Improving the quality of tightened joints of metal-cutting machines

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Abstract. Tightened joints using threaded assembly are widely used in modern metal cutting machines. Reliable operation of such joints is determined by the value of the required initial tightening and its stability during operation of the machines. It has been established that the cause of the weakening of the thread in joints is their self-loosening at significant relative lateral displacements of the fastened parts, at which the screw tilts with slipping in the thread and along the supporting surface of its head. The self-loosening in the bolted joint of the wedge adjusting devices regulating the rolling of the multi-purpose machines of the drilling-milling-boring group, loaded with an inertial load, which reduces their geometric accuracy, is considered. A model has been developed for the process of creating an interference fit and the conditions under which the screws of adjusting devices are unscrewed are determined. Measures have been developed to determine the rational parameters of screws and their tightening values, ensuring its stability during the operation of the machines.

1. Introduction
The operability of metal-cutting machines is largely determined by the joint stability of the tightening, which is usually provided by various threaded assemblies (bolted, screw, stud, etc.) [1-2]. In modern machines, threaded parts make up over 60% of the total number of parts. The widespread use of threaded joints is explained by their structural and technological advantages. However, the experience in operating machines showed that the reliable operation of threaded joints is largely determined by their correct initial tightening and its stability during operation.

To ensure the required value of the initial tightening of threaded assemblies, various device designs have been developed that allow controlling its value with different accuracy [3].

2. Status of the issue. Formulation of the problem
Operational observations of machines operating with significant dynamic loads show that their reliability is largely determined by the joint stability of the tightening, which may decrease over time. Moreover, the share of failures associated with loosening the tightening of threaded joints (LTTJ) may account for more than 20% of failures of the mechanical system of the machine. In some cases, the recovery of the machine after such a failure may take more than 10% of the time between failures [4-5].

Despite the fact that the stability of the tightening of threaded joints has a significant impact on the performance of the machines, to date there is a limited number of works devoted to the study of this issue, and their results do not allow to formulate any generalizations [4, 6].
In this regard, threaded joints are identified whose initial tightening forces may weaken, and indicate in the operating instructions the frequency of their tightening only after prolonged monitoring of the operation of the machines, or extensive, various locking means are used, which complicates the design and increases manufacturing costs [2, 7, 8]. At the same time, to improve the performance of machines, all these issues should be addressed at the design stage.

There are various explanations for the mechanisms of LTTJ both with longitudinal and transverse loads [9, 10, 11, 12, 13]. Analysis of literary sources showed that self-loosening under longitudinal loads occurs with more significant impacts than with transverse ones. This applies to construction, agricultural machinery, exploration equipment, etc. In metal-cutting machines, self-loosening of threaded joints occurs at significant relative lateral displacements of the fastened parts. In this case, the most convincing explanation of the self-loosening process is bending and tilting the screw when slipping in the thread and along the supporting surface of its head, arising under significant transverse loads [4, 10, 13].

From the standpoint of the loading of machine joints leading to LTTJ, two groups of joints can be distinguished: joints loaded with a significant transverse dynamic load, in which the acting forces can be determined by calculation, and joints loaded only with an inertial load during a kinematic disturbance of oscillations with uncertain parameters. As an example of a joint loaded with a significant transverse dynamic load, studies were carried out at the joint "cutter assembly" of an aggregate-milling machine. The obtained results and improvement measures are described in [4]. As the joints loaded with inertial load, the joints of the wedge adjusting devices for creating interference fit in the roller guides of the multi-purpose drilling, milling and boring group were selected.

3. The effect of the tightened joints of the wedge adjusting devices regulating the rolling of the machines on their geometric accuracy and dynamic quality

Operational observations of four machines of the IR500PMF4 model during the year showed that the interference in their guides can change over time, which leads to a reorientation of the nodes and, accordingly, a decrease in geometric accuracy [14]. For instance, as for the multipurpose machine IR500PMF4, when the table is reoriented by 1 μm, the deviation from the perpendicularity of its movement relative to the rack is 1 μm.

The reorientation of the table and the column of each machine were determined using two measuring heads, with a graduation of 1 μm, which were fixed in magnetic stands mounted on the guide rails. The reorientation value was established by the difference in the readings of these heads when moving the table or column in different directions by 20 μm. Moreover, the small amount of displacement made it possible to exclude the influence of inaccuracies in the manufacture of guide rails on the measurement result. Figure 1 shows the reorientation Ω of the table (a) and the column (b) of the model IR500PMF4 machine from changes in interference in the rolling guides over time.
Figure 1 shows that during the operation of the machines the reorientation of the table increased significantly, and the columns practically did not change. This is due to more stable working conditions of the column guides than the table. With an increase in the intensity of processing modes, the magnitude of the reorientation of nodes increases. So the machine on which the rough operations were carried out was put out for repair, because of the significant reorientation of the table, ledges appeared when milling planes.

To assess the effect of interference in the rolling guides on the reorientation of nodes and their dynamic characteristics under production conditions, experiments were conducted in which the reorientation and the amplitude of the vibration velocity of the headstock of the model IR500PMF4 machine were measured. Fluctuations in the spindle head were excited by the rotation of an unbalanced disk mounted in the spindle. With a perturbing force of 30÷100daN, the perturbation frequency was 45÷75Hz. Measurements showed that with an increase in the tightening torque of the screws moving the adjustment wedge from 1 to 3N∙m, the reorientation of the headstock decreased by 5 μm, and the resonant amplitude of the vibration velocity (at a frequency of 66 Hz) - by 1.5 m/s. Thus, it was found that during the operation of multi-purpose machines, intense dynamic loads weaken the interference in the rolling guides, which impairs their geometric accuracy and dynamic characteristics of the nodes and the machine as a whole.

