + \left( \frac{M_3}{a_2} \right)^{\frac{1}{b_2}}, \]
where $M_2$ is the moment determined by friction losses in the bearings of the lead screw and the transfer SNR; $M_3$ is the moment determined by friction losses in the guides; $a_1$ and $b_1$ are parameters that determine the total torsional rigidity of the coupling between the engine and the lead screw and the body of the lead screw from the coupling to the running nut; $a_2$ and $b_2$ are the parameters that determine the total axial stiffness of the bearings of the lead screw, the transfer SNR and the body parts, in which the supports and the transfer SNR are located, which are reduced to a twisting system.

Stiffness $c_1$ is defined as the sum of two series-connected constant torsional stiffnesses: the stiffness of the coupling between the engine and the lead screw and the stiffness of the body of the lead screw from the coupling to the running nut.

Due to the independence of stiffness $c_1$ on the angle of twist of the drive, we have $b_1 = 1$ and $c_1 = a_1$, where the value $a_1$ is based on the design and dimensions of the coupling and the lead screw.

The elements defining the value $c_2$ are essentially non-linear.

3. Determination of the axial stiffness of the feed drive

The design scheme for determining the axial stiffness of a drive with double-sided fastening of the lead screw, taking into account the use of preloads in its elements, can be represented as a combination of serial and parallel joints of elastic elements (figure 1), where $c_0^I$ and $c_0^II$ are the rigidity of the left and right body parts supports, respectively; $c^I$ and $c^II$ are the rigidity of the bearings (taking into account the package of fastening and adjusting parts) of the left and right bearings, respectively; $c_G$ - stiffness of semi-nuts the transfer SNR; $c_F$ - apron stiffness; $K_1^V$ and $K_2^V$ - stiffness of the body parts of the lead screw; $F$ - friction force in the guides. The presented calculation scheme can be replaced by an equivalent one (figure 2), where $A$ and $B$ are equivalent stiffness, respectively, of the left and right sides of the lead screw with supports.

![Figure 1](image1.png)

**Figure 1.** The design scheme of axial stiffness of the feed drive with double-sided fastening of the lead screw

![Figure 2](image2.png)

**Figure 2.** Equivalent design for axial feed drive stiffness
Total axial stiffness of the chain

$$c_{ax} = \frac{F}{\delta_F + \delta_G + \delta_V},$$

(6)

where \(\delta_F; \delta_G; \delta_V\) are elastic deformation, respectively, of the apron, the transfer SNR and the lead screw with the supports.

According to experiments conducted on CNC lathes mod. 16B16F3 and 16B16T1, the deformation of the apron is linearly dependent on the operating operational loads: \(\delta_F = F / c_F\), where \(c_F = \text{const}\) (figure 3).

\[\text{Figure 3. Deformation of the apron machine mod. 16B16T1 when applied to the caliper load } F\]

The value \(\delta_G\) can be determined by writing the equilibrium equations for a ball nut under the action of a force \(F\):

$$F = c'_G(\Delta_G + \delta_G) - c'_G(\Delta_G - \delta_G),$$

(7)

where \(\Delta_G\) is preload in half-nuts.

In accordance with the accepted notation (3) and according to [14], we have

$$c'_G = a_G \sqrt{\Delta_G + \delta_G}; \quad c'_G = a_G \sqrt{\Delta_G - \delta_G}.$$

(8)

The value of \(\delta_G\) is approximately an order of magnitude less than \(\Delta_G\), since the force of \(F\) does not exceed 10% of the effort required to relieve the preload in the transfer SNR. Therefore, \(\delta_G\) can be found their equations (7) taking into account (8) and using the expansion of expressions \((1 \pm \delta_G / \Delta_G)^{1.5}\) in a power series:

$$\delta_G \approx \frac{F}{3a_G \sqrt{\Delta_G}}.$$

(9)

Deformation \(\delta_V\):

$$\delta_V = \frac{F}{A + B},$$

(10)

where

$$A = \frac{1}{K_Y^p} + \frac{1}{2c' \left(\Delta'_1 + \Delta'_2\right)} + \frac{1}{2 \sqrt{c'_0 P'_*}};$$

(11)
\[ B = \frac{1}{K_2^V} + \frac{1}{2c^{II}(\Delta_1^{II} + \Delta_2^{II})} + \frac{1}{2\sqrt{c_0^II p_*^II}}; \]  

(12)

\( \Delta_1^l \) and \( \Delta_2^l \) are preliminary deformation of the left and right bearings of the left support, respectively; \( \Delta_1^r \) and \( \Delta_2^r \) are preliminary deformation of the left and right bearings of the right support, respectively; \( p_*^l \) and \( p_*^{II} \) are efforts acting respectively in the left and right supports and determined by the working force \( F \) and the pre-stretching of the lead screw:

\[
p_*^l = \left[ \frac{\tau(Q - \sqrt{Q^2 - F^2}) + \frac{1}{2}(\Delta_0^l + \Delta_0^{II})}{\psi(\sqrt{Q + F} - \sqrt{Q} - \tau(\sqrt{Q - F} - \sqrt{Q}) + NF} + \right.
\]

\[
\left. + \frac{1}{2}NF(\sqrt{Q + F} + \sqrt{Q})}{\psi(\sqrt{Q + F} - \sqrt{Q} - \tau(\sqrt{Q - F} - \sqrt{Q}) + NF} \right]^{\gamma^2},
\]

(13)

\[
N = \frac{2}{K_1^V + K_2^V} + \frac{1}{2c^{II}(\Delta_1^l + \Delta_2^l)} = \frac{2}{K_1^V + K_2^V} + \frac{1}{2c^{II}(\Delta_1^{II} + \Delta_2^{II})},
\]

(16)

where \( Q \) is the preload force in the lead screw.