4. Theoretical and experimental studies of the process of creating interference in the rolling guides and its changes during the operation of machines

The process of creating interference and the behavior of the elements of the wedge adjusting devices under dynamic loads was investigated on a special stand consisting of a rectangular slider covered by roller bearings that are built into the housing and wedge adjusting devices, the design of which is shown in figure 2.

![Figure 2](image)

**Figure 2.** Wedge adjusting device for creating interference in the rolling guides of multi-purpose machines.

Such devices have one moving 2 and two fixing screws 1. The wedge adjustment device works as follows: first, using the screw 2, the movable wedge 3, moving by means of the roller support, creates the required interference, and then the screws 1 fix its position.

To control the tightening forces of the screws, they were made tensometric. The displacements of the movable wedges with roller bearings were controlled by contactless inductive sensors. The experiments showed that when tightening the fixing screws, significant displacements of the moving wedge are possible, as a result of which the tightness decreases. In this case, the force acting on the moving screw increases sharply and can exceed the permissible.

An examination of the forces acting on the elements of the wedge adjusting device and the displacements of these elements made it possible to obtain the dependences for determining the required tightening torque $P_z$ of the moving screw to create the required interference value, the full force $P_{2z} = P_z + \Delta P_z$, acting on the moving screw after tightening the fixing screws, and the forces $P_{k}$=...
acting on the wedge after creating an interference fit and shifting the wedge when tightening all the screws. In these equations, $\Delta P_z$ is the additional force in the moving screw that occurs when the fixing screws are tightened with a total force $Q$. Based on the condition that the movements of the elements of the wedge adjusting device are equal, $\Delta P_z = Q \cdot C / C_\Sigma$ is obtained, where $C$ is the axial stiffness of the elements of the wedge connection and the slats, $C_\Sigma$ is the total axial stiffness of the elements of the wedge joint and the moving screw.

The obtained dependences made it possible to evaluate the change in the force acting on the wedge when creating interference fit. When tightening only the transfer screw, this force is maximum and its value $P_K = P_Z$. When tightening the fixing screws with a small force, providing elastic displacement of the wedge, this force first decreases to zero, and then changes sign and increases in absolute value; from the moment a wedge slip occurs, the absolute value of this effort begins to decrease. To study the behavior of the elements of the wedge adjusting device in dynamic processes using a centrifugal vibrator mounted on the stand body, as well as when lightly tapping the case with a copper hammer, oscillations in these elements were excited in three mutually perpendicular directions. During the experiments, the displacement of the moving wedge and the changes in the forces in the moving and fixing screws were observed. The results of the experiments showed that harmonic vibrations, the amplitude and disturbing force of which corresponded to those usually operating in machines, caused insignificant changes in the interference and efforts in the screws. Impulse (shock) disturbances of a certain intensity (characterized by the amplitude of vibration acceleration of the stand body), acting in a plane perpendicular to the axis of the screws, led to the self-loosening of the moving screw. This can be explained by the fact that the pulsed perturbations caused the wedge to oscillate at its natural frequencies, which, according to calculations, are 3÷6 Hz.

Figure 3 shows the dependence of the amplitude $a$ of the acceleration of the test bench housing on the force $Q$ of the fixing screws with an initial tightening force of the moving screw of 100 daN.

![Figure 3. Dependence of the amplitude $a$ of the acceleration of the test bench housing on the force $Q$ of the fixing screws with an initial tightening force of the moving screw of 100 daN.](image)

Figure 3 shows that with an increase in the force $Q$, the amplitude of vibration accelerations increases (the same is noted with an increase in the force $P_Z$). Impulse disturbances acting in the direction of the axis of the screws do not cause their self-loosening, which corresponds to the results of previous studies [4].

Measurements of the vibration acceleration amplitude of the IR500PMF4 machine table at the installation sites of the side roller bearings showed that when the satellite table is lowered after it is turned and the latter is captured to change it, the vibration acceleration level is close to that observed on the bench when the moving screw was loosened. If for any external load on the wedge-type control device, the force $P_K = 0$, then the tightness and the forces acting on the screws do not change.

5. Research results for wedge adjusting devices for rolling guides

Based on the results of the studies, the following requirements were formulated for wedge adjusting devices:
1) the necessary tightness in the bearing should be created at the maximum allowable tightening forces of the moving and fixing screws in order to reduce the risk of self-loosening;

2) the forces acting on the wedge after creating an interference must be close to zero in order to ensure the stability of its position under dynamic loads.

A comparative analysis of the most common designs of wedge adjusting devices (Figure 4) showed that when using locknuts or threaded sleeves in them (figure 4a and 4b), the second requirement is not fulfilled, since the forces acting on the wedge after creating an interference are not minimal. According to this indicator, a device (figure 4c) with conventional fixing screws is more preferable. To fulfill the first requirement, it is necessary that the diameters of the moving and fixing screws be the same. A comparison of wedge adjusting devices with a wedge slope of 1:10 and 1:20 showed that the difference in screw diameters is negligible.

It was found that the additional force acting on the wedge after creating interference depends on the tightening forces of the moving and fixing screws, as well as on the ratio of the stiffnesses of the elements of the wedge adjusting device. A ratio of stiffness at which the above requirements are met can be selected. Since the stiffness of the bar and the wedge is determined by their structural dimensions, it is easiest to vary the stiffness of the moving screw.

![Figure 4. Designs of wedge adjusting devices.](image)

6. Conclusion
The rational parameters of the moving screw of the wedge adjusting device (figure 4 c), the design of the Lipetsk Machine Tool Plant, were calculated. In some cases, it is necessary to use a moving screw of considerable length, which is relatively simple to implement in the device shown in figure 4d.

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