4. Testing methods for determining the insensitivity zone

Calculations carried out to drive the machine mod. 16B16T1 with the following initial data:

\( c^l = 21.6 \text{ N}/\mu\text{m}; \quad c^{II} = 25.0 \text{ N}/\mu\text{m}; \quad c_0^l = 70.0 \text{ N}/\mu\text{m}; \quad c_0^{II} = 15.0 \text{ N}/\mu\text{m}; \)

\( \Delta_1^l + \Delta_2^l = 21.7 \mu\text{m}; \quad Q = 970 \text{ N} \) showed (table 1) that the value of \( A + B \) is almost independent of the force \( F \). Thus, formula (6), taking into account the linear dependence of \( \delta_F; \delta_G; \delta_V \) on the force \( F \), can be written as

\[
\frac{1}{c_{ax}} = \frac{1}{c_F} + \frac{1}{3\delta_G\sqrt{\Delta_G}} + \frac{1}{A + B},
\]

(17)

and in the expression of stiffness \( c_2 \) we will have \( b_2 = 1 \) and \( c_2 = a_2 \).

| Equivalent stiffness, N/\mu m | Load \( F \), N |
|-----------------------------|---------------|
| \( A \)                     | 100 300 500 700 900 |
| 199.6                       | 203.9 207.8 211.4 215.3 |
| 134.4                       | 130.7 126.8 122.9 120.2 |
| 334.0                       | 334.6 334.6 334.3 335.5 |

As follows from the above expressions, the value of \( c_2 \) depends on the preload in the transfer SNR and the support lead screw.
The preload created when assembling the drive has a significant effect not only on the rigidity of the chain, but also on the friction in the system, that is, on the magnitude of the moment $M_2$.

The dependence of the idling moment of the transfer SNR on the preload is determined by the dependencies given in [14]. The total frictional moment in the supports depends on the design of the supports and can be determined using the dependencies of [15]. Friction in sliding guides often has the nature of mixed friction, the strength of which is determined by the well-known dependencies of the theory of hydrodynamic friction.

The calculation of the size of the dead zone according to the formula (5) for the drive of machine feeds mod. 16B16T1 with the following initial data: caliper weight 2340 N; coefficient of friction in the guide 0.18; the preload in the transfer SNR 3000 N; the torsional stiffness of the coupling 400 Nm/rad gave a value of 24.9 μm. At the same time, the calculation by formula (1) gives a value of 37.5 μm, that is, overestimated compared to that found by formula (5) by 50%.

5. Conclusions

Differentiated consideration of friction in determining the insensitivity zone allows you to outline the main ways to improve feed drives.

As the analysis of the calculation results shows, a decrease in the preload on the outer thrust bearings of the lead screw from 4000 N to 2000 N with a constant amount of stretching of the lead screw leads to a decrease in $IZ$ by only 6.5%. An increase in the tensile force of the lead screw from 750 N to 1500 N with a constant preload on the external thrust bearings (4000 N) leads to a decrease in $IZ$ by 2.8%. This suggests that the amount of preload on the thrust bearings and the tensile forces of the lead screw during assembly do not have a significant effect on the magnitude of the $IZ$. A change in the preload in the transfer SNR has even less of an effect on $IZ$.

Thus, the main ways to reduce $IZ$ is to tighten the design of the apron (or completely eliminate this element by changing the location of the lead screw on the machine) and reducing friction in the guides. So, for example, a decrease in the friction coefficient in the machine tool guides mod. 16B16T1 from 0.18 to 0.1 by applying pads of fluoroplastic, Teflon, etc. reduces the magnitude of the $IZ$ from 24.9 μm to 15.8 μm, and an increase in the rigidity of $c_F$ by a factor of 10, ceteris paribus, reduces the magnitude of the $IZ$ by 1.8 times.

As the experience of assembling machines with double-sided insertion of the lead screws with their simultaneous stretching has shown, there are difficulties in ensuring the required preloads in the bearings and the tensile forces of the lead screw. Therefore, the expected benefits of bilateral termination are often not achieved. At the same time, the replacement of a bilateral one-sided termination leads to an increase in $IZ$ only by 15%. Therefore, the analysis of the layout decisions of the lead screw assembly (one-sided and two-sided termination) according to the criterion of $IZ$ shows that the use of one-sided termination of the spindle screw is more rational.

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